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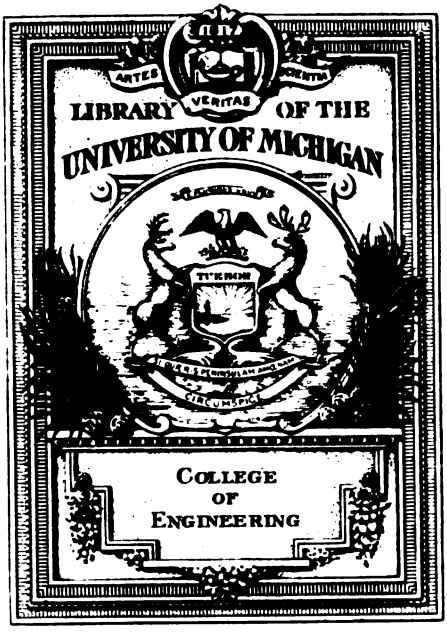
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MARINE ENGINEERS' HANDBOOK

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Marine Engineers' Handbook

PREPARED
BY A STAFF OF SPECIALISTS

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PREFACE

This handbook is compiled for the use of operating and designing engineers and of students. To cover the main subjects and the many ramifications of marine engineering it has been necessary to enlist the services of thirty specialists. Each specialist in turn has received the hearty cooperation of others in the same field. The best endeavor to coordinate the different practices has been put forth in order that all data may be reliable and unbiased.

Marine engineering differs from mechanical engineering in its state of development. While mechanical engineering practice is standardised in most of its branches, marine engineering is in a stage of transition. But one type of prime mover, the reciprocating engine, can be said to have reached its final development. Such being the case, and considering that many matters of design practice are jealously guarded as trade secrets, the task set for the contributors to this book has been severe. Combine this with the fact that much of the work represented herein was performed during War times when but little leisure was available, and the reader will understand the deep obligation that the Editor-in-Chief feels toward the authors of the various sections. A patriotic pride in aiding our growing merchant marine has been a controlling factor with many of them.

All the large shipyards have contributed to this work. In every instance credit has been given in the text. Hundreds of marine manufacturing companies have furnished data. The list is too large to attempt here but the reader will meet our friends on every page.

Many friends in the Bureau of Steam Engineering, Navy Department, have given valuable aid in searching records and checking data. The U. S. Naval Experiment Station at Annapolis, Md., furnished and checked much test data. The U. S. Naval Fuel Oil Testing Plant has made possible the very complete section on Oil Fuel Burning. The Journal of the American Society of Naval Engineers has furnished much useful material.

Professor Lionel S. Marks, the Editor-in-Chief of the Mechanical Engineers' Handbook, has allowed a most generous use of material from his classical work and has contributed much kindly and useful advice in the early stages of this undertaking.

The Editor-in-Chief wishes to express thanks to the publishers for assistance at many stages of this work, in obtaining material and contributors, but most of all for their courteous patience in the face of the many unavoidable delays throughout.

Finally, the reader must bear in mind that this handbook is a pioneer in its field. The trail has not always been plain to read. This edition aims at a general survey of the new territory. The lack of standardization in Marine engineering has made its presentation in handbook form somewhat difficult.

FRANK W. STERLING.

CAUTION

Many of the structures and processes herein described are protected by U. S. and foreign patents. It would unnecessarily lengthen the text to state this fact with reference to each case to which it would apply; but designers and users are cautioned against utilizing structures or processes without previously assuring themselves that such use is not subject to patent restrictions.

CONTENTS

(For Alphabetical Index, see p. 1459)

SECTION 1

MATHEMATICAL TABLES AND FORMULÆ AND MECHANICS OF RIGID BODIES

Mathematical Tables and Formulæ

	Page
Common Logarithms	2
Trigonometric Functions	6
Decimal Equivalents	11
Circles (Circumferences and Areas)	12
Conversion Tables:	
Acceleration of Gravity	14
Forces	14
Lengths	15
Areas	15
Volumes and Capacities	16
Velocities	16
Masses	17
Pressures	17
Energy, Work, Heat	18
Power	18
Density	19
Heat Transmission and Conduction	19
Trigonometric Formulæ	20
Differential Formulæ	22
Indefinite Integrals	23
Definite Integrals	25

Mechanics of Rigid Bodies

	Page
Definitions	27
Units	27
Kinematics	29
Statics	32
Kinetics	38

SECTION 2

NON-FERROUS AND FERROUS METALS AND HEAT

Non-ferrous Metals and Alloys

	Page
Pure Metals	41
Bronzes	53
Brasses	56
Strength of Metals and Alloys at High Temperatures	68
Bearing Metals	68
White Metal Alloys	78

Iron and Steel

	Page
Ores of Iron	79
Extraction of Iron from Ore	80
Blast Furnace Products	81
Metallography	82
Chemistry of the Purification Processes	87
Wrought Iron	88
Steel	89
Ingots	97
Mechanical Treatment	98
Heat Treatment	98
Influence of Chemical Composition on Physical Properties	103
Case Carburization or Partial Cementation	109
Cast Iron	111
Ferro Alloys	119
Specifications for Wrought Iron	121
Specifications for Steel	122
Specifications for Cast Iron	148
Protective Coatings for Iron and Steel	153

Screw Threads, Bolts and Nuts

	Page
Screw Threads	158
Bolts and Nuts	161

Oxy-acetylene Welding

Gases for Welding and Cutting	163
Burners	164
Expansion and Contraction	165
Welding Rods	166
Fluxes	167
Welding	168
Cutting	171

The Thermit Welding Process

Composition	172
Kinds of Thermit	172
Physical Properties	173
Large Welds	173
Crank Shaft Repairs	175
Pipe Welds	176

Heat

	Page
Temperature Measurement	177
Expansion	178
Specific Heat	180
Freezing Mixtures	183
Melting Points of Solids	183
Freezing and Boiling Points	185
Heat of Fusion and Latent Heat	185
Transmission of Heat	186
Thermal Conductivities	189
Thermodynamics	195
Perfect Gases	200
Gas Mixtures	201
Expansion of Gases	226
Ideal Cycles with Perfect Gases	204
Air Compression	206
Vapors, Properties of	208
Steam Tables	209
Refrigerants	243
Expansion of Vapors	247
Mixture of Gases and Vapor	223
The Steam Engine	226
Refrigeration	228
Flow of Gases and Vapors	227
Throttling	237

SECTION 3

FUELS AND COMBUSTION

OIL FUEL BURNING

Coal

	Page
Composition	241
Analysis	241–243
Classification	243
Marine Coals	243
Selection and Inspection of Coal for Marine Use	245
Spontaneous Combustion and Storage	245
U. S. Navy Coal	246
Standard Steaming	247
Sampling	247

Oil Fuel

Petroleum and Fuel Oil	250
Properties of Petroleum	251
Costs, Coal and Oil Fuel	252
Oil Fueling Stations	253
Advantages, Coal and Oil Fuel	254
Physical Properties	256
Heating Value	256
Viscosity	256
Viscosity Conversion Curves	260
Specific Gravity	262
Expansion	262
Specific Heat	267
Flash Point	268
Fire Point	269
Centrifuge	271
Conservation	273
Safety Precautions:	
Burning	274
Carrying	275
Loading	276

Commercial Gasoline

Properties	277
Specifications	277
Tests	278

Combustion

Air Required for	279
Excess Air	280
Volume Contraction	280
Combustion Products	280

Oil Fuel Burning

Atomization and Atomizers	281
Air for Combustion	287
Air Registers	292
Boiler Furnace Design	297
Furnace Insulation	297
Relations Between Rate of Combustion, Heating Surface, Furnace Volume and Air Pressure	299
Fuel Oil Equipment	301
Boiler Room Instruments	315
Operation	322
Precautions	326

SECTION 4

MARINE BOILERS

	PAGE
Development	333
Designation	339
Ideal Requirements	343
General Design	343
Effect on Design of	
Fuel	344
Draft	344
Steam Pressure	352
Steam Output	354
Size and Number of Units	357
Space and Weight	358
Boiler Sizes and Weights	358
Design of Scotch Boilers	385
Rules and Regulations Governing Boiler Construction	390
Factor of Safety	391
Hydraulic Tests	392
Riveted Joints	393
Calking	405
Welded Drums	406
Bumped or Dished Heads	406
Manholes and Handholes	408
Cylindrical Shells, Riveted	408
Flat Surface Stayed	410
Stays	413
Furnaces	415
Combustion Chambers Girders	420
Combustion Chamber Tube Sheet	421
Cylindrical Tube Sheets, Water Tube Boiler Shells	422
Diagonal Ligaments	422
Scotch Boiler Construction	426
Tubes	434
Boiler Setting	438
Boiler Mountings	440
Furnace Mountings	453
Superheaters	461
Care of Boilers	
Preservation of Idle Boilers	464
Prevention of Exterior Corrosion	464
Raising Steam	464
Removal of Oil	465
Internal Corrosion	466
Boiler Repairs	475
Fuel and Firing	477
Time Firing	478
Cleaning Fires	479
Air Required for Combustion	480
Analysis of Gases of Combustion	481
Removal of Scale	487
Boiler Tests	488
Boiler Materials, Inspection Requirements	514

SECTION 5

TURBINES AND MECHANICAL REDUCTION GEARS

Turbines

	PAGE
Properties of Steam	528
Adiabatic Expansion	529
Actual Expansion	533
Nozzles	538
Impulse Blading	543
Curtis Blading	546
Parsons Blading	548
Clearances and Leakage	555
Rotational Losses	561
Provisional Proportions of Turbine:	
Wheel Efficiencies	563
The Limiting Factors of Capacity	563
Provisional Proportions of an Impulse Turbine	564
Provisional Proportions of a Parsons Turbine	566
Gaging	570
Provisional Proportions of Direct-coupled Parsons Marine	572
Provisional Proportions of Astern Turbines	572
Provisional Proportions of Curtis-Rateau Turbines	572
Turbine Design:	
Design of a Simple Impulse Turbine	573
Design of a Simple Curtis Turbine	575
Design of a Rateau Turbine	577
Design of a Curtis-Rateau Turbine	580
Design of a Parsons Turbine	581
Design of a Curtis-Parsons Turbine	588
Miscellaneous Thermodynamic Considerations	589
Martin's "New Theory of the Steam Turbine"	590
Mechanical Design:	
Design of Turbine Discs	593
Critical Speed of Rotors	605
Design of Rotor Shafts	609
Drum Rotors	609
Shaft Couplings	609
Blading	611
Cylinder Construction	624
Turbine Drains	628

	PAGE
Dummy Pistons, Glands, Labyrinths	631
Casing Glands	633
Bearings	635
Balancing of Rotors	640
Miscellaneous Mechanical Equipment	646
Micrometer Gear	649
Bridge Gages	650
Governors and Valve Gears	653
Lubrication System:	
Bearings	661
Lubricating Oils	662
Types of Lubricating Systems	663
Oil Pumps	667
Strainers	669
Oil Filters	671
Oil Coolers	677
Tanks	678
Oil Piping	679
Piping Connections:	
Steam Lines	679
Traps and Drains	681
Gland Piping	682
Foundations	683
Lifting Gear	687
Mechanical Reduction Gearing	692
Electrical Gearing	692
U. S. S. Jupiter	693
U. S. S. New Mexico	694
U. S. S. Tennessee	701
S. S. Mjölner	703
S. S. Wulsty Castle	704
Hydraulic Gearing	707
Performance:	
Effects of Variation in Operating Conditions	710
Turbine Testing	712
Tests Data and Trials	715
Trial Performance of U. S. S. Wadsworth	720
Electric Drive Performance	720
Performance of Small Steam Turbines	722
Comparison of Marine Steam Turbines and Other Prime Movers	723
Types of American Steam Turbines	732
Westinghouse Marine Turbines	732
General Electric Turbines	745
De Laval Steam Turbines	755
Kerr Turbines	762
Poole Marine Turbines	767
Direct Drive Marine Propelling Turbines	774
The Ljungstrom Turbine	779
Sturtevant Turbines	784
Terry Steam Turbines	788
Engine Room Layouts	794
Mechanical Reduction Gears	
Types	806
Pinion	808
Gear Wheels	813
Testing	815
Care and Operation	815

SECTION 6

RECIPROCATING ENGINES

	PAGE
Classification, Development	821
Efficiency	
Thermal	827
Mechanical	830
Determining Size of the Engine	831
Proportioning of Engine Parts	837
Cylinders	840
Framing and Beds	850
Reciprocating Parts	855
Bearings	864
Shafting	871
Valve Gear	877
Reversing Gear	899
Turning Gear	904
Attached Pumps	906
Piping and Fittings	911
Drafting Room Method of Laying Down the Engine	918
Balancing	920
Analytical Method	921
Graphical Method	926
Means of Improving Balance	932
Turning Effort	934
Weights and Dimensions	938
Machining and Tolerances	939

SECTION 7

MARINE DIESEL ENGINES

	Page
Classification	945
The Diesel Engine:	
Operating Processes	947
Pressures	948
Ratio of Delivered to Indicated Power	948
Internal Power Consumption	949
Fuel Supply and Injection	949
Scavenging	950
Lubrication	950
Cooling	950
Fuel Consumption	951
Overload Capacity	954
Lubricating Oil Consumption	954
Four Cycle vs. Two Cycle	954
Steam vs. Diesel	956
Types of Diesel Engines	957
Semi-Diesel Engines	962
Fuel Oils	963

SECTION 8

VACUUM AND CONDENSERS

	Page
Vacuum Measurement	967
Limits of Vacuum	969
Thermodynamics	970
Condensers:	
Main Surface Condensers	975
Surface Condenser Design	976
Scoop Condensers	982
Keel Condensers	983
Auxiliary Condensers	980
Jet Condensers	983
Jet Condenser Design	987
Circulating Pumps	988
Air Pumps:	
Wet Air Pumps	991
Dry Air Pumps	992
Wet and Dry Air Pumps	997
Vacuum Augmenters	1000
Condensate Pumps	1000
Accessories	1001
Arrangement of Condensing Plants	1006

SECTION 9

SHIP FORMS AND POWERING AND SCREW PROPELLERS

Ships Forms and Powering

Ships Forms

	Page
Choice of Dimensions	1013
Length	1013
Breadth	1013
Draft and Depth	1014
Economical Speed	1014
Prismatic Coefficient	1014
Curve of Sectional Areas	1014
Parallel Middle Body	1018
Vertical Prismatic Coefficient	1021
Height of Metacenter	1021
Load Water Line	1022
Body Plan	1022

Simplified Forms

	Page
McEntee's Form	1023
Whittlesey's Form	1023

Powering of Ships

	Page
Steven's Formula	1025
Effective Horsepower from Model Tests	1026

Screw Propellers

	Page
Thrust Deduction and Wake Gain	1028
Slip Block Coefficients:	
Wing Screws	1030
Single Screw Ships	1031
Tunnel Boats	1031
Determinntion of Thrust Deduction Factor	1032
Mean Relative Tip Clearance	1033
Resistance of Hull Appendages	1034
Basic Condition for Analysis and Design	1035
Cavitation	1043
Analysis and Design of Propellers	1046
Design of the Propeller	1055
Forms for Computation	1056
Standard Propeller Hubs	1062

	Page
Dummy Pistons, Glands, Labyrinths	631
Casing Glands	633
Bearings	635
Balancing of Rotors	640
Miscellaneous Mechanical Equipment	646
Micrometer Gear	649
Bridge Gages	650
Governors and Valve Gears	653
Lubrication System:	
Bearings	661
Lubricating Oils	662
Types of Lubricating Systems	663
Oil Pumps	667
Strainers	669
Oil Filters	671
Oil Coolers	677
Tanks	678
Oil Piping	679
Piping Connections:	
Steam Lines	679
Traps and Drains	681
Gland Piping	682
Foundations	683
Lifting Gear	687
Mechanical Reduction Gearing	692
Electrical Gearing	692
U. S. S. Jupiter	693
U. S. S. New Mexico	694
U. S. S. Tennessee	701
S. S. Mjölner	703
S. S. Wulsty Castle	704
Hydraulic Gearing	707
Performance:	
Effects of Variation in Operating Conditions	710
Turbine Testing	712
Tests Data and Trials	715
Trial Performance of U. S. S. Wadsworth	720
Electric Drive Performance	720
Performance of Small Steam Turbines	722
Comparison of Marine Steam Turbines and Other Prime Movers	723
Types of American Steam Turbines	732
Westinghouse Marine Turbines	732
General Electric Turbines	745
De Laval Steam Turbines	755
Kerr Turbines	762
Poole Marine Turbines	767
Direct Drive Marine Propelling Turbines	774
The Ljungstrom Turbine	779
Sturtevant Turbines	784
Terry Steam Turbines	788
Engine Room Layouts	794
Mechanical Reduction Gears	
Types	806
Pinion	808
Gear Wheels	813
Testing	815
Care and Operation	815

SECTION 6

RECIPROCATING ENGINES

	Page
Classification, Development	821
Efficiency	
Thermal	827
Mechanical	830
Determining Size of the Engine	831
Proportioning of Engine Parts	837
Cylinders	840
Framing and Beds	850
Reciprocating Parts	855
Bearings	864
Shafting	871
Valve Gear	877
Reversing Gear	899
Turning Gear	904
Attached Pumps	906
Piping and Fittings	911
Drafting Room Method of Laying Down the Engine	918
Balancing	920
Analytical Method	921
Graphical Method	926
Means of Improving Balance	932
Turning Effort	934
Weights and Dimensions	938
Machining and Tolerances	939

SECTION 7

MARINE DIESEL ENGINES

	Page
Classification	945
The Diesel Engine:	
Operating Processes	947
Pressures	948
Ratio of Delivered to Indicated Power	948
Internal Power Consumption	949
Fuel Supply and Injection	949
Scavenging	950
Lubrication	950
Cooling	950
Fuel Consumption	951
Overload Capacity	954
Lubricating Oil Consumption	954
Four Cycle vs. Two Cycle	954
Steam vs. Diesel	956
Types of Diesel Engines	957
Semi-Diesel Engines	962
Fuel Oils	963

SECTION 8

VACUUM AND CONDENSERS

	Page
Vacuum Measurement	967
Limits of Vacuum	969
Thermodynamics	970
Condensers:	
Main Surface Condensers	975
Surface Condenser Design	976
Scoop Condensers	982
Keel Condensers	983
Auxiliary Condensers	980
Jet Condensers	983
Jet Condenser Design	987
Circulating Pumps	988
Air Pumps:	
Wet Air Pumps	991
Dry Air Pumps	992
Wet and Dry Air Pumps	997
Vacuum Augmenters	1000
Condensate Pumps	1000
Accessories	1001
Arrangement of Condensing Plants	1006

SECTION 9

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	Page
Choice of Dimensions	1013
Length	1013
Breadth	1013
Draft and Depth	1014
Economical Speed	1014
Prismatic Coefficient	1014
Curve of Sectional Areas	1014
Parallel Middle Body	1018
Vertical Prismatic Coefficient	1021
Height of Metacenter	1021
Load Water Line	1022
Body Plan	1022

Simplified Forms

McEntee's Form	1023
Whittlesey's Form	1023

Powering of Ships

	Page
Steven's Formula	1025
Effective Horsepower from Model Tests	1026

Screw Propellers

Thrust Deduction and Wake Gain	1028
Slip Block Coefficients:	
Wing Screws	1030
Single Screw Ships	1031
Tunnel Boats	1031
Determinntion of Thrust Deduction Factor	1032
Mean Relative Tip Clearance	1033
Resistance of Hull Appendages	1034
Basic Condition for Analysis and Design	1035
Cavitation	1043
Analysis and Design of Propellers	1046
Design of the Propeller	1055
Forms for Computation	1056
Standard Propeller Hubs	1062

SECTION 10

AUXILIARY MACHINERY

Evaporators and Distillers

	PAGE
Evaporators:	
Marine Service	1079
Types of Plants	1079
Capacity, Space, Weight	1080
Types of Evaporators	1080
Materials of Construction	1081
Accessories	1085
Methods of Installation	1086
Regenerative Compressor	1091
Piping Arrangement	1093
Operation	1094
Distillers:	
Marine Service	1096
Commercial Types	1097
Evaporator Feed Heaters	1098

Feed Water Heaters

Types and Materials	1103
Installation	1104
Size Required	1107
Design	1108
Operation	1110

Oil Coolers

Design	1111
Commercial Types	1113
Operation	1114

Centrifugal Pumps

Characteristic Curves	1115
Construction	1117
Methods of Driving	1122
Circulating Pumps	1123
Boiler Feed Pumps	1127

Reciprocating Pumps

Types	1131
Construction	1133
Commercial Specifications	1133
Tables of Commercial Sizes	1134
Cargo Oil Pumps	1149
Performance Factors	1151
Design Factors	1152
Valve Service	1153

Centrifugal Fans

Definitions, Formulæ and Types of Fans	1163
Type of Fan Impellers	1170
Parallel Operation	1175

Heating and Ventilating

Ash Handling Machinery, Miscellaneous

Heating	1177
Newport News S. B. & D. D. Co., Practice	1178
U. S. Naval Specifications	1180
Ventilation	1181
Ventilators	1184
Combined Heating and Ventilation	1189
Thermofan System	1189
Siroccozone System	1191
Ash Handling Machinery	1192
Whistle and Siren	1196

Air Compressors

Properties of Air	1198
Flow of Air	1199
Work of Compression	1200
Air Compressors	1201
Testing Air Compressors	1203
Operation and Maintenance	1204

Deck Auxiliaries

Steering Gear	1206
Automatic Follow up Gear for Steering Engine	1218
Rudder Torque	1218
Size of Steering Gear	1219
Steering Gear Control	1222
Windlasses	1230
Cargo Winches	1234
Electric Winches	1241
Warping Winches	1242
Capstans	1242

SECTION 11

PIPING, VALVES AND FITTINGS
REDUCING VALVES AND PUMP GOVERNORS
STEAM TRAPS
PIPE COVERING AND LAGGING

Piping, Valves and Fittings

	Page
Plans	1245
Materials	1254
Thickness	1261
Strength	1262
Pipe Bends and Expansion	1266
Pipe Connections:	
Pipe Threads	1271
Flanges	1272
Fittings	1284
Painting Pipes	1290
Zinc Boxes	1291
Valves	1291
Globe, Angle and Cross	1294
Check	1297
Gate	1299
Cocks and Blow-off	1301
Tables	1302
Safety and Relief Valves	1324

Reducing Valves and Pump Governors

Reducing Valves	1332
Pump Governors	1339

Steam Traps

	Page
Purpose	1340
Types	1342
Design	1343
Ratings and Capacities	1345
Installation, Care and Operation	1346
Testing	1347
Specifications	1347

Pipe Covering and Lagging

Heat Losses from Bare Metal Surfaces	1348
Losses Through and Savings from Insulation	1350
Standard Commercial Sizes, Pipe Covering	1358
Pipe Covering Specifications	1359

SECTION 12

MARINE ELECTRICAL INSTALLATION

	Page
Methods of Installation:	
Classification Society Rules	1363
Drawings	1364
Wire Sizes	1364
Protection of Conductors	1365
Types of Conductors	1366
Generating Sets	1367
Marine Switchboards	1371
Lighting System:	
Circuits	1371
Fittings	1372
Running Lights	1373
Lamps	1374
Searchlights	1375
Power System	1376
Signalling Systems:	
Engine Telegraphs	1377
Telephones	1377
Fire Alarms	1377
Submarine Signals	1378
Radio Telegraph	1378

SECTION 13

LUBRICATION AND LUBRICANTS

	Page
Lubrication	1381
Friction	1381
Lubricants:	
Mineral Oils	1383
Animal, Fish and Vegetable Oils	1383
Fixed Oils and Compound Oils	1383
Greases	1384
Graphite	1385
Selecting Lubricant for Service	1386
Engine and Machine Oil	1386
Forced Lubrication Oils	1388

	Page
Marine Engine Oils	1388
Cylinder Oils	1391
Air Compressor Oil	1392
Ice Machine Oil	1392
Motor Oils	1392
Physical Constants of Oils:	
Viscosity	1394
Flash and Fire	1394
Emulsification Test	1395
Water and Sediment	1395
Cold Test	1396
Specific Gravity	1397
Color	1397
Odor	1398
Carbon	1399

SECTION 14

MEASURING HORSEPOWER OF MARINE ENGINES

	Page
Power	1403
Indicators and Indicator Cards	1404
Essential Requirements of Good Indicator	1405
Indicator Springs	1405
Indicator Cord	1406
Reducing Motions	1406
Taking the Card	1408
Indicator Cards	1409
Combined Indicator Cards	1410
Effect of Varying Cut-off on Power Distribution	1411
Steam Consumption from Indicator Card	1412
Calculating the I. H. P.:	
M. E. P.	1412
Planimeters	1413
Manograph	1416
Torsion Meters:	
Shaft Calibration	1417
Gary-Cummings Torsion Meter	1420
Hopkinson Thring Torsion Meter	1423
Denny-Edgecombe Torsion Meter	1425

SECTION 15

TESTS, TRIALS AND INSPECTION OF MACHINERY, CLASSIFICATION SOCIETIES

Tests, Trials, Inspections

	Page
Tests for Machinery before same is placed on board	1429
Piping before same is placed on board	1430
Machinery after same is placed on board	1431
Machinery, preparatory to Trials	1433
Trials of Machinery	1434
Dock Trials	1435
Contract Acceptance Trials	1435
Commissioning, Post Repair and Laying Up Trials	1448

Classification Societies

	Page
Names and Addresses of Societies	1450
Steamboat Inspection Service	1450
Publications	1451
Legal Status	1451
Bureau of Standards	1452
Publications	1452
Legal Status	1452
American Bureau of Shipping	1452
Publications	1453
Legal Status	1453
American Society of Testing Materials	1453
Purpose	1454
Publications	1454
American Society of Mechanical Engineers	1454
Publications	1454
Legal Status	1454
Bureau of Explosives	1455
American Institute of Electrical Engineers	1455
Publications	1456
Legal Status	1456
Master Car Builder's Association	1456
Purpose	1456
American Railway Engineering Association	1457
Purpose	1457
Publications	1457
Legal Status	1457
America Institute of Weights and Measures	1457
Purpose	1457
Publications	1458
American Society of Naval Engineers	1458
Purpose	1458
Publication	1458

SECTION 1

MATHEMATICAL TABLES AND FORMULÆ AND MECHANICS OF RIGID BODIES

BY

MAYSON W. TORBET, Captain of Engrs., U.S.C.G., B. Mar. E., Mem. Council, A.S.N.E.

CONTENTS

MATHEMATICAL TABLES AND FORMULÆ

	Page
Common Logarithms	2
Trigonometric Functions	6
Decimal Equivalents	11
Circles (Circumferences and Areas)	12
Conversion Tables:	
Acceleration of Gravity	14
Forces	14
Lengths	15
Areas	15
Volumes and Capacities	16
Velocities	16
Masses	17
Pressures	17
Energy, Work, Heat	18
Power	18
Density	19
Heat Transmission and Conduction	19
Trigonometric Formulæ	20
Differential Formulæ	22
Indefinite Integrals	23
Definite Integrals	25

MECHANICS OF RIGID BODIES
By M. W. TORBET

Definitions	27
Units	27
Kinematics	29
Statics	32
Kinetics	38

COMMON LOGARITHMS (*special table*)

Number	0	1	2	3	4	5	6	7	8	9	Avg. dif.
1.00	0.0000	0004	0009	0013	0017	0022	0026	0030	0035	0039	4
1.01	0043	0048	0052	0056	0060	0065	0069	0073	0077	0082	
1.02	0086	0090	0095	0099	0103	0107	0111	0116	0120	0124	
1.03	0128	0133	0137	0141	0145	0149	0154	0158	0162	0166	
1.04	0170	0175	0179	0183	0187	0191	0195	0199	0204	0208	
1.05	0212	0216	0220	0224	0228	0233	0237	0241	0245	0249	
1.06	0253	0257	0261	0265	0269	0273	0278	0282	0286	0290	
1.07	0294	0298	0302	0306	0310	0314	0318	0322	0326	0330	
1.08	0334	0338	0342	0346	0350	0354	0358	0362	0366	0370	
1.09	0374	0378	0382	0386	0390	0394	0398	0402	0406	0410	
1.10	0.0414	0418	0422	0426	0430	0434	0438	0441	0445	0449	
1.11	0453	0457	0461	0465	0469	0473	0477	0481	0484	0488	
1.12	0492	0496	0500	0504	0508	0512	0515	0519	0523	0527	
1.13	0531	0535	0538	0542	0546	0550	0554	0558	0561	0565	
1.14	0569	0573	0577	0580	0584	0588	0592	0596	0599	0603	
1.15	0607	0611	0615	0618	0622	0626	0630	0633	0637	0641	
1.16	0645	0648	0652	0656	0660	0663	0667	0671	0674	0678	
1.17	0682	0686	0689	0693	0697	0700	0704	0708	0711	0715	
1.18	0719	0722	0726	0730	0734	0737	0741	0745	0748	0752	
1.19	0755	0759	0763	0766	0770	0774	0777	0781	0785	0788	
1.20	0.0792	0795	0799	0803	0806	0810	0813	0817	0821	0824	
1.21	0828	0831	0835	0839	0842	0846	0849	0853	0856	0860	
1.22	0864	0867	0871	0874	0878	0881	0885	0888	0892	0896	
1.23	0899	0903	0906	0910	0913	0917	0920	0924	0927	0931	
1.24	0934	0938	0941	0945	0948	0952	0955	0959	0962	0966	
1.25	0969	0973	0976	0980	0983	0986	0990	0993	0997	1000	3
1.26	1004	1007	1011	1014	1017	1021	1024	1028	1031	1035	
1.27	1038	1041	1045	1048	1052	1055	1059	1062	1065	1069	
1.28	1072	1075	1079	1082	1086	1089	1092	1096	1099	1103	
1.29	1106	1109	1113	1116	1119	1123	1126	1129	1133	1136	
1.30	0.1139	1143	1146	1149	1153	1156	1159	1163	1166	1169	
1.31	1173	1176	1179	1183	1186	1189	1193	1196	1199	1202	
1.32	1206	1209	1212	1216	1219	1222	1225	1229	1232	1235	
1.33	1239	1242	1245	1248	1252	1255	1258	1261	1265	1268	
1.34	1271	1274	1278	1281	1284	1287	1290	1294	1297	1300	
1.35	1303	1307	1310	1313	1316	1319	1323	1326	1329	1332	
1.36	1335	1339	1342	1345	1348	1351	1355	1358	1361	1364	
1.37	1367	1370	1374	1377	1380	1383	1386	1389	1392	1396	
1.38	1399	1402	1405	1408	1411	1414	1418	1421	1424	1427	
1.39	1430	1433	1436	1440	1443	1446	1449	1452	1455	1458	
1.40	0.1461	1464	1467	1471	1474	1477	1480	1483	1486	1489	
1.41	1492	1495	1498	1501	1504	1508	1511	1514	1517	1520	
1.42	1523	1526	1529	1532	1535	1538	1541	1544	1547	1550	
1.43	1553	1556	1559	1562	1565	1569	1572	1575	1578	1581	
1.44	1584	1587	1590	1593	1596	1599	1602	1605	1608	1611	
1.45	1614	1617	1620	1623	1626	1629	1632	1635	1638	1641	
1.46	1644	1647	1649	1652	1655	1658	1661	1664	1667	1670	
1.47	1673	1676	1679	1682	1685	1688	1691	1694	1697	1700	
1.48	1703	1706	1708	1711	1714	1717	1720	1723	1726	1729	
1.49	1732	1735	1738	1741	1744	1746	1749	1752	1755	1758	

Moving the decimal point n places to the right [or left] in the number requires adding $+n$ [or $-n$] in the body of the table (see p. 4).

COMMON LOGARITHMS (*special table, continued*)

Number	0	1	2	3	4	5	6	7	8	9
1.50	0.1761	1764	1767	1770	1772	1775	1778	1781	1784	1787
1.51	1790	1793	1796	1798	1801	1804	1807	1810	1813	1816
1.52	1818	1821	1824	1827	1830	1833	1836	1838	1841	1844
1.53	1847	1850	1853	1855	1858	1861	1864	1867	1870	1872
1.54	1875	1878	1881	1884	1886	1889	1892	1895	1898	1901
1.55	1903	1906	1909	1912	1915	1917	1920	1923	1926	1928
1.56	1931	1934	1937	1940	1942	1945	1948	1951	1953	1956
1.57	1959	1962	1965	1967	1970	1973	1976	1978	1981	1984
1.58	1987	1989	1992	1995	1998	2000	2003	2006	2009	2011
1.59	2014	2017	2019	2022	2025	2028	2030	2033	2036	2038
1.60	0.2041	2044	2047	2049	2052	2055	2057	2060	2063	2066
1.61	2068	2071	2074	2076	2079	2082	2084	2087	2090	2092
1.62	2095	2098	2101	2103	2106	2109	2111	2114	2117	2119
1.63	2122	2125	2127	2130	2133	2135	2138	2140	2143	2146
1.64	2148	2151	2154	2156	2159	2162	2164	2167	2170	2172
1.65	2175	2177	2180	2183	2185	2188	2191	2193	2196	2198
1.66	2201	2204	2206	2209	2212	2214	2217	2219	2222	2225
1.67	2227	2230	2232	2235	2238	2240	2243	2245	2248	2251
1.68	2253	2256	2258	2261	2263	2266	2269	2271	2274	2276
1.69	2279	2281	2284	2287	2289	2292	2294	2297	2299	2302
1.70	0.2304	2307	2310	2312	2315	2317	2320	2322	2325	2327
1.71	2330	2333	2335	2338	2340	2343	2345	2348	2350	2353
1.72	2355	2358	2360	2363	2365	2368	2370	2373	2375	2378
1.73	2380	2383	2385	2388	2390	2393	2395	2398	2400	2403
1.74	2405	2408	2410	2413	2415	2418	2420	2423	2425	2428
1.75	2430	2433	2435	2438	2440	2443	2445	2448	2450	2453
1.76	2455	2458	2460	2463	2465	2467	2470	2472	2475	2477
1.77	2480	2482	2485	2487	2490	2492	2494	2497	2499	2502
1.78	2504	2507	2509	2512	2514	2516	2519	2521	2524	2526
1.79	2529	2531	2533	2536	2538	2541	2543	2545	2548	2550
1.80	0.2553	2555	2558	2560	2562	2565	2567	2570	2572	2574
1.81	2577	2579	2582	2584	2586	2589	2591	2594	2596	2598
1.82	2601	2603	2605	2608	2610	2613	2615	2617	2620	2622
1.83	2625	2627	2629	2632	2634	2636	2639	2641	2643	2646
1.84	2648	2651	2653	2655	2658	2660	2662	2665	2667	2669
1.85	2672	2674	2676	2679	2681	2683	2686	2688	2690	2693
1.86	2695	2697	2700	2702	2704	2707	2709	2711	2714	2716
1.87	2718	2721	2723	2725	2728	2730	2732	2735	2737	2739
1.88	2742	2744	2746	2749	2751	2753	2755	2758	2760	2762
1.89	2765	2767	2769	2772	2774	2776	2778	2781	2783	2785
1.90	0.2788	2790	2792	2794	2797	2799	2801	2804	2806	2808
1.91	2810	2813	2815	2817	2819	2822	2824	2826	2828	2831
1.92	2833	2835	2838	2840	2842	2844	2847	2849	2851	2853
1.93	2856	2858	2860	2862	2865	2867	2869	2871	2874	2876
1.94	2878	2880	2882	2885	2887	2889	2891	2894	2896	2898
1.95	2900	2903	2905	2907	2909	2911	2914	2916	2918	2920
1.96	2923	2925	2927	2929	2931	2934	2936	2938	2940	2942
1.97	2945	2947	2949	2951	2953	2956	2958	2960	2962	2964
1.98	2967	2969	2971	2973	2975	2978	2980	2982	2984	2986
1.99	2989	2991	2993	2995	2997	2999	3002	3004	3006	3008

COMMON LOGARITHMS

Number	0	1	2	3	4	5	6	7	8	9	Avg. diff.
1.0	0.0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	
1.1	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	
1.2	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	
1.3	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	
1.4	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	
1.5	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	
1.6	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	
1.7	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	
1.8	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	
1.9	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	
2.0	0.3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
2.1	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
2.2	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
2.3	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	18
2.4	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	17
2.5	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
2.6	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
2.7	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
2.8	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
2.9	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
3.0	0.4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
3.1	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
3.2	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
3.3	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
3.4	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
3.5	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
3.6	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
3.7	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
3.8	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
3.9	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
4.0	0.6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
4.1	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
4.2	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
4.3	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
4.4	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
4.5	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
4.6	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	10
4.7	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
4.8	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
4.9	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9

See pages 2–8

$\log \pi = 0.4971$ $\log \pi/2 = 0.1961$ $\log \pi^2 = 0.9943$ $\log \sqrt{\pi} = 0.2486$
$\log e = 0.4343$ $\log (0.4343) = 0.6378 - 1$

These two pages give the common logarithms of numbers between 1 and 10, correct to four places. Moving the decimal point n places to the right [or left] in the number is equivalent to adding n [or $-n$] to the logarithm. Thus, $\log 0.017453 = 0.2419 - 2$, which may also be written $\bar{2}.2419$ or $8.2419 - 10$.

$$\log (ab) = \log a + \log b \qquad \log (a^N) = N \log a$$

$$\log \left(\frac{a}{b}\right) = \log a - \log b \qquad \log \left(\sqrt[N]{a}\right) = \frac{1}{N} \log a$$

SECTION 11

PIPING, VALVES AND FITTINGS
REDUCING VALVES AND PUMP GOVERNORS
STEAM TRAPS
PIPE COVERING AND LAGGING

Piping, Valves and Fittings

	Page
Plans	1245
Materials	1254
Thickness	1261
Strength	1262
Pipe Bends and Expansion	1266
Pipe Connections:	
Pipe Threads	1271
Flanges	1272
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Painting Pipes	1290
Zinc Boxes	1291
Valves	1291
Globe, Angle and Cross	1294
Check	1297
Gate	1299
Cocks and Blow-off	1301
Tables	1302
Safety and Relief Valves	1324
Reducing Valves and Pump Governors	
Reducing Valves	1332
Pump Governors	1339

Steam Traps

	Page
Purpose	1340
Types	1342
Design	1343
Ratings and Capacities	1345
Installation, Care and Operation	1346
Testing	1347
Specifications	1347

Pipe Covering and Lagging

Heat Losses from Bare Metal Surfaces	1348
Losses Through and Savings from Insulation	1350
Standard Commercial Sizes, Pipe Covering	1358
Pipe Covering Specifications	1359

SECTION 12

MARINE ELECTRICAL INSTALLATION

	Page
Methods of Installation:	
Classification Society Rules	1363
Drawings	1364
Wire Sizes	1364
Protection of Conductors	1365
Types of Conductors	1366
Generating Sets	1367
Marine Switchboards	1371
Lighting System:	
Circuits	1371
Fittings	1372
Running Lights	1373
Lamps	1374
Searchlights	1375
Power System	1376
Signalling Systems:	
Engine Telegraphs	1377
Telephones	1377
Fire Alarms	1377
Submarine Signals	1378
Radio Telegraph	1378

SECTION 13

LUBRICATION AND LUBRICANTS

	Page
Lubrication	1381
Friction	1381
Lubricants:	
Mineral Oils	1383
Animal, Fish and Vegetable Oils	1383
Fixed Oils and Compound Oils	1383
Greases	1384
Graphite	1385
Selecting Lubricant for Service	1386
Engine and Machine Oil	1386
Forced Lubrication Oils	1388

TRIGONOMETRIC FUNCTIONS (at intervals of 10′)

Annex—10 in columns marked *.

Degrees	Radians	Sines Nat.	Sines Log.*	Cosines Nat.	Cosines Log.*	Tangents Nat.	Tangents Log.*	Cotangents Nat.	Cotangents Log.	Radians	Degrees
0° 00′	0.0000	.0000	∞	1.0000	0.0000	.0000	∞	∞	∞	1.5708	90° 00′
10	0.0029	.0029	7.4637	1.0000	.0000	.0029	7.4637	343.77	2.5363	1.5679	50
20	0.0058	.0058	.7648	1.0000	.0000	.0058	.7648	171.89	.2352	1.5650	40
30	0.0087	.0087	.9408	1.0000	.0000	.0087	.9409	114.59	.0591	1.5621	30
40	0.0116	.0116	8.0658	0.9999	.0000	.0116	8.0658	85.940	1.9342	1.5592	20
50	0.0145	.0145	.1627	.9999	.0000	.0145	.1627	68.750	.8373	1.5563	10
1° 00′	0.0175	.0175	8.2419	.9998	9.9999	.0175	8.2419	57.290	1.7581	1.5533	89° 00′
10	0.0204	.0204	.3088	.9998	.9999	.0204	.3089	49.104	.6911	1.5504	50
20	0.0233	.0233	.3668	.9997	.9999	.0233	.3669	42.964	.6331	1.5475	40
30	0.0262	.0262	.4179	.9997	.9999	.0262	.4181	38.188	.5819	1.5446	30
40	0.0291	.0291	.4637	.9996	.9998	.0291	.4638	34.368	.5362	1.5417	20
50	0.0320	.0320	.5050	.9995	.9998	.0320	.5053	31.242	.4947	1.5388	10
2° 00′	0.0349	.0349	8.5428	.9994	9.9997	.0349	8.5431	28.636	1.4569	1.5359	88° 00′
10	0.0378	.0378	.5776	.9993	.9997	.0378	.5779	26.432	.4221	1.5330	50
20	0.0407	.0407	.6097	.9992	.9996	.0407	.6101	24.542	.3899	1.5301	40
30	0.0436	.0436	.6397	.9990	.9996	.0437	.6401	22.904	.3599	1.5272	30
40	0.0465	.0465	.6677	.9989	.9995	.0466	.6682	21.470	.3318	1.5243	20
50	0.0495	.0494	.6940	.9988	.9995	.0495	.6945	20.206	.3055	1.5213	10
3° 00′	0.0524	.0523	8.7188	.9986	9.9994	.0524	8.7194	19.081	1.2806	1.5184	87° 00′
10	0.0553	.0552	.7423	.9985	.9993	.0553	.7429	18.075	.2571	1.5155	50
20	0.0582	.0581	.7645	.9983	.9993	.0582	.7652	17.169	.2348	1.5126	40
30	0.0611	.0610	.7857	.9981	.9992	.0612	.7865	16.350	.2135	1.5097	30
40	0.0640	.0640	.8059	.9980	.9991	.0641	.8067	15.605	.1933	1.5068	20
50	0.0669	.0669	.8251	.9978	.9990	.0670	.8261	14.924	.1739	1.5039	10
4° 00′	0.0698	.0698	8.8436	.9976	9.9989	.0699	8.8446	14.301	1.1554	1.5010	86° 00′
10	0.0727	.0727	.8613	.9974	.9989	.0729	.8624	13.727	.1376	1.4981	50
20	0.0756	.0756	.8783	.9971	.9988	.0758	.8795	13.197	.1205	1.4952	40
30	0.0785	.0785	.8946	.9969	.9987	.0787	.8960	12.706	.1040	1.4923	30
40	0.0814	.0814	.9104	.9967	.9986	.0816	.9118	12.251	.0882	1.4893	20
50	0.0844	.0843	.9256	.9964	.9985	.0846	.9272	11.826	.0728	1.4864	10
5° 00′	0.0873	.0872	8.9403	.9962	9.9983	.0875	8.9420	11.430	1.0580	1.4835	85° 00′
10	0.0902	.0901	.9545	.9959	.9982	.0904	.9563	11.059	.0437	1.4806	50
20	0.0931	.0929	.9682	.9957	.9981	.0934	.9701	10.712	.0299	1.4777	40
30	0.0960	.0958	.9816	.9954	.9980	.0963	.9836	10.385	.0164	1.4748	30
40	0.0989	.0987	.9945	.9951	.9979	.0992	.9966	10.078	.0034	1.4719	20
50	0.1018	.1016.	9.0070	.9948	.9977	.1022	9.0093	9.7882	0.9907	1.4690	10
6° 00′	0.1047	.1045	9.0192	.9945	9.9976	.1051	9.0216	9.5144	0.9784	1.4661	84° 00′
10	0.1076	.1074	.0311	.9942	.9975	.1080	.0336	9.2553	.9664	1.4632	50
20	0.1105	.1103	.0426	.9939	.9973	.1110	.0453	9.0098	.9547	1.4603	40
30	0.1134	.1132	.0539	.9936	.9972	.1139	.0567	8.7769	.9433	1.4574	30
40	0.1164	.1161	.0648	.9932	.9971	.1169	.0678	8.5555	.9322	1.4544	20
50	0.1193	.1190	.0755	.9929	.9969	.1198	.0786	8.3450	.9214	1.4515	10
7° 00′	0.1222	.1219	9.0859	.9925	9.9968	.1228	9.0891	8.1443	0.9109	1.4486	83° 00′
10	0.1251	.1248	.0961	.9922	.9966	.1257	.0995	7.9530	.9005	1.4457	50
20	0.1280	.1276	.1060	.9918	.9964	.1287	.1096	7.7704	.8904	1.4428	40
30	0.1309	.1305	.1157	.9914	.9963	.1317	.1194	7.5958	.8806	1.4399	30
40	0.1338	.1334	.1252	.9911	.9961	.1346	.1291	7.4287	.8709	1.4370	20
50	0.1367	.1363	.1345	.9907	.9959	.1376	.1385	7.2687	.8615	1.4341	10
8° 00′	0.1396	.1392	9.1436	.9903	9.9958	.1405	9.1478	7.1154	0.8522	1.4312	82° 00′
10	0.1425	.1421	.1525	.9899	.9956	.1435	.1569	6.9682	.8431	1.4283	50
20	0.1454	.1449	.1612	.9894	.9954	.1465	.1658	6.8269	.8342	1.4254	40
30	0.1484	.1478	.1697	.9890	.9952	.1495	.1745	6.6912	.8255	1.4224	30
40	0.1513	.1507	.1781	.9886	.9950	.1524	.1831	6.5606	.8169	1.4195	20
50	0.1542	.1536	.1863	.9881	.9948	.1554	.1915	6.4348	.8085	1.4166	10
9° 00′	0.1571	.1564	9.1943	.9877	9.9946	.1584	9.1997	6.3138	0.8003	1.4137	81° 00′
		Nat.	Log.*	Nat.	Log.*	Nat.	Log.*	Nat.	Log.		
		Cosines		Sines		Cotangents		Tangents		Radians	Degrees

TRIGONOMETRIC FUNCTIONS (*continued*)

Annex—10 in columns marked *.

Degrees	Radians	Sines Nat.	Sines Log.*	Cosines Nat.	Cosines Log.*	Tangents Nat.	Tangents Log.*	Cotangents Nat.	Cotangents Log.		
9° 00′	0.1571	.1564	9.1943	.9877	9.9946	.1584	9.1997	6.3138	0.8003	1.4137	81° 00′
10	0.1600	.1593	.2022	.9872	.9944	.1614	.2078	6.1970	.7922	1.4108	50
20	0.1629	.1622	.2100	.9868	.9942	.1644	.2158	6.0844	.7842	1.4079	40
30	0.1658	.1650	.2176	.9863	.9940	.1673	.2236	5.9758	.7764	1.4050	30
40	0.1687	.1679	.2251	.9858	.9938	.1703	.2313	5.8708	.7687	1.4021	20
50	0.1716	.1708	.2324	.9853	.9936	.1733	.2389	5.7694	.7611	1.3992	10
10° 00′	0.1745	.1736	9.2397	.9848	9.9934	.1763	9.2463	5.6713	0.7537	1.3963	80° 00′
10	0.1774	.1765	.2468	.9843	.9931	.1793	.2536	5.5764	.7464	1.3934	50
20	0.1804	.1794	.2538	.9838	.9929	.1823	.2609	5.4845	.7391	1.3904	40
30	0.1833	.1822	.2606	.9833	.9927	.1853	.2680	5.3955	.7320	1.3875	30
40	0.1862	.1851	.2674	.9827	.9924	.1883	.2750	5.3093	.7250	1.3846	20
50	0.1891	.1880	.2740	.9822	.9922	.1914	.2819	5.2257	.7181	1.3817	10
11° 00′	0.1920	.1908	9.2806	.9816	9.9919	.1944	9.2887	5.1446	0.7113	1.3788	79° 00′
10	0.1949	.1937	.2870	.9811	.9917	.1974	.2953	5.0658	.7047	1.3759	50
20	0.1978	.1965	.2934	.9805	.9914	.2004	.3020	4.9894	.6980	1.3730	40
30	0.2007	.1994	.2997	.9799	.9912	.2035	.3085	4.9152	.6915	1.3701	30
40	0.2036	.2022	.3058	.9793	.9909	.2065	.3149	4.8430	.6851	1.3672	20
50	0.2065	.2051	.3119	.9787	.9907	.2095	.3212	4.7729	.6788	1.3643	10
12° 00′	0.2094	.2079	9.3179	.9781	9.9904	.2126	9.3275	4.7046	0.6725	1.3614	78° 00′
10	0.2123	.2108	.3238	.9775	.9901	.2156	.3336	4.6382	.6664	1.3584	50
20	0.2153	.2136	.3296	.9769	.9899	.2186	.3397	4.5736	.6603	1.3555	40
30	0.2182	.2164	.3353	.9763	.9896	.2217	.3458	4.5107	.6542	1.3526	30
40	0.2211	.2193	.3410	.9757	.9893	.2247	.3517	4.4494	.6483	1.3497	20
50	0.2240	.2221	.3466	.9750	.9890	.2278	.3576	4.3897	.6424	1.3468	10
13° 00′	0.2269	.2250	9.3521	.9744	9.9887	.2309	9.3634	4.3315	0.6366	1.3439	77° 00′
10	0.2298	.2278	.3575	.9737	.9884	.2339	.3691	4.2747	.6309	1.3410	50
20	0.2327	.2306	.3629	.9730	.9881	.2370	.3748	4.2193	.6252	1.3381	40
30	0.2356	.2334	.3682	.9724	.9878	.2401	.3804	4.1653	.6196	1.3352	30
40	0.2385	.2363	.3734	.9717	.9875	.2432	.3859	4.1126	.6141	1.3323	20
50	0.2414	.2391	.3786	.9710	.9872	.2462	.3914	4.0611	.6086	1.3294	10
14° 00′	0.2443	.2419	9.3837	.9703	9.9869	.2493	9.3968	4.0108	0.6032	1.3265	76° 00′
10	0.2473	.2447	.3887	.9696	.9866	.2524	.4021	3.9617	.5979	1.3235	50
20	0.2502	.2476	.3937	.9689	.9863	.2555	.4074	3.9136	.5926	1.3206	40
30	0.2531	.2504	.3986	.9681	.9859	.2586	.4127	3.8667	.5873	1.3177	30
40	0.2560	.2532	.4035	.9674	.9856	.2617	.4178	3.8208	.5822	1.3148	20
50	0.2589	.2560	.4083	.9667	.9853	.2648	.4230	3.7760	.5770	1.3119	10
15° 00′	0.2618	.2588	9.4130	.9659	9.9849	.2679	9.4281	3.7321	0.5719	1.3090	75° 00′
10	0.2647	.2616	.4177	.9652	.9846	.2711	.4331	3.6891	.5669	1.3061	50
20	0.2676	.2644	.4223	.9644	.9843	.2742	.4381	3.6470	.5619	1.3032	40
30	0.2705	.2672	.4269	.9636	.9839	.2773	.4430	3.6059	.5570	1.3003	30
40	0.2734	.2700	.4314	.9628	.9836	.2805	.4479	3.5656	.5521	1.2974	20
50	0.2763	.2728	.4359	.9621	.9832	.2836	.4527	3.5261	.5473	1.2945	10
16° 00′	0.2793	.2756	9.4403	.9613	9.9828	.2867	9.4575	3.4874	0.5425	1.2915	74° 00′
10	0.2822	.2784	.4447	.9605	.9825	.2899	.4622	3.4495	.5378	1.2886	50
20	0.2851	.2812	.4491	.9596	.9821	.2931	.4669	3.4124	.5331	1.2857	40
30	0.2880	.2840	.4533	.9588	.9817	.2962	.4716	3.3759	.5284	1.2828	30
40	0.2909	.2868	.4576	.9580	.9814	.2994	.4762	3.3402	.5238	1.2799	20
50	0.2938	.2896	.4618	.9572	.9810	.3026	.4808	3.3052	.5192	1.2770	10
17° 00′	0.2967	.2924	9.4659	.9563	9.9806	.3057	9.4853	3.2709	0.5147	1.2741	73° 00′
10	0.2996	.2952	.4700	.9555	.9802	.3089	.4898	3.2371	.5102	1.2712	50
20	0.3025	.2979	.4741	.9546	.9798	.3121	.4943	3.2041	.5057	1.2683	40
30	0.3054	.3007	.4781	.9537	.9794	.3153	.4987	3.1716	.5013	1.2654	30
40	0.3083	.3035	.4821	.9528	.9790	.3185	.5031	3.1397	.4969	1.2625	20
50	0.3113	.3062	.4861	.9520	.9786	.3217	.5075	3.1084	.4925	1.2595	10
18° 00′	0.3142	.3090	9.4900	.9511	9.9782	.3249	9.5118	3.0777	0.4882	1.2566	72° 00′
		Nat.	Log.*	Nat.	Log.*	Nat.	Log.*	Nat.	Log.		
		Cosines		Sines		Cotangents		Tangents		Radians	Degrees

TRIGONOMETRIC FUNCTIONS (*continued*)

Annex—10 in columns marked *.

Degrees	Radians	Sines Nat.	Sines Log.*	Cosines Nat.	Cosines Log.*	Tangents Nat.	Tangents Log.*	Cotangents Nat.	Cotangents Log.	Radians	Degrees
18° 00′	0.3142	.3090	9.4900	.9511	9.9782	.3249	9.5118	3.0777	0.4882	1.2566	72° 00′
10	0.3171	.3118	.4939	.9502	.9778	.3281	.5161	3.0475	.4839	1.2537	50
20	0.3200	.3145	.4977	.9492	.9774	.3314	.5203	3.0178	.4797	1.2508	40
30	0.3229	.3173	.5015	.9483	.9770	.3346	.5245	2.9887	.4755	1.2479	30
40	0.3258	.3201	.5052	.9474	.9765	.3378	.5287	2.9600	.4713	1.2450	20
50	0.3287	.3228	.5090	.9465	.9761	.3411	.5329	2.9319	.4671	1.2421	10
19° 00′	0.3316	.3256	9.5126	.9455	9.9757	.3443	9.5370	2.9042	0.4630	1.2392	71° 00′
10	0.3345	.3283	.5163	.9446	.9752	.3476	.5411	2.8770	.4589	1.2363	50
20	0.3374	.3311	.5199	.9436	.9748	.3508	.5451	2.8502	.4549	1.2334	40
30	0.3403	.3338	.5235	.9426	.9743	.3541	.5491	2.8239	.4509	1.2305	30
40	0.3432	.3365	.5270	.9417	.9739	.3574	.5531	2.7980	.4469	1.2275	20
50	0.3462	.3393	.5306	.9407	.9734	.3607	.5571	2.7725	.4429	1.2246	10
20° 00′	0.3491	.3420	9.5341	.9397	9.9730	.3640	9.5611	2.7475	0.4389	1.2217	70° 00′
10	0.3520	.3448	.5375	.9387	.9725	.3673	.5650	2.7228	.4350	1.2188	50
20	0.3549	.3475	.5409	.9377	.9721	.3706	.5689	2.6985	.4311	1.2159	40
30	0.3578	.3502	.5443	.9367	.9716	.3739	.5727	2.6746	.4273	1.2130	30
40	0.3607	.3529	.5477	.9356	.9711	.3772	.5766	2.6511	.4234	1.2101	20
50	0.3636	.3557	.5510	.9346	.9706	.3805	.5804	2.6279	.4196	1.2072	10
21° 00′	0.3665	.3584	9.5543	.9336	9.9702	.3839	9.5842	2.6051	0.4153	1.2043	69° 00′
10	0.3694	.3611	.5576	.9325	.9697	.3872	.5879	2.5826	.4121	1.2014	50
20	0.3723	.3638	.5609	.9315	.9692	.3906	.5917	2.5605	.4083	1.1985	40
30	0.3752	.3665	.5641	.9304	.9687	.3939	.5954	2.5386	.4046	1.1956	30
40	0.3782	.3692	.5673	.9293	.9682	.3973	.5991	2.5172	.4009	1.1926	20
50	0.3811	.3719	.5704	.9283	.9677	.4006	.6028	2.4960	.3972	1.1897	10
22° 00′	0.3840	.3746	9.5736	.9272	9.9672	.4040	9.6064	2.4751	0.3936	1.1868	68° 00′
10	0.3869	.3773	.5767	.9261	.9667	.4074	.6100	2.4545	.3900	1.1839	50
20	0.3898	.3800	.5798	.9250	.9661	.4108	.6136	2.4342	.3864	1.1810	40
30	0.3927	.3827	.5828	.9239	.9656	.4142	.6172	2.4142	.3828	1.1781	30
40	0.3956	.3854	.5859	.9228	.9651	.4176	.6208	2.3945	.3792	1.1752	20
50	0.3985	.3881	.5889	.9216	.9646	.4210	.6243	2.3750	.3757	1.1723	10
23° 00′	0.4014	.3907	9.5919	.9205	9.9640	.4245	9.6279	2.3559	0.3721	1.1694	67° 00′
10	0.4043	.3934	.5948	.9194	.9635	.4279	.6314	2.3369	.3686	1.1665	50
20	0.4072	.3961	.5978	.9182	.9629	.4314	.6348	2.3183	.3652	1.1636	40
30	0.4102	.3987	.6007	.9171	.9624	.4348	.6383	2.2998	.3617	1.1606	30
40	0.4131	.4014	.6036	.9159	.9618	.4383	.6417	2.2817	.3583	1.1577	20
50	0.4160	.4041	.6065	.9147	.9613	.4417	.6452	2.2637	.3548	1.1548	10
24° 00′	0.4189	.4067	9.6093	.9135	9.9607	.4452	9.6486	2.2460	0.3514	1.1519	66° 00′
10	0.4218	.4094	.6121	.9124	.9602	.4487	.6520	2.2286	.3480	1.1490	50
20	0.4247	.4120	.6149	.9112	.9596	.4522	.6553	2.2113	.3447	1.1461	40
30	0.4276	.4147	.6177	.9100	.9590	.4557	.6587	2.1943	.3413	1.1432	30
40	0.4305	.4173	.6205	.9088	.9584	.4592	.6620	2.1775	.3380	1.1403	20
50	0.4334	.4200	.6232	.9075	.9579	.4628	.6654	2.1609	.3346	1.1374	10
25° 00′	0.4363	.4226	9.6259	.9063	9.9573	.4663	9.6687	2.1445	0.3313	1.1345	65° 00′
10	0.4392	.4253	.6286	.9051	.9567	.4699	.6720	2.1283	.3280	1.1316	50
20	0.4422	.4279	.6313	.9038	.9561	.4734	.6752	2.1123	.3248	1.1286	40
30	0.4451	.4305	.6340	.9026	.9555	.4770	.6785	2.0965	.3215	1.1257	30
40	0.4480	.4331	.6366	.9013	.9549	.4806	.6817	2.0809	.3183	1.1228	20
50	0.4509	.4358	.6392	.9001	.9543	.4841	.6850	2.0655	.3150	1.1199	10
26° 00′	0.4538	.4384	9.6418	.8988	9.9537	.4877	9.6882	2.0503	0.3118	1.1170	64° 00′
10	0.4567	.4410	.6444	.8975	.9530	.4913	.6914	2.0353	.3086	1.1141	50
20	0.4596	.4436	.6470	.8962	.9524	.4950	.6946	2.0204	.3054	1.1112	40
30	0.4625	.4462	.6495	.8949	.9518	.4986	.6977	2.0057	.3023	1.1083	30
40	0.4654	.4488	.6521	.8936	.9512	.5022	.7009	1.9912	.2991	1.1054	20
50	0.4683	.4514	.6546	.8923	.9505	.5059	.7040	1.9768	.2960	1.1025	10
27° 00′	0.4712	.4540	9.6570	.8910	9.9499	.5095	9.7072	1.9626	0.2928	1.0996	63° 00′
		Nat.	Log.*	Nat.	Log.*	Nat.	Log.*	Nat.	Log.		
		Cosines		Sines		Cotangents		Tangents		Radians	Degrees

TRIGONOMETRIC FUNCTIONS *(continued)*

Annex – 10 in columns marked.*

Degrees	Radians	Sines Nat.	Sines Log.*	Cosines Nat.	Cosines Log.*	Tangents Nat.	Tangents Log.*	Cotangents Nat.	Cotangents Log.	Radians	Degrees
27° 00′	0.4712	.4540	9.6570	.8910	9.9499	.5095	9.7072	1.9626	0.2928	1.0996	63° 00′
10	0.4741	.4566	.6595	.8897	.9492	.5132	.7103	1.9486	.2897	1.0966	50
20	0.4771	.4592	.6620	.8884	.9486	.5169	.7134	1.9347	.2866	1.0937	40
30	0.4800	.4617	.6644	.8870	.9479	.5206	.7165	1.9210	.2835	1.0908	30
40	0.4829	.4643	.6668	.8857	.9473	.5243	.7196	1.9074	.2804	1.0879	20
50	0.4858	.4669	.6692	.8843	.9466	.5280	.7226	1.8940	.2774	1.0850	10
28° 00′	0.4887	.4695	9.6716	.8829	9.9459	.5317	9.7257	1.8807	0.2743	1.0821	62° 00′
10	0.4916	.4720	.6740	.8816	.9453	.5354	.7287	1.8676	.2713	1.0792	50
20	0.4945	.4746	.6763	.8802	.9446	.5392	.7317	1.8546	.2683	1.0763	40
30	0.4974	.4772	.6787	.8788	.9439	.5430	.7348	1.8418	.2652	1.0734	30
40	0.5003	.4797	.6810	.8774	.9432	.5467	.7378	1.8291	.2622	1.0705	20
50	0.5032	.4823	.6833	.8760	.9425	.5505	.7408	1.8165	.2592	1.0676	10
29° 00′	0.5061	.4848	9.6856	.8746	9.9418	.5543	9.7438	1.8040	0.2562	1.0647	61° 00′
10	0.5091	.4874	.6878	.8732	.9411	.5581	.7467	1.7917	.2533	1.0617	50
20	0.5120	.4899	.6901	.8718	.9404	.5619	.7497	1.7796	.2503	1.0588	40
30	0.5149	.4924	.6923	.8704	.9397	.5658	.7526	1.7675	.2474	1.0559	30
40	0.5178	.4950	.6946	.8689	.9390	.5696	.7556	1.7556	.2444	1.0530	20
50	0.5207	.4975	.6968	.8675	.9383	.5735	.7585	1.7437	.2415	1.0501	10
30° 00′	0.5236	.5000	9.6990	.8660	9.9375	.5774	9.7614	1.7321	0.2386	1.0472	60° 00′
10	0.5265	.5025	.7012	.8646	.9368	.5812	.7644	1.7205	.2356	1.0443	50
20	0.5294	.5050	.7033	.8631	.9361	.5851	.7673	1.7090	.2327	1.0414	40
30	0.5323	.5075	.7055	.8616	.9353	.5890	.7701	1.6977	.2299	1.0385	30
40	0.5352	.5100	.7076	.8601	.9346	.5930	.7730	1.6864	.2270	1.0356	20
50	0.5381	.5125	.7097	.8587	.9338	.5969	.7759	1.6753	.2241	1.0327	10
31° 00′	0.5411	.5150	9.7118	.8572	9.9331	.6009	9.7788	1.6643	0.2212	1.0297	59° 00′
10	0.5440	.5175	.7139	.8557	.9323	.6048	.7816	1.6534	.2184	1.0268	50
20	0.5469	.5200	.7160	.8542	.9315	.6088	.7845	1.6426	.2155	1.0239	40
30	0.5498	.5225	.7181	.8526	.9308	.6128	.7873	1.6319	.2127	1.0210	30
40	0.5527	.5250	.7201	.8511	.9300	.6168	.7902	1.6212	.2098	1.0181	20
50	0.5556	.5275	.7222	.8496	.9292	.6208	.7930	1.6107	.2070	1.0152	10
32° 00′	0.5585	.5299	9.7242	.8480	9.9284	.6249	9.7958	1.6003	0.2042	1.0123	58° 00′
10	0.5614	.5324	.7262	.8465	.9276	.6289	.7986	1.5900	.2014	1.0094	50
20	0.5643	.5348	.7282	.8450	.9268	.6330	.8014	1.5798	.1986	1.0065	40
30	0.5672	.5373	.7302	.8434	.9260	.6371	.8042	1.5697	.1958	1.0036	30
40	0.5701	.5398	.7322	.8418	.9252	.6412	.8070	1.5597	.1930	1.0007	20
50	0.5730	.5422	.7342	.8403	.9244	.6453	.8097	1.5497	.1903	0.9977	10
33° 00′	0.5760	.5446	9.7361	.8387	9.9236	.6494	9.8125	1.5399	0.1875	0.9948	57° 00′
10	0.5789	.5471	.7380	.8371	.9228	.6536	.8153	1.5301	.1847	0.9919	50
20	0.5818	.5495	.7400	.8355	.9219	.6577	.8180	1.5204	.1820	0.9890	40
30	0.5847	.5519	.7419	.8339	.9211	.6619	.8208	1.5108	.1792	0.9861	30
40	0.5876	.5544	.7438	.8323	.9203	.6661	.8235	1.5013	.1765	0.9832	20
50	0.5905	.5568	.7457	.8307	.9194	.6703	.8263	1.4919	.1737	0.9803	10
34° 00′	0.5934	.5592	9.7476	.8290	9.9186	.6745	9.8290	1.4826	0.1710	0.9774	56° 00′
10	0.5963	.5616	.7494	.8274	.9177	.6787	.8317	1.4733	.1683	0.9745	50
20	0.5992	.5640	.7513	.8258	.9169	.6830	.8344	1.4641	.1656	0.9716	40
30	0.6021	.5664	.7531	.8241	.9160	.6873	.8371	1.4550	.1629	0.9687	30
40	0.6050	.5688	.7550	.8225	.9151	.6916	.8398	1.4460	.1602	0.9657	20
50	0.6080	.5712	.7568	.8208	.9142	.6959	.8425	1.4370	.1575	0.9628	10
35° 00′	0.6109	.5736	9.7586	.8192	9.9134	.7002	9.8452	1.4281	0.1548	0.9599	55° 00′
10	0.6138	.5760	.7604	.8175	.9125	.7046	.8479	1.4193	.1521	0.9570	50
20	0.6167	.5783	.7622	.8158	.9116	.7089	.8506	1.4106	.1494	0.9541	40
30	0.6196	.5807	.7640	.8141	.9107	.7133	.8533	1.4019	.1467	0.9512	30
40	0.6225	.5831	.7657	.8124	.9098	.7177	.8559	1.3934	.1441	0.9483	20
50	0.6254	.5854	.7675	.8107	.9089	.7221	.8586	1.3848	.1414	0.9454	10
36° 00′	0.6283	.5878	9.7692	.8090	9.9080	.7265	9.8613	1.3764	0.1387	0.9425	54° 00′
		Nat.	Log.*	Nat.	Log.*	Nat.	Log.*	Nat.	Log.		
		Cosines		Sines		Cotangents		Tangents		Radians	Degrees

TRIGONOMETRIC FUNCTIONS (*continued*)

Annex—10 in columns marked *.

Degrees	Radians	Sines Nat.	Sines Log.*	Cosines Nat.	Cosines Log.*	Tangents Nat.	Tangents Log.*	Cotangents Nat.	Cotangents Log.	Radians	Degrees
36° 00′	0.6283	.5878	9.7692	.8090	9.9080	.7265	9.8613	1.3764	0.1387	0.9425	54° 00′
10	0.6312	.5901	.7710	.8073	.9070	.7310	.8639	1.3680	.1361	0.9396	50
20	0.6341	.5925	.7727	.8056	.9061	.7355	.8666	1.3597	.1334	0.9367	40
30	0.6370	.5948	.7744	.8039	.9052	.7400	.8692	1.3514	.1308	0.9338	30
40	0.6400	.5972	.7761	.8021	.9042	.7445	.8718	1.3432	.1282	0.9308	20
50	0.6429	.5995	.7778	.8004	.9033	.7490	.8745	1.3351	.1255	0.9279	10
37° 00′	0.6458	.6018	9.7795	.7986	9.9023	.7536	9.8771	1.3270	0.1229	0.9250	53° 00′
10	0.6487	.6041	.7811	.7969	.9014	.7581	.8797	1.3190	.1203	0.9221	50
20	0.6516	.6065	.7828	.7951	.9004	.7627	.8824	1.3111	.1176	0.9192	40
30	0.6545	.6088	.7844	.7934	.8995	.7673	.8850	1.3032	.1150	0.9163	30
40	0.6574	.6111	.7861	.7916	.8985	.7720	.8876	1.2954	.1124	0.9134	20
50	0.6603	.6134	.7877	.7898	.8975	.7766	.8902	1.2876	.1098	0.9105	10
38° 00′	0.6632	.6157	9.7893	.7880	9.8965	.7813	9.8928	1.2799	0.1072	0.9076	52° 00′
10	0.6661	.6180	.7910	.7862	.8955	.7860	.8954	1.2723	.1046	0.9047	50
20	0.6690	.6202	.7926	.7844	.8945	.7907	.8980	1.2647	.1020	0.9018	40
30	0.6720	.6225	.7941	.7826	.8935	.7954	.9006	1.2572	.0994	0.8988	30
40	0.6749	.6248	.7957	.7808	.8925	.8002	.9032	1.2497	.0968	0.8959	20
50	0.6778	.6271	.7973	.7790	.8915	.8050	.9058	1.2423	.0942	0.8930	10
39° 00′	0.6807	.6293	9.7989	.7771	9.8905	.8098	9.9084	1.2349	0.0916	0.8901	51° 00′
10	0.6836	.6316	.8004	.7753	.8895	.8146	.9110	1.2276	.0890	0.8872	50
20	0.6865	.6338	.8020	.7735	.8884	.8195	.9135	1.2203	.0865	0.8843	40
30	0.6894	.6361	.8035	.7716	.8874	.8243	.9161	1.2131	.0839	0.8814	30
40	0.6923	.6383	.8050	.7698	.8864	.8292	.9187	1.2059	.0813	0.8785	20
50	0.6952	.6406	.8066	.7679	.8853	.8342	.9212	1.1988	.0788	0.8756	10
40° 00′	0.6981	.6428	9.8081	.7660	9.8843	.8391	9.9238	1.1918	0.0762	0.8727	50° 00′
10	0.7010	.6450	.8096	.7642	.8832	.8441	.9264	1.1847	.0736	0.8698	50
20	0.7039	.6472	.8111	.7623	.8821	.8491	.9289	1.1778	.0711	0.8668	40
30	0.7069	.6494	.8125	.7604	.8810	.8541	.9315	1.1708	.0685	0.8639	30
40	0.7098	.6517	.8140	.7585	.8800	.8591	.9341	1.1640	.0659	0.8610	20
50	0.7127	.6539	.8155	.7566	.8789	.8642	.9366	1.1571	.0634	0.8581	10
41° 00′	0.7156	.6561	9.8169	.7547	9.8778	.8693	9.9392	1.1504	0.0608	0.8552	49° 00′
10	0.7185	.6583	.8184	.7528	.8767	.8744	.9417	1.1436	.0583	0.8523	50
20	0.7214	.6604	.8198	.7509	.8756	.8796	.9443	1.1369	.0557	0.8494	40
30	0.7243	.6626	.8213	.7490	.8745	.8847	.9468	1.1303	.0532	0.8465	30
40	0.7272	.6648	.8227	.7470	.8733	.8899	.9494	1.1237	.0506	0.8436	20
50	0.7301	.6670	.8241	.7451	.8722	.8952	.9519	1.1171	.0481	0.8407	10
42° 00′	0.7330	.6691	9.8255	.7431	9.8711	.9004	9.9544	1.1106	0.0456	0.8378	48° 00′
10	0.7359	.6713	.8269	.7412	.8699	.9057	.9570	1.1041	.0430	0.8348	50
20	0.7389	.6734	.8283	.7392	.8688	.9110	.9595	1.0977	.0405	0.8319	40
30	0.7418	.6756	.8297	.7373	.8676	.9163	.9621	1.0913	.0379	0.8290	30
40	0.7447	.6777	.8311	.7353	.8665	.9217	.9646	1.0850	.0354	0.8261	20
50	0.7476	.6799	.8324	.7333	.8653	.9271	.9671	1.0786	.0329	0.8232	10
43° 00′	0.7505	.6820	9.8338	.7314	9.8641	.9325	9.9697	1.0724	0.0303	0.8203	47° 00′
10	0.7534	.6841	.8351	.7294	.8629	.9380	.9722	1.0661	.0278	0.8174	50
20	0.7563	.6862	.8365	.7274	.8618	.9435	.9747	1.0599	.0253	0.8145	40
30	0.7592	.6884	.8378	.7254	.8606	.9490	.9772	1.0538	.0228	0.8116	30
40	0.7621	.6905	.8391	.7234	.8594	.9545	.9798	1.0477	.0202	0.8087	20
50	0.7650	.6926	.8405	.7214	.8582	.9601	.9823	1.0416	.0177	0.8058	10
44° 00′	0.7679	.6947	9.8418	.7193	9.8569	.9657	9.9848	1.0355	0.0152	0.8029	46° 00′
10	0.7709	.6967	.8431	.7173	.8557	.9713	.9874	1.0295	.0126	0.7999	50
20	0.7738	.6988	.8444	.7153	.8545	.9770	.9899	1.0235	.0101	0.7970	40
30	0.7767	.7009	.8457	.7133	.8532	.9827	.9924	1.0176	.0076	0.7941	30
40	0.7796	.7030	.8469	.7112	.8520	.9884	.9949	1.0117	.0051	0.7912	20
50	0.7825	.7050	.8482	.7092	.8507	.9942	.9975	1.0058	.0025	0.7883	10
45° 00′	0.7854	.7071	9.8495	.7071	9.8495	1.0000	0.0000	1.0000	0.0000	0.7854	45° 00′
		Nat.	Log.*	Nat.	Log.*	Nat.	Log.*	Nat.	Log.		
		Cosines		Sines		Cotangents		Tangents		Radians	Degrees

DECIMAL EQUIVALENTS

From minutes and seconds into decimal parts of a degree

0′	0°.0000	0″	0°.0000
1	.0167	1	.0003
2	.0333	2	.0006
3	.05	3	.0008
4	.0667	4	.0011
5′	.0833	5″	.0014
6	.10	6	.0017
7	.1167	7	.0019
8	.1333	8	.0022
9	.15	9	.0025
10′	0°.1667	10″	0°.0028
1	.1833	1	.0031
2	.20	2	.0033
3	.2167	3	.0036
4	.2333	4	.0039
15′	.25	15″	.0042
6	.2667	6	.0044
7	.2833	7	.0047
8	.30	8	.005
9	.3167	9	.0053
20′	0°.3333	20″	0°.0056
1	.35	1	.0058
2	.3667	2	.0061
3	.3833	3	.0064
4	.40	4	.0067
25′	.4167	25″	.0069
6	.4333	6	.0072
7	.45	7	.0075
8	.4667	8	.0078
9	.4833	9	.0081
30′	0°.50	30″	0°.0083
1	.5167	1	.0086
2	.5333	2	.0089
3	.55	3	.0092
4	.5667	4	.0094
35′	.5833	35″	.0097
6	.60	6	.01
7	.6167	7	.0103
8	.6333	8	.0106
9	.65	9	.0108
40′	0°.6667	40″	0°.0111
1	.6833	1	.0114
2	.70	2	.0117
3	.7167	3	.0119
4	.7333	4	.0122
45′	.75	45″	.0125
6	.7667	6	.0128
7	.7833	7	.0131
8	.80	8	.0133
9	.8167	9	.0136
50′	0°.8333	50″	0°.0139
1	.85	1	.0142
2	.8667	2	.0144
3	.8833	3	.0147
4	.90	4	.015
55′	.9167	55″	.0153
6	.9333	6	.0156
7	.95	7	.0158
8	.9667	8	.0161
9	.9833	9	.0164
60′	1.00	60″	0°.0167

From decimal parts of a degree into minutes and seconds (exact values)

0°.00	0′	0°.50	30′
1	0′ 36″	1	30′ 36″
2	1′ 12″	2	31′ 12″
3	1′ 48″	3	31′ 48″
4	2′ 24″	4	32′ 24″
0°.05	3′	0°.55	33′
6	3′ 36″	6	33′ 36″
7	4′ 12″	7	34′ 12″
8	4′ 48″	8	34′ 48″
9	5′ 24″	9	35′ 24″
0°.10	6′	0°.60	36′
1	6′ 36″	1	36′ 36″
2	7′ 12″	2	37′ 12″
3	7′ 48″	3	37′ 48″
4	8′ 24″	4	38′ 24″
0°.15	9′	0°.65	39′
6	9′ 36″	6	39′ 36″
7	10′ 12″	7	40′ 12″
8	10′ 48″	8	40′ 48″
9	11′ 24″	9	41′ 24″
0°.20	12′	0°.70	42′
1	12′ 36″	1	42′ 36″
2	13′ 12″	2	43′ 12″
3	13′ 48″	3	43′ 48″
4	14′ 24″	4	44′ 24″
0°.25	15′	0°.75	45′
6	15′ 36″	6	45′ 36″
7	16′ 12″	7	46′ 12″
8	16′ 48″	8	46′ 48″
9	17′ 24″	9	47′ 24″
0°.30	18′	0°.80	48′
1	18′ 36″	1	48′ 36″
2	19′ 12″	2	49′ 12″
3	19′ 48″	3	49′ 48″
4	20′ 24″	4	50′ 24″
0°.35	21′	0°.85	51′
6	21′ 36″	6	51′ 36″
7	22′ 12″	7	52′ 12″
8	22′ 48″	8	52′ 48″
9	23′ 24″	9	53′ 24″
0°.40	24′	0°.90	54′
1	24′ 36″	1	54′ 36″
2	25′ 12″	2	55′ 12″
3	25′ 48″	3	55′ 48″
4	26′ 24″	4	56′ 24″
0°.45	27′	0°.95	57′
6	27′ 36″	6	57′ 36″
7	28′ 12″	7	58′ 12″
8	28′ 48″	8	58′ 48″
9	29′ 24″	9	59′ 24″
0°.50	30′	1°.00	60′

0°.000	0″.0
1	3″.6
2	7″.2
3	10″.8
4	14″.4
0°.005	18″
6	21″.6
7	25″.2
8	28″.8
9	32″.4
0°.010	36″

Common fractions

8 ths	16 ths	32 nds	64 ths	Exact decimal values
			1	.01 5625
		1	2	.03 125
			3	.04 6875
	1	2	4	.06 25
			5	.07 8125
		3	6	.09 375
			7	.10 9375
1	2	4	8	.12 5
			9	.14 0625
		5	10	.15 625
			11	.17 1875
	3	6	12	.18 75
			13	.20 3125
		7	14	.21 875
			15	.23 4375
2	4	8	16	.25
			17	.26 5625
		9	18	.28 125
			19	.29 6875
	5	10	20	.31 25
			21	.32 8125
		11	22	.34 375
			23	.35 9375
3	6	12	24	.37 5
			25	.39 0625
		13	26	.40 625
			27	.42 1875
	7	14	28	.43 75
			29	.45 3125
		15	30	.46 875
			31	.48 4375
4	8	16	32	.50
			33	.51 5625
		17	34	.53 125
			35	.54 6875
	9	18	36	.56 25
			37	.57 8125
		19	38	.59 375
			39	.60 9375
5	10	20	40	.62 5
			41	.64 0625
		21	42	.65 625
			43	.67 1875
	11	22	44	.68 75
			45	.70 3125
		23	46	.71 875
			47	.73 4375
6	12	24	48	.75
			49	.76 5625
		25	50	.78 125
			51	.79 6875
	13	26	52	.81 25
			53	.82 8125
		27	54	.84 375
			55	.85 9375
7	14	28	56	.87 5
			57	.89 0625
		29	58	.90 625
			59	.92 1875
	15	30	60	.93 75
			61	.95 3125
		31	62	.96 875
			63	.98 4375

CIRCUMFERENCES AND AREAS OF CIRCLES BY EIGHTHS, ETC.

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
			⅞	2.749	.6013	4	12.57	12.57	9	28.27	63.62
1/64	.04909	.00019	57/64	2.798	.6230	1/16	12.76	12.96	1/8	28.67	65.40
1/32	.09817	.00077	29/32	2.847	.6450	1/8	12.96	13.36	1/4	29.06	67.20
3/64	.1473	.00173	59/64	2.896	.6675	3/16	13.16	13.77	3/8	29.45	69.03
1/16	.1963	.00307	15/16	2.945	.6903	1/4	13.35	14.19	1/2	29.85	70.88
5/64	.2454	.00479	61/64	2.994	.7135	5/16	13.55	14.61	5/8	30.24	72.76
3/32	.2945	.00690	31/32	3.043	.7371	3/8	13.74	15.03	3/4	30.63	74.66
7/64	.3436	.00940	63/64	3.093	.7610	7/16	13.94	15.47	7/8	31.02	76.59
1/8	.3927	.01227	1	3.142	.7854	1/2	14.14	15.90	10	31.42	78.54
9/64	.4418	.01553	1/16	3.338	.8866	9/16	14.33	16.35	1/8	31.81	80.52
5/32	.4909	.01917	1/8	3.534	.9940	5/8	14.53	16.80	1/4	32.20	82.52
11/64	.5400	.02320	3/16	3.731	1.108	11/16	14.73	17.26	3/8	32.59	84.54
3/16	.5890	.02761	1/4	3.927	1.227	3/4	14.92	17.72	1/2	32.99	86.59
13/64	.6381	.03241	5/16	4.123	1.353	13/16	15.12	18.19	5/8	33.38	88.66
7/32	.6872	.03758	3/8	4.320	1.485	7/8	15.32	18.67	3/4	33.77	90.76
15/64	.7363	.04314	7/16	4.516	1.623	15/16	15.51	19.15	7/8	34.16	92.89
1/4	.7854	.04909	1/2	4.712	1.767	5	15.71	19.63	11	34.56	95.03
17/64	.8345	.05542	9/16	4.909	1.917	1/16	15.90	20.13	1/8	34.95	97.21
9/32	.8836	.06213	5/8	5.105	2.074	1/8	16.10	20.63	1/4	35.34	99.40
19/64	.9327	.06922	11/16	5.301	2.237	3/16	16.30	21.14	3/8	35.74	101.6
5/16	.9817	.07670	3/4	5.498	2.405	1/4	16.49	21.65	1/2	36.13	103.9
21/64	1.031	.08456	13/16	5.694	2.580	5/16	16.69	22.17	5/8	36.52	106.1
11/32	1.080	.09281	7/8	5.890	2.761	3/8	16.89	22.69	3/4	36.91	108.4
23/64	1.129	.1014	15/16	6.087	2.948	7/16	17.08	23.22	7/8	37.31	110.8
3/8	1.178	.1104	2	6.283	3.142	1/2	17.28	23.76	12	37.70	113.1
25/64	1.227	.1198	1/16	6.480	3.341	9/16	17.48	24.30	1/8	38.09	115.5
13/32	1.276	.1296	1/8	6.676	3.547	5/8	17.67	24.85	1/4	38.48	117.9
27/64	1.325	.1398	3/16	6.872	3.758	11/16	17.87	25.41	3/8	38.88	120.3
7/16	1.374	.1503	1/4	7.069	3.976	3/4	18.06	25.97	1/2	39.27	122.7
29/64	1.424	.1613	5/16	7.265	4.200	13/16	18.26	26.53	5/8	39.66	125.2
15/32	1.473	.1726	3/8	7.461	4.430	7/8	18.46	27.11	3/4	40.06	127.7
31/64	1.522	.1843	7/16	7.658	4.666	15/16	18.65	27.69	7/8	40.45	130.2
1/2	1.571	.1963	1/2	7.854	4.909	6	18.85	28.27	13	40.84	132.7
33/64	1.620	.2088	9/16	8.050	5.157	1/8	19.24	29.46	1/8	41.23	135.3
17/32	1.669	.2217	5/8	8.247	5.412	1/4	19.63	30.68	1/4	41.63	137.9
35/64	1.718	.2349	11/16	8.443	5.673	3/8	20.03	31.92	3/8	42.02	140.5
9/16	1.767	.2485	3/4	8.639	5.940	1/2	20.42	33.18	1/2	42.41	143.1
37/64	1.816	.2625	13/16	8.836	6.213	5/8	20.81	34.47	5/8	42.80	145.8
19/32	1.865	.2769	7/8	9.032	6.492	3/4	21.21	35.78	3/4	43.20	148.5
39/64	1.914	.2916	15/16	9.228	6.777	7/8	21.60	37.12	7/8	43.59	151.2
5/8	1.963	.3068	3	9.425	7.069	7	21.99	38.48	14	43.98	153.9
41/64	2.013	.3223	1/16	9.621	7.366	1/8	22.38	39.87	1/8	44.37	156.7
21/32	2.062	.3382	1/8	9.817	7.670	1/4	22.78	41.28	1/4	44.77	159.5
43/64	2.111	.3545	3/16	10.01	7.980	3/8	23.17	42.72	3/8	45.16	162.3
11/16	2.160	.3712	1/4	10.21	8.296	1/2	23.56	44.18	1/2	45.55	165.1
45/64	2.209	.3883	5/16	10.41	8.618	5/8	23.95	45.66	5/8	45.95	168.0
23/32	2.258	.4057	3/8	10.60	8.946	3/4	24.35	47.17	3/4	46.34	170.9
47/64	2.307	.4236	7/16	10.80	9.281	7/8	24.74	48.71	7/8	46.73	173.8
3/4	2.356	.4418	1/2	11.00	9.621	8	25.13	50.27	15	47.12	176.7
49/64	2.405	.4604	9/16	11.19	9.968	1/8	25.53	51.85	1/8	47.52	179.7
25/32	2.454	.4794	5/8	11.39	10.32	1/4	25.92	53.46	1/4	47.91	182.7
51/64	2.503	.4987	11/16	11.58	10.68	3/8	26.31	55.09	3/8	48.30	185.7
13/16	2.553	.5185	3/4	11.78	11.04	1/2	26.70	56.75	1/2	48.69	188.7
53/64	2.602	.5386	13/16	11.98	11.42	5/8	27.10	58.43	5/8	49.09	191.7
27/32	2.651	.5591	7/8	12.17	11.79	3/4	27.49	60.13	3/4	49.48	194.8
55/64	2.700	.5800	15/16	12.37	12.18	7/8	27.88	61.86	7/8	49.87	197.9

CIRCUMFERENCES AND AREAS BY EIGHTHS—(*continued*)

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
16	50.27	201.1	19 ⅛	61.26	298.6	22	72.26	415.5	25	91.11	660.5
⅛	50.66	204.2	¼	61.65	302.5	⅛	72.65	420.0	⅛	91.89	672.0
¼	51.05	207.4	⅜	62.05	306.4	¼	73.04	424.6	½	92.68	683.5
⅜	51.44	210.6	⅝	62.44	310.2	⅜	73.43	429.1	⅞	93.46	695.1
½	51.84	213.8	20	62.83	314.2	½	73.83	433.7	26	94.25	706.9
⅝	52.23	217.1	⅛	63.22	318.1	⅝	74.22	438.4	⅛	95.03	718.7
¾	52.62	220.4	¼	63.62	322.1	¾	74.61	443.0	½	95.82	730.6
⅞	53.01	223.7	⅜	64.01	326.1	⅞	75.01	447.7	⅞	96.60	742.6
17	53.41	227.0	½	64.40	330.1	24	75.40	452.4	27	97.39	754.8
⅛	53.80	230.3	⅝	64.80	334.1	⅛	76.18	461.9	⅛	98.17	767.0
¼	54.19	233.7	¾	65.19	338.2	½	76.97	471.4	½	98.96	779.3
⅜	54.59	237.1	⅞	65.58	342.2	¾	77.75	481.1	⅞	99.75	791.7
½	54.98	240.5	21	65.97	346.4	25	78.54	490.9	28	100.5	804.2
⅝	55.37	244.0	⅛	66.37	350.5	⅛	79.33	500.7	⅛	101.3	816.9
¾	55.76	247.4	¼	66.76	354.7	½	80.11	510.7	½	102.1	829.6
⅞	56.16	250.9	⅜	67.15	358.8	¾	80.90	520.8	⅞	102.9	842.4
18	56.55	254.5	½	67.54	363.1	26	81.68	530.9	29	103.7	855.3
⅛	56.94	258.0	⅝	67.94	367.3	¼	82.47	541.2	⅛	104.5	868.3
¼	57.33	261.6	¾	68.33	371.5	½	83.25	551.5	½	105.2	881.4
⅜	57.73	265.2	⅞	68.72	375.8	¾	84.04	562.0	⅞	106.0	894.6
½	58.12	268.8	22	69.12	380.1	27	84.82	572.6	30	106.8	907.9
⅝	58.51	272.4	⅛	69.51	384.5	⅛	85.61	583.2	⅛	107.6	921.3
¾	58.90	276.1	¼	69.90	388.8	½	86.39	594.0	½	108.4	934.8
⅞	59.30	279.8	⅜	70.29	393.2	¾	87.18	604.8	⅞	109.2	948.4
19	59.69	283.5	½	70.69	397.6	28	87.96	615.8	35	110.0	962.1
⅛	60.08	287.3	⅝	71.08	402.0	⅛	88.75	626.8	⅛	110.7	975.9
¼	60.48	291.0	¾	71.47	406.5	½	89.54	637.9	½	111.5	989.8
⅜	60.87	294.8	⅞	71.86	411.0	¾	90.32	649.2	⅞	112.3	1003.8

AREAS OF CIRCLES. Diameters in Feet and Inches, Areas in Square Feet

Feet	Inches											
	0	1	2	3	4	5	6	7	8	9	10	11
0	.0000	.0055	.0218	.0491	.0873	.1364	.1963	.2673	.3491	.4418	.5454	.6600
1	.7854	.9218	1.069	1.227	1.396	1.576	1.767	1.969	2.182	2.405	2.640	2.885
2	3.142	3.409	3.687	3.976	4.276	4.587	4.909	5.241	5.585	5.940	6.305	6.681
3	7.069	7.467	7.876	8.296	8.727	9.168	9.621	10.08	10.56	11.04	11.54	12.05
4	12.57	13.10	13.64	14.19	14.75	15.32	15.90	16.50	17.10	17.72	18.35	18.99
5	19.63	20.29	20.97	21.65	22.34	23.04	23.76	24.48	25.22	25.97	26.73	27.49
6	28.27	29.07	29.87	30.68	31.50	32.34	33.18	34.04	34.91	35.78	36.67	37.57
7	38.48	39.41	40.34	41.28	42.24	43.20	44.18	45.17	46.16	47.17	48.19	49.22
8	50.27	51.32	52.38	53.46	54.54	55.64	56.75	57.86	58.99	60.13	61.28	62.44
9	63.62	64.80	66.00	67.20	68.42	69.64	70.88	72.13	73.39	74.66	75.94	77.24
10	78.54	79.85	81.18	82.52	83.86	85.22	86.59	87.97	89.36	90.76	92.18	93.60
11	95.03	96.48	97.93	99.40	100.9	102.4	103.9	105.4	106.9	108.4	110.0	111.5
12	113.1	114.7	116.3	117.9	119.5	121.1	122.7	124.4	126.0	127.7	129.4	131.0
13	132.7	134.4	136.2	137.9	139.6	141.4	143.1	144.9	146.7	148.5	150.3	152.1
14	153.9	155.8	157.6	159.5	161.4	163.2	165.1	167.0	168.9	170.9	172.8	174.8

From Inches and Fractions of an Inch to Decimals of a Foot

Inches	1	2	3	4	5	6	7	8	9	10	11
Feet	.0833	.1667	.2500	.3333	.4167	.5000	.5833	.6667	.7500	.8333	.9167

Inches	⅛	¼	⅜	½	⅝	¾	⅞
Feet	.0104	.0208	.0313	.0417	.0521	.0625	.0729

Example. 5 ft. 7⅜ in. = 5.0 + 0.5833 + 0.0313 = 5.6146 ft.

Conversion Tables

All measures of length, area, and cubic measures in the following tables are derived from the international meter. The metric system of weights and measures was legalized and its use made permissive in the United States by an Act of Congress, passed in 1866. In 1891, the Office of Weights and Measures (now Bureau of Standards) fixed the value of the United States yard in terms of the international meter, as follows: **1 yard = 3600/3937 meters.** At the same time, the pound was fixed in terms of the international kilogram, as follows: **1 pound = 453.59243 grams.** Measures of capacity are based on the following relations: **1 cubic decimeter = 1 liter; 1 U. S. gallon = 231 cu. in.; 1 Imperial gallon (British) = 277.418 cu. in.** **A liter** is the volume occupied by 1 kilogram of water under a pressure of 76 cm. of mercury and at a temperature of 4 deg. C.

By **standard locality** is meant any locality where g_c = 980.665 cm. per sec. per sec., or 32.1740 ft. per sec. per sec. This value, g_0, is assumed to be the value of g at sea level and latitude 45 deg., and is called **standard gravity.**

Acceleration of Gravity

(U. S. Coast and Geodetic Survey, 1912)

Latitude, deg.	g		g/g_0	Latitude, deg.	g		g/g_0
	Cm./sec.^{2}	Ft./sec.^{2}			Cm./sec.^{2}	Ft./sec.^{2}	
0	978.0	32.088	0.9973	50	981.1	32.187	1.0004
10	978.2	32.093	0.9975	60	981.9	32.215	1.0013
20	978.6	32.108	0.9979	70	982.6	32.238	1.0020
30	979.3	32.130	0.9986	80	983.1	32.253	1.0024
40	980.2	32.158	0.9995	90	983.2	32.258	1.0026

Correction for altitude above sea level: −0.3 cm. per sec.^{2} for each 1000 meters, −0.003 ft. per sec.^{2} for each 1000 feet.

Force Equivalents

Dynes × 10^6	1	$\overline{0}.9807$ $\overline{1}.99149$	$\overline{0}.4448$ $\overline{1}.64819$	$\overline{0}.01383$ $\overline{2}.14067$
Kilograms	1.020 0.00848	1	$\overline{0}.4536$ $\overline{1}.65667$	$\overline{0}.01410$ $\overline{2}.14916$
Pounds	2.248 0.03518	2.205 0.34334	1	$\overline{0}.03108$ $\overline{2}.49249$
Poundals	72.33 1.85933	70.93 1.85084	32.17 1.50780	1

Length Equivalents

Centimeters	Inches	Feet	Yards	Meters	Chains	Kilometers	Miles[1]
1	0.3937 $\bar{1}.59517$	0.03281 $\bar{2}.51598$	0.01094 $\bar{2}.03886$	0.01 $\bar{2}.00000$	0.0_24971 $\bar{4}.69644$	10^{-5} $\bar{5}.00000$	0.0_36214 $\bar{6}.79335$
2.540 0.40483	1	0.0_26333 $\bar{4}.92082$	0.02778 $\bar{2}.44370$	0.0254 $\bar{2}.40483$	0.0_41263 $\bar{5}.10127$	0.0_4254 $\bar{5}.40483$	0.0_41578 $\bar{5}.19818$
30.48 1.48402	12 1.07918	1	0.3333 $\bar{1}.52288$	0.3048 $\bar{1}.48402$	0.01515 $\bar{2}.18046$	0.0_33098 $\bar{4}.48402$	0.0_31645 $\bar{4}.21608$
91.14 1.96114	36 1.55630	3 0.47712	1	0.9144 $\bar{1}.96114$	0.04545 $\bar{2}.65758$	0.0_39144 $\bar{4}.96114$	0.0_35682 $\bar{4}.75449$
100 2.00000	39.37 1.59517	3.281 0.51598	1.0936 0.03886	1 	0.04971 $\bar{2}.69644$	0.001 $\bar{3}.00000$	0.0_36214 $\bar{4}.79335$
2012 3.30356	792 2.89873	66 1.81954	22 1.34242	20.12 1.30356	1 	0.02012 $\bar{2}.30356$	0.0125 $\bar{2}.09691$
100000 5.00000	39370 4.59517	3281 3.51598	1093.6 3.03886	1000 3.00000	49.71 1.69644	1 	0.6214 $\bar{1}.79335$
160925 5.20665	63360 4.80182	5280 3.72268	1760 3.24551	1609 3.20665	80 1.90309	1.609 0.20665	1

[1] One Nautical mile = 6080.2 feet.

Area Equivalents

Square centimeters	Square meters	Square inches[1]	Square feet	Sqaure yards	Square miles
1	0.0001	0.155 $\bar{1}.19033$	0.001076 $\bar{3}.03197$	0.0001196 $\bar{4}.07773$	$0.0_{11}3861$ $\bar{11}.58670$
10,000 4.00000	1	1550 3.19033	10.76 1.03197	1.196 0.07773	0.0_43861 $\bar{7}.58670$
6.452 0.80967	0.0_26452 $\bar{4}.80967$	1 	0.006944 $\bar{3}.84164$	0.0011 $\bar{3}.04139$	0.0_7491 $\bar{10}.39637$
929 2.96803	0.0929 $\bar{2}.96803$	144 2.15836	1 	0.1111 $\bar{1}.04576$	0.0_53587 $\bar{8}.55473$
8361 39.2227	0.8361 $\bar{1}.92227$	1296 3.11260	9 0.95424[1]	1 	0.0_43228 $\bar{7}.50898$
	2589_28 6.41230		27878400 7.44527	3097600 6.49102	1
10^{10}	10^6		10764100 7.03197	1196011 6.07773	0.3861 $\bar{1}.58670$

[1] One circular mil = area of circle 0.001 in. in diam. 1,000,000 cir. mils = 1 cir. in. The equivalents are given in the heavier type. Logarithms of the equivalents are given immediately below.

Subscripts after any figure, 0_3, 9_4, etc., mean that that figure is to be repeated the indicated number of times.

Volume and Capacity Equivalents

Cubic inches	Cubic feet	Cubic yards	Cubic centimeters	Cubic meters	U. S. liquid measure		Liters
					Quarts	Gallons	
1 $\bar{1}.78550$	0.0_25787 $\bar{2}.76246$	0.0_42143 $\bar{5}.33109$	16.387 1.21450	0.0_416387 $\bar{5}.21450$	0.01732 $\bar{2}.23845$	0.0_34329 $\bar{3}.63639$	0.01639 $\bar{2}.21450$
1728 3.23754	1	0.03704 $\bar{2}.56864$	28317 4.45205	0.028317 $\bar{2}.45205$	29.92 1.47599	7.481 0.87393	28.32 1.45205
46656 4.66891	27 1.43136	1	764557 5.88341	0.764557 $\bar{1}.88341$	807.9 2.90736	202.0 2.30530	764.6 2.88341
0.061 $\bar{2}.78550$	0.0_4353 $\bar{5}.54796$	0.0_61308 $\bar{6}.11661$	1 10^{-6}	0.001057 $\bar{3}.02394$	0.0_32642 $\bar{4}.42188$	0.001
61024 4.78550	35.315 1.54796	1.308 0.11661	10^6	1	1057 3.02394	264.2 3.42188	1000
57.75 1.76155	0.03342 $\bar{2}.52401$	0.001238 $\bar{3}.09264$	946.3 2.97589	0.000946 $\bar{4}.97589$	1	0.25 $\bar{1}.39794$	0.9464 $\bar{1}.97606$
231 2.36361	0.1337 $\bar{1}.12607$	0.004951 3.69470	3785 3.57807	0.0003785 $\bar{4}.57807$	4 0.60206	1	3.785 0.57812
61.02 1.78550	0.03531 $\bar{2}.54795$	0.001308 $\bar{3}.11650$	1000	0.001	1.057 0.02394	0.2642 $\bar{1}.42188$	1

Velocity Equivalents

Centimeters per sec.	Meters per sec.	Meters per min.	Kilometers per hour	Feet per sec.	Feet per min.	Miles per hour	Knots
1	0.01	0.6 $\bar{1}.77815$	0.036 $\bar{2}.55630$	0.03281 $\bar{2}.51598$	1.9685 0.29414	0.02237 $\bar{2}.34965$	0.01942 $\bar{2}.28825$
100 2.00000	1	60 1.77815	3.6 0.55630	3.281 0.51598	196.85 2.29414	2.237 0.34965	1.942 0.28825
1.667 0.22184	0.01667 $\bar{2}.22184$	1	0.06 $\bar{2}.77815$	0.05468 $\bar{2}.73783$	3.281 0.51598	0.03728 $\bar{2}.57150$	0.03237 $\bar{2}.51018$
27.78 1.44370	0.2778 $\bar{1}.44370$	16.67 1.22184	1	0.9113 $\bar{1}.95968$	54.68 1.73783	0.6214 $\bar{1}.79335$	0.53960 $\bar{1}.73207$
30.48 1.48402	0.3048 $\bar{1}.48402$	18.29 1.26217	1.097 0.04032	1	60 1.77815	0.6818 $\bar{1}.83367$	0.59209 $\bar{1}.77226$
0.5080 $\bar{1}.70586$	0.005080 $\bar{3}.70586$	0.3048 $\bar{1}.48402$	0.01829 $\bar{2}.26217$	0.01667 $\bar{2}.22185$	1	0.01136 $\bar{2}.05553$	0.00987 $\bar{3}.99423$
44.70 1.65035	0.4470 $\bar{1}.65035$	26.82 1.42850	1.609 0.20670	1.467 0.16633	88 1.94448	1	0.86839 $\bar{1}.93871$
51.497 1.71178	0.51497 $\bar{1}.71178$	30.898 1.48993	1.8532 0.26793	1.68894 0.22761	101.337 2.00577	1.15155 0.06128	1

The equivalents are given in the heavier type. Logarithms of the equivalents are given immediately below.

Subscripts after any figure, 0_3, 9_4, etc., mean that that figure is to be repeated the indicated number of times.

Mass Equivalents

Kilograms	Grains	Ounces		Pounds		Tons		
		Troy and apoth.	Avoir-dupois	Troy and apoth.	Avoir-dupois	Short	Long	Metric
1	15432 4.18843	32.15 1.50719	35.27 1.54745	2.6792 0.42801	2.205 0.34333	0.0_21102 $\bar{3}.04230$	0.0_29842 $\bar{4}.99309$	0.001 $\bar{3}.00000$
0.0_46480 $\bar{5}.81157$	1	0.0_32083 $\bar{3}.31876$	0.0_32286 $\bar{3}.35902$	0.0_31736 $\bar{4}.23958$	0.0_31429 $\bar{4}.15490$	0.0_37143 $\bar{3}.85387$	0.0_37638 $\bar{3}.80465$	0.0_36480 $\bar{3}.81157$
0.03110 $\bar{2}.49281$	480 2.68124	1	1.09714 0.04026	0.06333 $\bar{2}.92082$	0.06857 $\bar{2}.83614$	0.0_43429 $\bar{5}.53511$	0.0_43061 $\bar{5}.48590$	0.0_43110 $\bar{5}.49281$
0.02835 $\bar{2}.45255$	437.5 2.64098	0.9115 $\bar{1}.95974$	1	0.07595 $\bar{2}.88056$	0.0625 $\bar{2}.79588$	0.0_43125 $\bar{5}.49485$	0.0_42790 $\bar{5}.44563$	0.0_42835 $\bar{5}.45255$
0.3732 $\bar{1}.57199$	5760 3.76042	12 1.07918	13.17 1.11944	1 0	0.8229 $\bar{1}.91532$	0.0_44114 $\bar{4}.61429$	0.0_43673 $\bar{4}.56508$	0.0_43732 $\bar{4}.57199$
0.4536 $\bar{1}.65667$	7000 3.84510	14.58 $1\frac{1}{2}$.16388	16 1.20412	1.215 0.08468	1 0	0.0005 $\bar{4}.69897$	0.0_34464 $\bar{4}.64975$	0.0_34536 $\bar{4}.65667$
907.2 2.95770	140_6 7.14613	29167 4.46489	320_3 4.50515	2431 3.38571	2000 3.30103	1 0	0.8929 $\bar{1}.95078$	0.9072 $\bar{1}.95770$
1016 3.00691	15680_4 7.19535	326_8 4.51411	35840 4.55437	2722 3.43492	2240 3.35025	1.12 0.04922	1 0	1.016 0.00691
1000 3.00000	15432356 7.18843	32151 4.50719	35274 4.54745	2679 3.42801	2205 3.34333	1.102 0.04230	0.9842 $\bar{1}.99309$	1 0

Pressure Equivalents

Megabars or megadynes per sq. cm.	Kilograms per sq. cm. (Metric atmospheres)	Pounds per sq. in.	Short tons per sq. ft.	Atmospheres	Columns of mercury at temperature 15 deg. C.		Columns of water at temperature 15 deg. C.		
					Meters	Inches	Meters	Inches	Feet
1	1.0197 0.00848	14.50 1.16148	1.044 0.01882	0.9869 $\bar{1}.99427$	0.7500 $\bar{1}.87508$	29.53 1.47025	10.21 1.00886	401.8 2.60402	33.48 1.52484
0.9807 $\bar{1}.99152$	1	14.22 1.15300	1.024 0.01034	0.9678 $\bar{1}.98579$	0.7355 $\bar{1}.86660$	28.96 1.46177	10.01 1.00028	394.0 2.59555	32.84 1.51636
0.06895 $\bar{2}.83853$	0.07031 $\bar{2}.84700$	1	0.072 $\bar{2}.85733$	0.06804 $\bar{2}.83279$	0.05171 $\bar{2}.71360$	2.036 0.30876	0.7037 $\bar{1}.84738$	27.70 1.44254	2.309 0.36336
0.9576 $\bar{1}.98119$	0.9765 $\bar{1}.98966$	13.89 1.14267	1	0.9450 $\bar{1}.97545$	0.7182 $\bar{1}.85627$	28.28 1.45143	9.773 0.99004	384.8 2.58521	32.06 1.50608
1.0133 0.00573	1.0333 0.01421	14.70 1.16722	1.058 0.02455	1	0.76 $\bar{1}.88081$	29.92 1.47598	10.34 1.01459	407.2 2.60976	33.93 1.53058
1.3333 0.12493	1.3596 0.13340	19.34 1.28640	1.392 0.14373	1.316 0.11919	1	39.37 1.59517	13.61 1.13378	535.7 2.72894	44.64 1.64976
0.03386 $\bar{2}.52975$	0.03453 $\bar{2}.53823$	0.4912 $\bar{1}.69124$	0.03536 $\bar{2}.54857$	0.03342 $\bar{2}.52402$	0.02540 $\bar{2}.40484$	1	0.3456 $\bar{1}.53861$	13.61 1.13378	1.134 0.05460
0.09798 $\bar{2}.99114$	0.09991 $\bar{2}.99962$	1.421 0.15262	0.1023 $\bar{1}.00996$	0.09670 $\bar{2}.98541$	0.07349 $\bar{2}.86622$	2.893 0.46139	1	39.37 1.59517	3.281 0.55198
0.002489 $\bar{3}.39598$	0.002538 $\bar{3}.40446$	0.03610 $\bar{2}.55746$	0.002599 $\bar{3}.41479$	0.002456 $\bar{3}.39024$	0.001867 $\bar{3}.27106$	0.07349 $\bar{2}.86622$	0.02540 $\bar{2}.40484$	1	0.08333 $\bar{2}.92082$
0.02986 $\bar{2}.47516$	0.03045 $\bar{2}.48364$	0.4332 $\bar{1}.63664$	0.03119 $\bar{2}.49397$	0.02947 $\bar{2}.46942$	0.02240 $\bar{2}.35024$	0.8819 $\bar{1}.94540$	0.3048 $\bar{1}.48402$	12 1.07918	1

The equivalents are given in the heavier type. Logarithms of the equivalents are given immediately below.

Subscripts after any figure, 0_3, 9_4, etc., mean that that figure is to be repeated the indicated number of times.

Energy or Work Equivalents

Joules = 10^7 ergs	Kilogram-meters	Foot-pounds	Kilo-watt-hours	Cheval-vapeur-hours	Horse-power-hours	Liter-atmos-pheres	Kilo-gram-calories	British thermal units
1	0.10197	0.7376	$0.0_2$2778	$0.0_3$3777	$0.0_3$3725	0.009869	$0.0_2$2390	$0.0_3$9486
	$\bar{1}$.00848	$\bar{1}$.86780	7.44370	7.57711	7.57118	$\bar{3}$.99427	$\bar{4}$.37848	$\bar{4}$.97709
9.80665	1	7.233	$0.0_2$2724	$0.0_4$37037	$0.0_3$3653	0.09678	0.002344	0.009302
0.9915207		0.85932	6.43522	6.56863	6.56265	$\bar{2}$.98579	$\bar{3}$.37000	$\bar{3}$.96861
1.356	0.1383	1	$0.0_2$3766	$0.0_4$51206	$0.0_4$50505	0.01338	$0.0_3$3241	0.001286
0.13220	$\bar{1}$.14068		7.57590	7.70933	7.70333	$\bar{2}$.12647	$\bar{4}$.51068	$\bar{3}$.10929
3.6×10^6	3.671×10^5	2.655×10^6	1	1.3596	1.341	35528	860.5	3415
6.55630	5.56478	6.42410		0.13342	0.12743	4.55057	2.93478	3.53339
2.648×10^6	270000.	1.9529×10^6	0.7355	1	0.9863	26131.	632.9	2512 •
6.42288	5.43136	6.29068	$\bar{1}$.86658		$\bar{1}$.99401	4.41715	2.80135	3.39998
2.6845×10^6	2.7375×10^5	1.98×10^6	0.7457	1.0139	1	26494	641.7	2547
6.42887	5.43735	6.29667	$\bar{1}$.87257	0.00598		4.42314	2.80735	3.40595
101.33	10.333	74.73	$0.0_2$2815	$0.0_3$3827	$0.0_3$3774	1	0.02422	0.09612
2.00678	1.01421	1.87358	$\bar{3}$.44943	$\bar{3}$.58284	$\bar{3}$.57686		$\bar{2}$.38425	$\bar{2}$.98281
4183	426.6	3086	0.001162	0.001580	0.001558	41.29	1	3.968
3.62153	2.63000	3.48932	$\bar{3}$.06532	$\bar{3}$.19864	$\bar{3}$.19265	1.61579		0.59861
1054	107.5	777.52	$0.0_2$2928	$0.0_3$3981	$0.0_3$3927	10.40	0.25200	1
3.02291	2.03139	2.89071	$\bar{4}$.46661	$\bar{4}$.60008	$\bar{4}$.59405	1.01719	$\bar{1}$.40139	

Power Equivalents

Horse power (550 standard ft.-lb. per sec.)	Kilo-watts (1000 joules per sec.)	Cheval-vapeur (metric H.P.)	Ponce-lets	M.-kg. per sec.	Ft.-lbs. per sec.	Kg.-cal. per sec.	B.t.u per sec.
1	0.7457	1.014	0.7604	76.04	550	0.1783	0.7074
	$\bar{1}$.87256	0.00599	$\bar{1}$.88105	1.88105	2.74036	$\bar{1}$.25104	$\bar{1}$.84965
1.341	1	1.360	1.020	102.0	737.6	0.2390	0.9486
0.12743		0.13343	0.00848	2.00848	2.86780	$\bar{1}$.37848	$\bar{1}$.97709
0.9863	0.7355	1	0.75	75	542.3	0.1758	0.6977
$\bar{1}$.99402	$\bar{1}$.86659		$\bar{1}$.87506	1.87506	2.73438	$\bar{1}$.24506	$\bar{1}$.84367
1.315	0.9807	1.333	1	100	723.3	0.2344	0.9303
0.11896	$\bar{1}$.99152	0.12493		2.00000	2.85932	$\bar{1}$.37000	$\bar{1}$.96861
0.01315	0.009807	0.01333	0.01	1	7.233	0.002344	0.009303
$\bar{2}$.11896	$\bar{3}$.99152	$\bar{2}$.12493	$\bar{2}$.00000		0.85932	$\bar{3}$.37000	$\bar{3}$.96861
0.00182	0.001356	0.00184	0.00138	0.1383	1	$0.0_3$3241	0.001286
$\bar{3}$.25946	$\bar{3}$.13219	$\bar{3}$.26562	$\bar{3}$.14067	$\bar{1}$.14067		$\bar{4}$.51068	$\bar{3}$.10929
5.610	4.183	5.688	4.266	426.6	3086	1	3.968
0.74896	0.62153	0.75494	0.63000	2.63000	3.48932		0.59861
1.414	1.054	1.433	1.075	107.5	777.5	0.2520	1
0.15085	0.02291	0.15632	0.03139	2.03139	2.89071	$\bar{1}$.40138	

The equivalents are given in the heavier type. Logarithms of the equivalents are given immediately below.

Subscripts after any figure, 0_3, 9_4, etc., mean that that figure is to be repeated the indicated number of times.

Density Equivalents and Conversion Factors

Equivalents					Conversion factors				
Grams per cu. cm.	Lb. per cu. in.	Lbs. per cu. ft.	Short tons (2000 lbs.) per cu. yd.	Lbs. per U. S. gal.		Grams per cu. cm. to lbs. per cu. ft.	Lb. per cu. ft. to grams per cu. cm.	Grams per cu. cm. to short tons per cu. yd.	Short tons per cu. yd. to grams per cu. cm.
1	0.03613	62.43	0.8428	8.345	1	62.43	0.01602	0.8428	1.186
	$\bar{2}$.55787	1.79539	$\bar{1}$.92572	0.92143	2	124.90	0.03204	1.6860	2.373
27.68	1	1728	23.33	231	3	187.30	0.04806	2.5280	3.600
1.44217		3.23754	1.36792	2.36361	4	249.70	0.06407	3.3710	4.746
0.01602	0.0$_2$5787	1	0.0135	0.1337	5	312.40	0.08009	4.2140	5.933
$\bar{2}$.20466	$\bar{4}$.76245		$\bar{2}$.13033	$\bar{1}$.12613	6	374.60	0.09611	5.0570	7.119
1.186	0.04286	74.07	1	9.902	7	437.00	0.11210	5.9000	8.306
0.07428	$\bar{2}$.63205	1.86964		0.99572	8	499.40	0.12820	6.7420	9.492
0.1198	0.004329	7.481	0.1010	1	9	561.90	0.14420	7.5850	10.680
$\bar{1}$.07855	$\bar{3}$.63639	0.87396	$\bar{1}$.00432		10	624.30	0.16020	8.4280	11.870

Conversion of Heat Transmission and Conduction

	Small calories per sq. cm. to B.t.u. per sq. ft.	B.t.u. per sq. ft. to small calories per sq. cm.	Small calories per sq. cm. per cm. to B.t.u. per sq. ft. per in.	B.t.u. per sq. ft. per in. to small calories per sq. cm. per cm.	Small calories per sec. per sq. cm. per 1 deg. C. per cm. thick, to B.t.u. per hr. per sq. ft. per 1 deg. F. per in. thick	B.t.u. per hr. per sq. ft. per 1 deg. F. per in. thick to small calories per sec. per sq. cm. per 1 deg. C. per cm. thick
1	3.687	0.2712	1.451	0.6892	2.903×10^0	$0.0_3 3445$
2	7.374	0.5424	2.902	1.378	5.806×10^0	$0.0_3 6890$
3	11.06	0.8136	4.353	2.068	8.709×10^0	$0.0_3 1034$
4	14.75	1.085	5.804	2.757	11.61×10^0	$0.0_3 1378$
5	18.44	1.356	7.255	3.446	14.52×10^0	$0.0_3 1722$
6	22.12	1.627	8.706	4.135	17.42×10^0	$0.0_3 2067$
7	25.81	1.898	10.16	4.824	20.32×10^0	$0.0_3 2412$
8	29.50	2.170	11.61	5.514	23.22×10^0	$0.0_3 2756$
9	33.18	2.441	13.06	6.203	26.13×10^0	$0.0_3 3100$

NOTE. 1 gram-calorie per sq. cm. = 3.687 B.t.u. per sq. ft.
1 gram-calorie per sq. cm. per cm. = 1.451 B.t.u. per sq. ft. per in.
1 gram-calorie per sec. per sq. cm. for a temp. grad. of 1 deg. C. per cm.
= 360 kilogram-calories per hour per sq. m. for a temp. grad. of 1 deg. C. per m.
= 2.903×10^0 B.t.u. per hour per sq. ft. for a temp. grad. of 1 deg. F. per in.

MATHEMATICAL FORMULÆ

TRIGONOMETRIC FORMULÆ

$\sin(-x) = -\sin x$; $\cos(-x) = \cos x$; $\tan(-x) = -\tan x$.

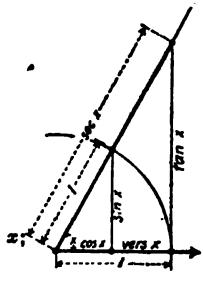

FIG. 1

$\sin^2 x + \cos^2 x = 1$; $\tan x = \dfrac{\sin x}{\cos x}$; $\cot x = \dfrac{1}{\tan x} = \dfrac{\cos x}{\sin x}$;

$1 + \tan^2 x = \sec^2 x = \dfrac{1}{\cos^2 x}$; $1 + \cot^2 x = \operatorname{cosec}^2 x = \dfrac{1}{\sin^2 x}$;

$\sin x = \sqrt{1 - \cos^2 x} = \dfrac{\tan x}{\sqrt{1 + \tan^2 x}} = \dfrac{1}{\sqrt{1 + \cot^2 x}}$;

$\cos x = \sqrt{1 - \sin^2 x} = \dfrac{1}{\sqrt{1 + \tan^2 x}} = \dfrac{\cot x}{\sqrt{1 + \cot^2 x}}$;

$\sin 2x = 2 \sin x \cos x$; $\sin x = 2 \sin \frac{1}{2}x \cos \frac{1}{2}x$;

$\cos 2x = \cos^2 x - \sin^2 x = 1 - 2\sin^2 x = 2\cos^2 x - 1$;

$\tan 2x = \dfrac{2 \tan x}{1 - \tan^2 x}$; $\cot 2x = \dfrac{\cot^2 x - 1}{2 \cot x}$;

$\sin 3x = 3 \sin x - 4 \sin^3 x$; $\tan 3x = \dfrac{3 \tan x - \tan^3 x}{1 - 3 \tan^2 x}$;

$\cos 3x = 4 \cos^3 x - 3 \cos x$;

$\sin^{-1} x$ (read: anti-sine of x, or inverse sine of x; sometimes written arc sin x) means the principle angle whose sine is x. Similarly for $\cos^{-1} x$, $\tan^{-1} x$, etc. (The principal angle means an angle between $-90°$ and $+90°$ in case of $\sin^{-1}$ and $\tan^{-1}$, and between $0°$ and $180°$ in the case of $\cos^{-1}$.)

In the following the + or − sign is to be used according to the sign of the left-hand side of the equation.

$\sin(nx) = n \sin x \cos^{n-1} x - (n)_3 \sin^3 x \cos^{n-3} x + (n)_5 \sin^5 x \cos^{n-5} x - \ldots$;

$\cos(nx) = \cos^n x - (n)_2 \sin^2 x \cos^{n-2} x + (n)_4 \sin^4 x \cos^{n-4} x - \ldots$;

where $(n)_2$, $(n)_3$, are the binomial coefficients.

$\sin \frac{1}{2}x = \pm \sqrt{\frac{1}{2}(1 - \cos x)}$; $1 - \cos x = 2 \sin^2 \frac{1}{2}x$;

$\cos \frac{1}{2}x = \pm \sqrt{\frac{1}{2}(1 + \cos x)}$; $1 + \cos x = 2 \cos^2 \frac{1}{2}x$;

$\tan \frac{1}{2}x = \pm \sqrt{\dfrac{1 - \cos x}{1 + \cos x}} = \dfrac{\sin x}{1 + \cos x} = \dfrac{1 - \cos x}{\sin x}$;

$\tan\left(\dfrac{x}{2} + 45°\right) = \pm \sqrt{\dfrac{1 + \sin x}{1 - \sin x}}$

$\sin(x + y) = \sin x \cos y + \cos x \sin y$;
$\cos(x + y) = \cos x \cos y - \sin x \sin y$;
$\tan(x + y) = [\tan x + \tan y] \div [1 - \tan x \tan y]$;
$\cot(x + y) = [\cot x \cot y - 1] \div [\cot x + \cot y]$;
$\sin(x - y) = \sin x \cos y - \cos x \sin y$;
$\cos(x - y) = \cos x \cos y + \sin x \sin y$;
$\tan(x - y) = [\tan x - \tan y] \div [1 + \tan x \tan y]$;

$\cot(x-y) = [\cot x \cot y + 1] \div [\cot y - \cot x]$;
$\sin x + \sin y = 2 \sin \frac{1}{2}(x+y) \cos \frac{1}{2}(x-y)$;
$\sin x - \sin y = 2 \cos \frac{1}{2}(x+y) \sin \frac{1}{2}(x-y)$;
$\cos x + \cos y = 2 \cos \frac{1}{2}(x+y) \cos \frac{1}{2}(x-y)$;
$\cos x - \cos y = -2 \sin \frac{1}{2}(x+y) \sin \frac{1}{2}(x-y)$;

$$\tan x + \tan y = \frac{\sin(x+y)}{\cos x \cos y}; \quad \cot x + \cot y = \frac{\sin(x+y)}{\sin x \sin y};$$

$$\tan x - \tan y = \frac{\sin(x-y)}{\cos x \cos y}; \quad \cot x - \cot y = \frac{\sin(y-x)}{\sin x \sin y};$$

$\sin^2 x - \sin^2 y = \cos^2 y - \cos^2 x = \sin(x+y) \sin(x-y)$;
$\cos^2 x - \sin^2 y = \cos^2 y - \sin^2 x = \cos(x+y) \cos(x-y)$;
$\sin(45° + x) = \cos(45° - x)$; $\tan(45° + x) = \cot(45° - x)$;
$\sin(45° - x) = \cos(45° + x)$; $\tan(45° - x) = \cot(45° + x)$.

Relations Between Three Angles of Plane Triangles

$A + B + C = 180°$.
$\sin A + \sin B + \sin C = 4 \cos \frac{1}{2}A \cos \frac{1}{2}B \cos \frac{1}{2}C$;
$\cos A + \cos B + \cos C = 4 \sin \frac{1}{2}A \sin \frac{1}{2}B \sin \frac{1}{2}C + 1$;
$\sin A + \sin B - \sin C = 4 \sin \frac{1}{2}A \sin \frac{1}{2}B \cos \frac{1}{2}C$;
$\cos A + \cos B - \cos C = 4 \cos \frac{1}{2}A \cos \frac{1}{2}B \sin \frac{1}{2}C - 1$;
$\sin^2 A + \sin^2 B + \sin^2 C = 2 \cos A \cos B \cos C + 2$;
$\sin^2 A + \sin^2 B - \sin^2 C = 2 \sin A \sin B \cos C$;
$\tan A + \tan B + \tan C = \tan A \tan B \tan C$;
$\cot \frac{1}{2}A + \cot \frac{1}{2}B + \cot \frac{1}{2}C = \cot \frac{1}{2}A \cot \frac{1}{2}B \cot \frac{1}{2}C$
$\cot A \cot B + \cot A \cot C + \cot B \cot C = 1$;
$\sin 2A + \sin 2B + \sin 2C = 4 \sin A \sin B \sin C$;
$\sin 2A + \sin 2B - \sin 2C = 4 \cos A \cos B \sin C$.

Relations Between Sides and Angles of Plane Triangles

The parts of a plane triangle are its three sides, a, b, c, and its three angles A, B, C (A being opposite a, B opposite b, C opposite C, and $A + B + C = 180°$). A triangle is, in general, determined by any three parts (not all angles).

$a : b : c :: \text{Sin } A : \text{Sin } B : \text{Sin C}$, known as **Law of Sines.**
$c^2 = a^2 + b^2 - 2ab \cos C$, known as **Law of Cosines.**
$c \cos B + b \cos C = a$; $\tan \frac{1}{2}(A - B) = [(a - b) \div (a + b)] \cot \frac{1}{2} C$.
$\cos A = (b^2 + c^2 - a^2) \div 2bc$ or $\text{vers } A = [a^2 - (b - c)^2] \div 2bc$
$\tan \frac{1}{2}B = r \div (s - b)$, $\tan \frac{1}{2}C = r \div (s - c)$, where $s = \frac{1}{2}(a + b + c)$, and $r = \sqrt{(s-a)(s-b)(s-c) \div s}$.

$\sin \frac{1}{2}A = \sqrt{(s-b)(s-c) \div bc}$; $\cos \frac{1}{2}A = \sqrt{s(s-a) \div bc}$.

Area $= \frac{1}{2}ab \sin C = \sqrt{s(s-a)(s-b)(s-c)} = rs$, where $s = \frac{1}{2}(a + b + c)$, and r = radius of inscribed circle $= \sqrt{(s-a)(s-b)(s-c) \div s}$.
Radius of circumscribed circle $= R$, where

$$2R = a \div \sin A = b \div \sin B = c \div \sin C; \quad r = 4R \sin \frac{A}{2} \sin \frac{B}{2} \sin \frac{C}{2} = \frac{abc}{4Rs}.$$

The length of the bisector of the angle C is

$$z = \frac{2\sqrt{abs(s-c)}}{a+b} = \frac{\sqrt{ab[(a+b)^2 - c^2]}}{a+b}.$$

The median from C to the middle point of c is $m = \frac{1}{2}\sqrt{2(a^2 + b^2) - c^2}$.

Relations Between Angles and Sides of Spherical Triangles

Let a, b, c be the sides of the spherical triangle, that is, portions of arcs of great circles of the sphere; and let A, B, C be the angles of the triangle, that is, the angles made by tangents drawn to the sides at their points of intersection on the sphere. The sum of the angles will always be greater than two right angles, and may be nearly six right angles. The angle $E = A + B + C - 180°$ is called the **spherical excess** of the triangle.

$$\frac{\sin a}{\sin A} = \frac{\sin b}{\sin B}; \quad \frac{\sin b}{\sin B} = \frac{\sin c}{\sin C}; \quad \frac{\sin c}{\sin C} = \frac{\sin a}{\sin A}$$

$$\cos a = \cos b \cos c + \sin b \sin c \cos A,$$

with similar formulæ for $\cos b$ and $\cos c$.

$$\cos A = -\cos B \cos C + \sin B \sin C \cos a,$$

with similar formulæ for $\cos B$ and $\cos C$.

In the special case of a right spherical triangle, in which $C = 90°$,

$$\cos c = \cos a \cos b = \cot A \cot B; \quad \cos a = \frac{\cos A}{\sin B}; \quad \cos b = \frac{\cos B}{\sin A};$$

$$\sin A = \frac{\sin a}{\sin c}; \quad \cos A = \frac{\tan b}{\tan c}; \quad \tan A = \frac{\tan a}{\sin b}.$$

$$\frac{\text{The area of a spherical triangle}}{\text{area of a great circle}} = \frac{\text{spherical excess}}{180°}.$$

DIFFERENTIAL FORMULÆ

In the following formulæ a, and n are constants, x, y and z are variables; $e = 2.71828$.

1. $d\,a = 0$.
2. $d(a + x) = dx$.
3. $d(ax) = a dx$.
4. $d(x + y + z + \ldots) = dx + dy + dx + \ldots$
5. $d(xy) = x dy + y dx$. For general case see 6.
6. $d(xyz\ldots) = (xyz\ldots)\left(\frac{dx}{x} + \frac{dy}{y} + \frac{dz}{z} + \ldots\right)$.
7. $d(x^n) = nx^{n-1}dx$. This formula is used in the case of reciprocals and roots as follows: $dx^2 = 3x^2dx$; $d\left(\frac{1}{x}\right) = dx^{-1} = -1x^{-2}\,dx = -\frac{dx}{x^2}$; $d\sqrt{x} = dx^{\frac{1}{2}} = \frac{1}{2}x^{-\frac{1}{2}}dx = \frac{1}{2}\frac{dx}{\sqrt{x}}$.
8. $d(e^x) = e^x dx$.
9. $d(a^x) = (\log_e a)\, a^x dx$.
10. $d \log_e x = \frac{dx}{x}$.
11. $d \log_{10} x = (\log_{10} e)\frac{dx}{x} = (0.4343\ldots)\frac{dx}{x}$.
12. $d \sin x = \cos x dx$.
13. $d \operatorname{cosec} x = -\cot x \operatorname{cosec} x dx$.
14. $d \cos x = -\sin x dx$.
15. $d \sec x = \tan x \sec x dx$.
16. $d \tan x = \sec^2 x dx$.
17. $d \cot x = -\operatorname{cosec}^2 x dx$.
18. $d \sin^{-1} x = \frac{dx}{\sqrt{1 - x^2}}$;
19. $d \operatorname{cosec}^{-1} x = -\frac{dx}{x\sqrt{x^2 - 1}}$.

20. $d\cos^{-1}x = -\frac{dx}{\sqrt{1-x^2}}$.

21. $d\sec^{-1}x = \frac{dx}{x\sqrt{x^2-1}}$.

22. $d\tan^{-1}x = \frac{dx}{1+x^2}$.

23. $d\cot^{-1}x = -\frac{dx}{1+x^2}$.

24. $d\log_e \sin x = \cot x dx$.

25. $d\log_e \tan x = \frac{2dx}{\sin 2x}$.

26. $d\log_e \cos x = -\tan x dx$.

27. $d\log_e \cot x = -\frac{2dx}{\sin 2x}$.

INDEFINITE INTEGRALS

To integrate a function it is necessary to find a function which, when differentiated, will produce the given function. This may be done by direct recognition, or by transforming the given function into a form where such recognition is possible. The following table gives the most common integrals; for a more extended list the reader is referred to B. O. Peirce's "Table of Integrals" (Ginn & Co.).

General Formulæ

1. $\int adu = a\int du = au + C$. 2. $\int (u+v)dx = \int udx + \int vdx$
3. $\int udv = uv - \int vdu$ 4. $\int f(x)dx = \int f[F(y)]F'(y)dy,\ x = F(y)$
5. $\int dy \int f(x,y)dx = \int dx \int f(x,y)dy$.

Fundamental Integrals

6. $\int x^n dx = \frac{x^{n+1}}{n+1} + C$; when n has any value except -1

7. $\int \frac{dx}{x} = \log_e x + C = \log_e cx$ 8. $\int e^x dx = e^x + C$

9. $\int \sin x dx = -\cos x + C$ 10. $\int \cos x dx = \sin x + C$

11. $\int \frac{dx}{\sin^2 x} = -\cot x + C$ 12. $\int \frac{dx}{\cos^2 x} = \tan x + C$

13. $\int \frac{dx}{\sqrt{1-x^2}} = \sin^{-1}x + C = -\cos^{-1}x + c$

14. $\int \frac{dx}{1+x^2} = \tan^{-1}x + C = -\cot^{-1}x + c$

Rational Functions

15. $\int (a+bx)^n dx = \frac{(a+bx)^{n+1}}{(n+1)b} + C$

16. $\int \frac{dx}{a+bx} = \frac{1}{b}\log_e(a+bx) + C = \frac{1}{b}\log_e c(a+bx)$

17 $\int \frac{1}{x^2}dx = -\frac{1}{x} + C$ 18. $\int \frac{dx}{(a+bx)^2} = -\frac{1}{b(a+bx)} + C$

19. $\int \frac{dx}{1-x^2} = \frac{1}{2}\log_e \frac{1+x}{1-x} + C = \tanh^{-1}x + C$, when $x < 1$

20. $\int \frac{dx}{x^2-1} = \frac{1}{2}\log_e \frac{x-1}{x+1} + C = -\coth^{-1} x + C,$ when $x > 1$

21. $\int \frac{dx}{a+bx^2} = \frac{1}{\sqrt{ab}} \tan^{-1}\left(\sqrt{\frac{b}{a}}x\right) + C$

22. $\int \frac{dx}{a-bx^2} = \frac{1}{2\sqrt{ab}} \log_e \frac{\sqrt{ab}+bx}{\sqrt{ab}-bx} + C$
$= \frac{1}{\sqrt{ab}} \tanh^{-1}\left(\sqrt{\frac{b}{a}}x\right) + C$ } when $a > 0$, $b > 0$

23. $\int \frac{dx}{a+2bx+cx^2} = \frac{1}{\sqrt{ac-b^2}} \tan^{-1} \frac{b+cx}{\sqrt{ac-b^2}} + C$ } when $ac - b^2 > 0$;
$= \frac{1}{2\sqrt{b^2-ac}} \log_e \frac{\sqrt{b^2-ac}-b-cx}{\sqrt{b^2-ac}+b+cx} + C$
$= -\frac{1}{\sqrt{b^2-ac}} \tanh^{-1} \frac{b+cx}{\sqrt{b^2-ac}} + C.$ } when $b^2 - ac > 0$;

24. $\int x^{m-1}(a+bx)^n dx = \frac{x^{m-1}(a+bx)^{n+1}}{(m+n)b}$
$- \frac{(m-1)a}{(m+n)b} \int x^{m-2}(a+bx)^n dx$
$= \frac{x^m(a+bx)^n}{m+n} + \frac{na}{m+n} \int x^{m-1}(a+bx)^{n-1} dx$

Irrational Functions

25. $\int \sqrt{a+bx}\, dx = \frac{2}{3b}(\sqrt{a+bx})^3 + C$

26. $\int \frac{dx}{\sqrt{a+bx}} = \frac{2}{b}\sqrt{a+bx} + C$

27. $\int \sqrt{a^2+x^2}\, dx = \frac{x}{2}\sqrt{a^2+x^2} + \frac{a^2}{2}\log_e\left(x+\sqrt{a^2+x^2}\right) + C$
$= \frac{x}{2}\sqrt{a^2+x^2} + \frac{a^2}{2}\sinh^{-1}\frac{x}{a} + c$

28. $\int \sqrt{a^2-x^2}\, dx = \frac{x}{2}\sqrt{a^2-x^2} + \frac{a^2}{2}\sin^{-1}\frac{x}{a} + C$

29. $\int \sqrt{x^2-a^2}\, dx = \frac{x}{2}\sqrt{x^2-a^2} - \frac{a^2}{2}\log_e\left(x+\sqrt{x^2-a^2}\right) + C$
$= \frac{x}{2}\sqrt{x^2-a^2} - \frac{a^2}{2}\cosh^{-1}\frac{x}{a} + c$

Transcendental Functions

30. $\int a^x dx = \frac{a^x}{\log_e a} + C$

31. $\int x^n e^{ax}\, dx = \frac{x^n e^{ax}}{a}\left[1 - \frac{n}{ax} + \frac{n(n-1)}{a^2x^2} - \ldots \pm \frac{n!}{a^n x^n}\right] + C$

32. $\int \log_e x\, dx = x\log_e x - x + C$

33. $\int \frac{\log_e x}{x^2}\,dx = -\frac{\log_e x}{x} - \frac{1}{x} + C$

34. $\int \frac{(\log_e x)^n}{x}\,dx = \frac{1}{n+1}(\log_e x)^{n+1} + C$

35. $\int \sin^2 x dx = -\frac{1}{4}\sin 2x + \frac{1}{2}x + C = -\frac{1}{2}\sin x \cos x + \frac{1}{2}x + C$

36. $\int \cos^2 x dx = \frac{1}{4}\sin 2x + \frac{1}{2}x + C = \frac{1}{2}\sin x \cos x + \frac{1}{2}x + C$

37. $\int \sin mx dx = -\frac{\cos mx}{m} + C$ 38. $\int \cos mx dx = \frac{\sin mx}{m} + C$

39. $\int \tan x dx = -\log_e \cos x + C$ 40. $\int \cot x dx = \log_e \sin x + C$

41. $\int \frac{dx}{\sin x} = \log_e \tan \frac{x}{2} + C$ 42. $\int \frac{dx}{\cos x} = \log_e \tan\left(\frac{\pi}{4} + \frac{x}{2}\right) + C$

43. $\int \frac{dx}{1+\cos x} = \tan \frac{x}{2} + C$ 44. $\int \frac{dx}{1-\cos x} = -\cot \frac{x}{2} + C$

45. $\int \sin^{-1} x dx = x \sin^{-1} x + \sqrt{1-x^2} + C$

46. $\int \cos^{-1} x dx = x \cos^{-1} x - \sqrt{1-x^2} + C$

47. $\int \tan^{-1} x dx = x \tan^{-1} x - \frac{1}{2}\log_e (1+x^2) + C$

48. $\int \cot^{-1} x dx = x \cot^{-1} x + \frac{1}{2}\log_e (1+x^2) + C$

DEFINITE INTEGRALS

The definite integral of $f(x)dx$ from $x = a$ to $x = b$, denoted by $\int_a^b f(x)dx$, is the limit (as n increases indefinitely) of a sum of n terms:

$$\int_a^b f(x)dx = \lim_{n=\infty} [f(x_1)\Delta x + f(x_2)\Delta x + f(x_3)\Delta x + \ldots + f(x_n)\Delta x],$$

built up as follows: Divide the interval from a to b into n equal parts, and call each part Δx, $= (b-a) \div n$; in each of these intervals take a value of x (say x_1, x_2, . . . x_n), find the value of the function $f(x)$ at each of these points, and multiply it by Δx, the width of the interval; then take the limit of the sum of the terms thus formed, when the number of terms increases indefinitely, while each individual term approaches zero.

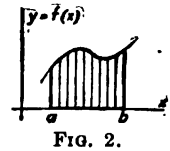

FIG. 2.

Geometrically, $\int_a^b f(x)dx$ is the area bounded by the curve $y = f(x)$, the x-axis, and the ordinates $x = a$ and $x = b$ (Fig. 2); that is, briefly, the "area under the curve, from a to b." The **fundamental theorem** for the evaluation of a definite integral is the following:

$$\int_a^b f(x)dx = \left[\int f(x)dx\right]_{x=b} - \left[\int f(x)dx\right]_{x=a};$$

that is, the definite integral is equal to the difference between two values of any one of the indefinite integrals of the function in question. In other words, the limit of a sum can be found whenever the function can be integrated.

Properties of Definite Integrals.

$$\int_a^b = -\int_b^a; \quad \int_a^c + \int_c^b = \int_a^b.$$

Theorem on Change of Variable. In evaluating $\int_{x=a}^{x=b} f(x)dx$, $f(x)dx$ may be replaced by its value in terms of a new variable t and dt, and $x = a$

and $x = b$ by the corresponding values of t, provided that throughout the interval the relation between x and t is a one-to-one correspondence (that is, to each value of x there corresponds one and only one value of t, and to each value of t there corresponds one and only one value of x).

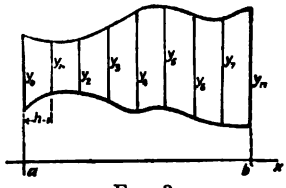

FIG. 3

Approximate Methods of Integration. (1) Expand the function in a power series, and integrate term by term.

(2). Use Simpson's rule or Tchibyscheff's rule.

(3). Use a planimeter. See p. 1414.

Simpson's Rule. Divide the given area, Fig. 3, into an even number of panels by parallel ordinates drawn at constant distance h apart; denote the length of these ordinates by $y_0, y_1, y_2, \ldots y_n$. Then to the sum of the end ordinates add four times the odd and twice the even, and multiply the sum thus obtained by one-third the common interval between ordinates to obtain the area: Area $= \frac{1}{3}h[(y_0 + y_n) + 4(y_1 + y_3 + y_5 \ldots) + 2(y_2 + y_4 + y_6 \ldots)]$. The greater the number of divisions the more accurate the result.

The number of panels need not necessarily be even; in a case where an odd number of panels is more advisable, to obtain the area add to the first ordinate four times the sum of the odd ones (except the last ordinate) and twice the sum of the *last* ordinate and of the even ones, and multiply the sum thus obtained by one-third the common interval between ordinates; in this case Area $= \frac{1}{3}h[y_0 + 4(y_1 + y_3 + y_5 \ldots) + 2(y_n + y_2 + y_4 \ldots)]$.

Tchibyscheff's Rule, based on similar assumptions to Simpson's Rule, employs fewer figures in its application and employs less ordinates to obtain a slightly more accurate result. The ordinates are not spaced equidistant, but in accordance with Table 1; having selected the number of ordinates to be used, this table gives the fractions of the half length of base at which they must be spaced, starting always from the center line of figure. The ordinates are then measured off and summed, the sum thus obtained being divided by the number of ordinates. The mean ordinate thus obtained is multiplied by the length of base to obtain the area, thus:

$$\frac{\text{Sum of ordinates}}{\text{No. of ordinates}} \times \text{Length of base} = \text{Area}$$

Table 1. Tchibyscheff's Ordinate Table

No. of ordinates	Distance of ordinates from middle of base in fractions of half the base length
2	0.5773
3	& 0.7071
4	0.1876, 0.7947
5	& 0.3745, 0.8325
6	0.2666, 0.4225, 0.8662
7	& 0.3239, 0.5297, 0.8839
9	& 0.1679, 0.5288, 0.6010, 0.9116
10	0.0838, 0.3127, 0.5000, 0.6873, 0.9162

MECHANICS OF RIGID BODIES

BY

M. W. TORBET

Mechanics is the **science** of **motion** and **force.**

Motion is **Change of Position.** Change of position is a distance and can be represented by a length L.

Velocity is **Rate of Change of Position.** When a change of position L takes place in a time T, the velocity is $L \div T$; example, miles per hour, feet per second, revolutions per minute.

Acceleration is **Rate of Change of Velocity.** When a change of velocity $V_2 - V_1$ takes place in a time T, the acceleration is

$(V_2 - V_1) \div T$ or $L \div T^2$; example feet per second per second.

Mass is **Quantity of Matter.** The mass of a body is the quantity of matter contained in the body.

Weight is the **attraction of the earth upon mass**; it varies with latitude and height above sea level; example, pound, kilogram, weight is mass $\times$ gravity, hence $W = Mg$ and $M = \frac{W}{g}$.

Momentum is **Quantity of Motion.** The momentum of a body in motion is the **mass of the body times its velocity**; hence $MV = ML \div T$.

Force is the Physical Quantity which Produces a Change in Momentum. The change of momentum is equal to the force acting, hence **force** is **mass times acceleration** or $F = M(V_2 - V_1) \div T = ML \div T^2$.

Work is **force times the distance through which the force acts,** hence $W = F \times L = ML^2 \div T^2$; example foot-pounds, ergs.

Energy is the Power to do Work. If a body gains or loses energy, work must have been done upon it or by it.

Kinetic energy is the energy possessed by a **body in motion**; example, a falling body, a projectile.

Potential energy is the energy possessed by a **body at rest**; example, a suspended weight, a compressed spring.

Power is the Rate of Doing Work. If a certain amount of work W is done in a time T, the power is $W \div T$ or $ML^2 \div T^3$.

UNITS

The Fundamental Units of Mechanics are Mass, Length and **Time.** The units of all other mechanical conceptions or quantities are derived from these fundamental units as shown in Table 1. All new formulæ should be checked to see that the dimensions of the result are consistent with the result desired. Dimensions used must belong to the same system, either **foot-pound, second** or **centimeter, gram, second** and **"absolute"** or **"gravitational."**

Table 1. Absolute Systems

Names of quantities	Dimensional formulæ	Names of units	
		Foot-pound-second	Centimeter-gram-second
Length	L	Foot	Centimeter
Mass	M	Pound	Gram
Time	T	Second	Second
Area	L^2	Square foot	Square centimeter
Volume	L^3	Cubic foot	Cubic centimeter
Moment of area	L^3	Ft.^{3}	Cm.^{3}
Moment of inertia (area)	L^4	Ft.^{4}	Cm.^{4}
Velocity	$L \div T$	Feet per second	Centimeters per second
Momentum	$ML \div T$	.....	"Bole" (proposed unit)
Acceleration	$L \div T^2$	Feet per second per second	Centimeters per second per second
Force	$ML \div T^2$	Poundal	Dyne.
Weight	$ML \div T^2$	Poundal	Dyne.
Moment of mass	ML	Pound-foot	Gram-centimeter
Moment of inertia (body)	ML^2		
Work	$ML^2 \div T^2$	Foot poundal	Erg
Energy	$ML^2 \div T^2$	Foot poundal	Erg
Power	$ML^2 \div T^3$	Foot poundals per second	Ergs per second

In **engineering calculations** generally, the **unit of force is taken as the weight of a pound** and the **fundamental units are length, force and time.** Their interrelations are shown in Table 2.

Table 2. Engineering or Gravitation Systems

Names of quantities	Dimensional formulæ	Names of units	
		Foot-pound (force)-second	Meter-kilogram-(force) second
Length	L	Foot	Meter
Force	F	Pound	Kilogram
Time	T	Second	Second
Area	L^2	Square foot	Square meter
Volume	L^3	Cubic foot	Cubic meter
Moment of area	L^3	Ft.^{3}	M.^{3}
Moment of inertia (area)	L^4	Ft.^{4}	M.^{4}
Velocity	$L \div T$	Feet per second	Meters per second
Acceleration	$L \div T^2$	Feet per second per second	Meters per second per second
Momentum	$F \times T$		
Work	$F \times L$	Foot-pound (ft.-lb.)	Kilogram-meter (kg.-m.)
Energy	$F \times L$	Foot-pound (ft.-lb.)	Kilogram-meter (kg.-m.)
Power	$FL \div T$	Foot-pounds per second (ft.-lb./sec.)	Kilogram meters per second (kg. m./sec.)
Weight	F	Pound (lb.)	Kilogram (kg.)
Mass	$FT^2 \div L$		
Moment of mass	FT^2		
Moment of inertia (body)	FLT^2		

Divisions of Mechanics

If only **length** and **time** are considered, **mechanics** is limited to the **geometry** of **motion** or **kinematics**; if only **length** and **mass** are considered, it is limited to **forces** at **rest** or **statics**; while if **mass, length** and **time** are considered, it deals with **bodies** and **forces in motion** or **kinetics.**

KINEMATICS

Motion in a Straight Line—Rectilinear Motion

Uniform Motion. If a point is moving in a straight line its position with reference to a fixed point in that line may be designated by a distance S. If equal distances ds are described in equal times dt, the velocity is said to be uniform. The rate of change of position or **Velocity** of the point is $v = ds/dt$; hence $s = \int v dt$. For uniform motion $V = S + T, S = V \times T, T = S \div V$.

Accelerated Motion. If the velocity of the moving point changes, the motion is said to be accelerated. The **acceleration** is the rate of change of the velocity, hence $a = \dfrac{dv}{dt} = \dfrac{d^2s}{dt^2}$.

If the velocity increases the acceleration is $+$, if it decreases, the acceleration is $-$.

Uniformly Accelerated or Retarded Motion. If the velocity varies but the **acceleration** is **constant,** the motion is said to be **uniformly accelerated** or **retarded.** The acceleration being constant,

$a = ds^2/dt^2, v = \dfrac{ds}{dt} = at + v_o, s = s_o + V_o t + \frac{1}{2} at^2$ and $\frac{1}{2}(v^2 - v_o^2) = a(s - s_o)$. Or

$$V = V_o + AT, \tfrac{1}{2}V^2 = AS, S = V_oT + \tfrac{1}{2} AT^2.$$

If the point starts from rest $V_o = 0$.

Simple Harmonic Motion. If a point moves in a circle with a constant velocity, its projection upon a straight line in the plane of the circle has a **Simple Harmonic Motion.** In Fig. 1 the line Op revolves about O with a constant angular velocity ω, q is the projection of p on the line AB, and the motion of q is Simple Harmonic.

If $O'Op$ is the angle ωt, the distance s of q from O' is $s = r \sin \omega t$; the velocity $v = ds/dt = \omega r \cos \omega t$ and the acceleration $a = dv/dt = ds/dt^2 = -\omega^2 r \sin \omega t = -\omega^2 s$.

The acceleration is always directed toward O' and is proportional to the distance s and the square of the angular velocity ω. When $\omega t = 2\pi$, $4\pi \ldots$ etc., s, v and a have the same values, hence the period or time between equal values is $t = 2\pi/\omega$.

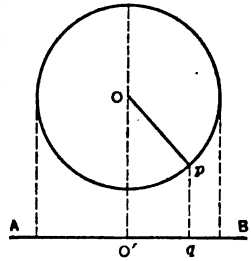

FIG. 1.

Composition and Resolution of Velocities. Since a velocity has direction and magnitude, it can be represented by a straight line; the direction of the line representing the direction of the motion and its length the velocity. Such a line is called a vector. Two velocities may be combined by means

of the **parallelogram of motion** as in Fig. 2, where V is the geometric sum or **resultant** of the velocities v_1 and v_2.

More than two velocities in the same plane may be combined by repeating the parallelogram and forming a polygon of forces, as in Fig. 3.

In such a polygon the lines representing velocities should be drawn with the arrows following each other, the closing side with its **direction reversed** will then be the resultant.

Velocities may be resolved into components in the same plane in like manner. The velocity to be resolved into components is considered a resultant, and if

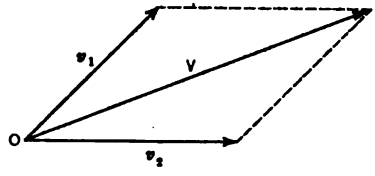

FIG. 2.

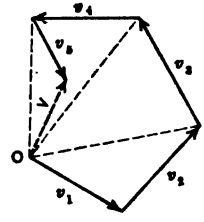

FIG. 3.

the directions of the components are known their magnitudes are obtained from the parallelogram. If the direction and magnitude of one component is known, the direction and magnitude of the other component are obtained from the parallelogram as before.

Velocities in different planes may be combined or resolved by means of the **parallelopiped of motion**, Fig. 4, the resultant being the diagonal of the parallelopiped, and the components being the three sides meeting at o.

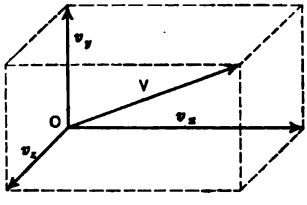

FIG. 4.

Composition and Resolution of Accelerations. Accelerations, having magnitude and direction, can be represented by vectors and combined and resolved by parallelograms and polygons in the same manner as velocities.

Motion in a Curve—Curvilinear Motion

If a point is moving in a curve its direction is tangent to the curved path and its linear velocity is $v = ds/dt$ as in rectilinear motion. Since successive tangents are not in the same direction, the acceleration is not in the same direction as the velocity. If at any point in the path the acceleration be resolved into components normal and tangent to the path and r and v be the radius of curvature of the path and the velocity tangent to the path at that point, then $a_n = v^2/r$ and $a_t = dv/dt$.

$$a = \sqrt{a_n^2 + a_t^2} = \sqrt{\frac{v^4}{r^2} + \left(\frac{dv}{dt}\right)^2}$$

If the curved path be a circle of radius r, the linear velocity being v and the angular velocity ω, then $v = \omega r$, $a_t = r\dfrac{d\omega}{dt}$ and $a_n = \dfrac{v^2}{r} = r\omega^2$.

If equal circular paths are traveled in equal periods of time, v is constant and $d\omega/dt = o$, $a_t = r\dfrac{d\omega}{dt} = o$ and $a_n = v^2/r = r\omega^2$. The angular velocity $\omega = 2\pi N$, N being revolutions per minute or per second as the case may be.

Motion in a Plane

While motion of a point in a plane may take place in any manner the motion of a body (plane figure) in a plane necessitates that the distances between the particles of which it is composed do not vary, that is to say, the body must keep its shape during the motion. Such a body is a **rigid body** and its motion is determined by two of its points.

If a body moves so that every one of its points move in a straight line, its motion is called **translation**; if every one of its points move in circles whose centres lie in a straight line perpendicular to the planes of the circles, the motion is called **rotation**.

The motion of any body in a plane can be resolved into a translation and a rotation. In Fig. 5, a body represented by two points A and B may be moved from the position AB to the position $A'B'$ by a translation through a distance s and a rotation through an angle θ.

A body may be moved in a plane from any initial position to any other position by a single rotation about a certain point called the centre of the motion. In Fig. 6, the body represented by two points A and B may be moved from the position AB to the position $A'B'$ by rotation about the centre O.

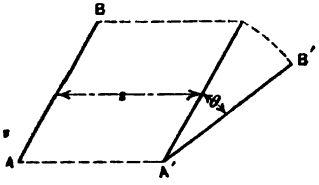

FIG. 5.

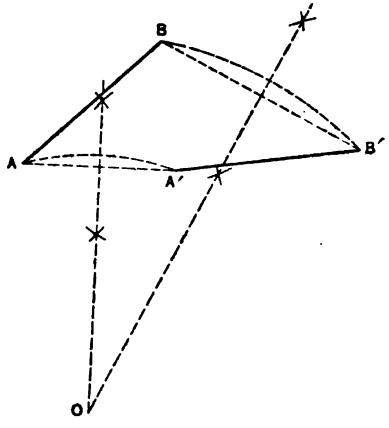

FIG. 6.

When a body is subjected to a combined translation and rotation it may be considered at any instant as rotating about the centre of the motion at that instant. Such a centre of motion is called an **instantaneous centre.**

For motion of a body in a plane the linear velocity and acceleration are the same as for motion in a straight line, $v = ds/dt$ and $a = dv/dt = \dfrac{d^2s}{dt^2}$, and the angular velocity and acceleration are the same as for motion in a circle, $\omega = d\theta/dt$ and $\alpha = d\omega/dt = d^2\theta/dt^2$.

The linear velocity at any instant is the angular velocity times the distance from the instantaneous centre.

Motion in Space

In manner similar to Fig. 6 it may be shown that a body may be moved from one position to another in space by a single rotation and a single translation.

The axis about which, at any instant, rotation takes place, is called the instantaneous axis.

STATICS

By definition **Force** is equal to mass times acceleration. If a particle has two equal and opposite accelerations its motion will not be changed. Therefore, the same effect follows if a particle is acted upon by two equal and opposite forces $F = ma$ and $F' = -ma$; the combined effect of such forces is zero and the particle, if at rest, remains at rest, and if moving with constant velocity in a straight line will continue to do so. The subject of **statics** considers only **forces on bodies at rest.**

Equilibrium. If the forces acting on a body do not produce any acceleration, they neutralize each other, that is to say, forces which produce no acceleration form a **system in equilibrium.** The body acted upon may be at rest or in motion, but if in motion the velocity must be constant. If a body tends to return to its original position after having been moved a small amount, it is said to be in stable equilibrium, if it tends to move further from its original position it is said to be in unstable equilibrium, and if it tends to do neither, it is said to be in neutral equilibrium.

External and Internal Forces. The force with which a body resists forces which tend to *lengthen* it is called **tension.** The force with which a body resists forces which tend to *shorten* it is called **compression.** The force with which a body resists forces which tend to *slide one part of the body on the other part* is called **shear.**

Composition and Resolution of Forces. Since a force has direction and magnitude it may be represented by a line or vector in manner similar to a velocity or acceleration, and if two or more forces act on a single point of a body, their resultant single force may be found from a parallelogram or polygon of forces. A resultant may be resolved into components in the same manner as velocities or accelerations. For **equilibrium** the resultant must be zero.

Couples and Moments. Two parallel, equal and opposite forces, $+F$ and $-F$, form a **couple,** whose **moment** is Fp, p being the perpendicular distance between the forces, Fig. 7. Such a pair of forces tends to produce a rotation and cannot be reduced to a single force. A couple is not affected by rotation or translation in the same plane. Couples in the same or parallel planes may be added algebraically. The direction or sense of a couple is + when it tends to produce rotation in a right hand or clockwise direction and − if the reverse is true.

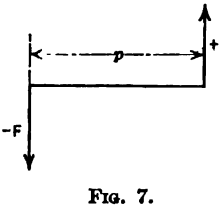

Fig. 7.

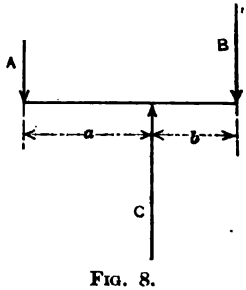

Fig. 8.

Parallel Forces—Principle of Levers. The resultant of two or more parallel forces is equal to the algebraic sum of the forces and acts through a point about which the sum of the moments of the forces is zero. In Fig. 8,

C is the resultant of A and B if $C = A + B$ and $Aa = -Bb$, $Aa + Bb = 0$. This is known as the principle of levers and may be used to solve all problems in connection with them.

General Conditions for Equilibrium. For equilibrium of a rigid body the resultant of all the forces acting upon it must be zero and the sum of all the moments of those forces about *any point* must also be zero. $\Sigma X = 0$, $\Sigma Y = 0$, $\Sigma Z = 0$, $\Sigma M = 0$.

Action of Couples on Beams—Bending Moment—Torque. The application to a beam of forces forming a couple tends to distort the beam. If the couple acts in the same plane as the axis of the beam, it tends to produce

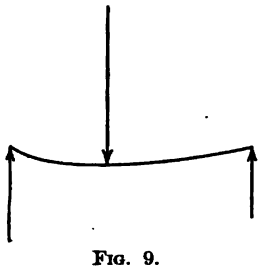

FIG. 9.

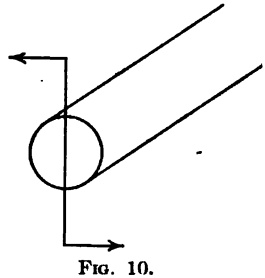

FIG. 10.

bending, Fig. 9, and is known as **bending moment,** while if it acts in a plane perpendicular to the axis it produces a twist, Fig. 10, and is known as twisting moment or **torque.**

Centres of Gravity

The action of gravity on a body produces a system of parallel forces acting on the elements of mass of the body; the resultant of these forces passes through a point such that the sum of the moments of the forces about that point is equal to zero. This point is known as the **centre of gravity.** The distances of the centre of gravity of a body from coördinate axes X, Y and Z are represented by $\bar{X}$, $\bar{Y}$ and $\bar{Z}$, and may be found by dividing the body into elements and adding the products of the elements and their respective distances from the axis, then

$$M\bar{x} = x_1m_1 + x_2m_2 + x_3m_3 \ldots, \text{ whence } \bar{x} = \frac{\Sigma mx}{M}$$

$$M\bar{y} = y_1m_1 + y_2m_2 + y_3m_3 \ldots, \text{ whence } \bar{y} = \frac{\Sigma MY}{M} \text{ and}$$

$$M\bar{z} = z_1m_1 + z_2m_2 + z_3m_3 \ldots, \text{ whence } \bar{Z} = \frac{\Sigma MZ}{M}, \text{ or}$$

$$\bar{X} = \frac{\int xdM}{\int dM}, \quad \bar{Y} = \frac{\int ydM}{\int dM}, \quad \bar{Z} = \frac{\int zdM}{\int dM}.$$

Straight Line. Centre of gravity is at the middle point of the line.

Arc of Circle, Case 1, Fig. 11, $\bar{x} = r \sin c/c$, $\bar{y} = 2r \sin^2 \frac{1}{2} c/c$; Case 2, Fig. 12, $\bar{x} = r \sin c/c$, $\bar{y} = 0$.

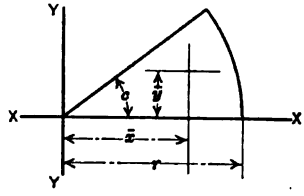

FIG. 11.

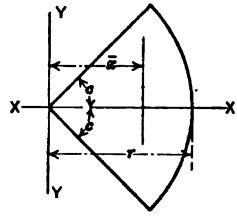

FIG. 12.

Triangle. Centre of gravity is at intersection of lines joining middle points of the sides and vertices, Fig. 13.

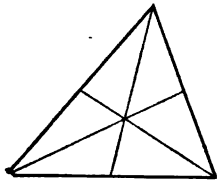

FIG. 13.

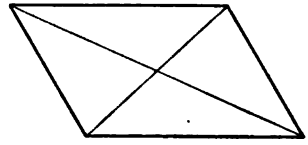

FIG. 14.

Parallelogram. Centre of gravity is at intersection of diagonals, Fig. 14.

Trapezoid. Centre of gravity lies on the line joining middle points of the parallel sides and at distances $y_a = h(a + 2b) \div 3(a + b)$ and $y_b = h(2a + b) \div 3(a + b)$ from the parallel sides a and b. The position of the centre of gravity may be obtained graphically as in Fig. 15.

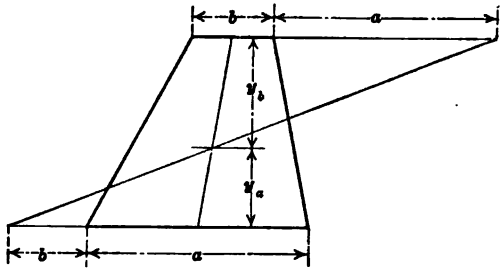

FIG. 15.

Any Quadrilateral may be divided into triangles whose centres of gravity are known and the centre of gravity of the quadrilateral found by adding

the products of the areas of the triangles and the distances of their centre of gravity from the axis and dividing the sum of the products by the area of the figure.

Segment of Circle. Fig. 16,

$\bar{x} = \frac{3}{4}r \sin^3 c \div (c - \sin c \cos c)$.

Sector of Circle. Fig. 17,

$\bar{x} = \frac{2}{3}r \sin c \div c, \quad \bar{y} = \frac{1}{2}r \sin^2 \frac{1}{2}c.$

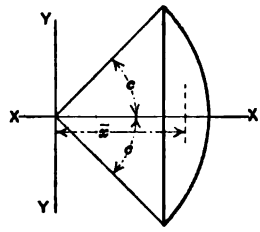

FIG. 16.

Any plane figure may be divided into elements perpendicular to the axis, adding the moments of these elements about the axis and dividing the sum of the moments by the area of the figure, Fig. 18.

A modification of "Simpson's Rules" may be applied in finding the centres of gravity of figures bounded by smooth curves, such as the lines of a ship. Equidistant ordinates are drawn and the end ordinate taken as the axis of moments. The ordinates are then multiplied by Simpson's multipliers and the products by the number of intervals from the axis as in the following table:

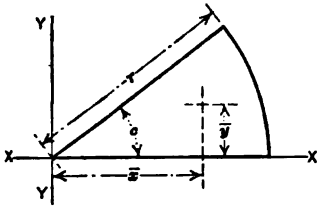

FIG. 17.

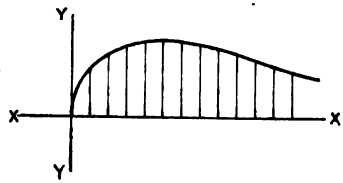

FIG. 18.

Interval or lever	Ordinate	Simpson's multiplier	Products for area	Products for moment
0	y_0	1	$1 \times y_0$	0
1	y_1	4	$4 \times y_1$	$1 \times 4 \times y_1$
2	y_2	2	$2 \times y_2$	$2 \times 2 \times y_2$
3	y_3	4	$4 \times y_3$	$3 \times 4 \times y_3$
4	y_4	2	$2 \times y_4$	$4 \times 2 \times y_4$
5	y_5	4	$4 \times y_5$	$5 \times 4 \times y_5$
6	y_6	1	$1 \times y_6$	$6 \times 1 \times y_6$
			Sum $= \Sigma A$	Sum $= \Sigma M$

Area $= \frac{1}{3}h \times \Sigma A$, moment $= \frac{1}{3}h^2 \Sigma M$

$\bar{x} = \frac{1}{3}h^2 \Sigma M \div \frac{1}{3}h \Sigma A = h \Sigma M \div \Sigma A$

Extensions of this method to computing centres of gravity of solids and centres of buoyance will be found in text-books and hand books on Naval Architecture.

In symmetrical figures or solids the centre of gravity lies on the axis of symmetry.

Prism or Cylinder with Parallel Bases. Centre of gravity lies midway between the bases on the line joining the centres of gravity of the bases.

Pyramid or Cone. Centre of gravity lies one-fourth the altitude above the base on the line joining the vertex with the centre of gravity of the base.

Determination of Centre of Gravity by Experiment. Suspend successively the body from two or more points and plumb down from each point of support. The centre of gravity lies at the intersection of the plumb lines. Place the body on knife edges on platform scales, Fig. 19, then the weight of the body is $w_1 + w_2$ and taking moments about one end, $W_2A = Wx$ and $\bar{x} = W_2A/W$.

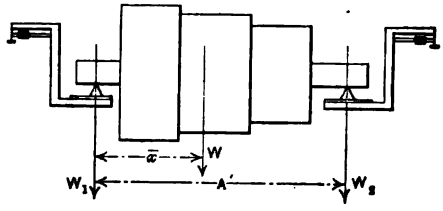

FIG. 19.

Moment of Inertia

If in a rigid body the mass m of each particle be multiplied by the square of its distance r from a given point, line, or plane, the sum of such products as mr^2 or $\Sigma mr^2 = m_1r_1^2 + m_2r_2^2 + m_3r_3^2 \ . \ . \ .$ computed for the entire body, is called the quadratic moment or **moment of inertia,** usually designated by I. If $\Sigma mr^2 = I = Mr_0^2$, the quantity r_0 is called the radius of inertia or **radius of gyration.**

Plane figures may be taken as infinitely thin solids and area substituted for mass, hence $I = \Sigma ar^2 = Ar_0^2$. The moment of inertia of a body composed of a number of parts about an axis is the sum of the moments of inertia of the several parts about that axis, $I = I_1 + I_2 + I_3 \ . \ . \ .$

Moments of Inertia about Parallel Axes. The moment of inertia of a body for any line or plane is the sum of the moment of inertia of the body for a parallel line or plane and the mass times the square of the distance between the two lines or planes, $I = I_0 + Md^2$.

Polar Moment of Inertia. The moment of inertia of a body about a point is called its polar moment of inertia about that point. The polar moment of inertia is equal to the sum of the moments of inertia about any two axes at right angles to each other and intersecting at the pole.

Moments of Inertia of Areas

Rectangle about one side, Fig. 20.

$$I = \frac{bh^3}{3}.$$

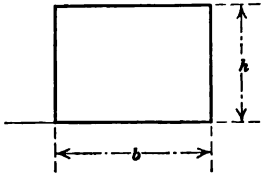

FIG. 20.

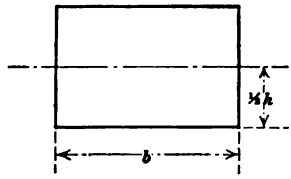

FIG. 21.

Rectangle, about centre of gravity, Fig. 21.

$$I = \frac{bh^3}{3} - bh\left(\frac{h}{2}\right)^2 = \frac{bh^3}{12}.$$

Triangle, about base, Fig. 22.

$$I = \frac{bh^3}{12}$$

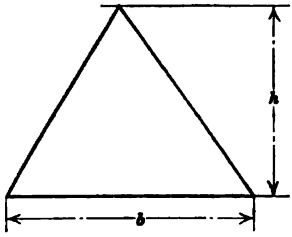

FIG. 22.

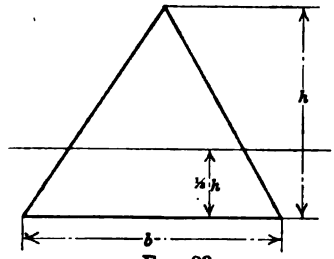

FIG. 23.

Triangle, about centre of gravity, Fig. 23.

$$I = \frac{bh^3}{12} - \frac{bh}{2}\left(\frac{h}{3}\right)^2 = \frac{bh^3}{36}.$$

Circle, about diameter, Fig. 24.

$$I = \frac{\pi d^4}{64} = \frac{\pi r^4}{4}.$$

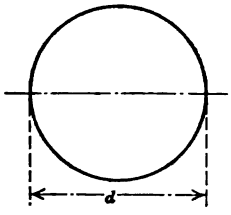

FIG. 24.

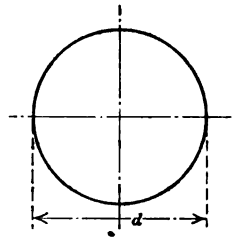

FIG. 25.

Circle, about centre (polar moment of inertia) Fig. 25.

$$J = \frac{\pi d^4}{32} = \frac{\pi r^4}{2}.$$

A modification of "Simpson's Rules" may be applied in finding the moments of inertia of figures bounded by smooth curves; equidistant ordinates are taken and computations made as in the following tables.

Moment of Inertia about Base

No. of interval	Ordinate	Ordinate^3 ÷ 3	Multiplier	Products
0	y_0	$\frac{1}{3}y_0^3$	1	$1 \times \frac{1}{3}y_0^3$
1	y_1	$\frac{1}{3}y_1^3$	4	$4 \times \frac{1}{3}y_1^3$
2	y_2	$\frac{1}{3}y_2^3$	2	$2 \times \frac{1}{3}y_2^3$
3	y_3	$\frac{1}{3}y_3^3$	4	$4 \times \frac{1}{3}y_3^3$
4	y_4	$\frac{1}{3}y_4^3$	2	$2 \times \frac{1}{3}y_4^3$
5	y_5	$\frac{1}{3}y_5^3$	4	$4 \times \frac{1}{3}y_5^3$
6	y_6	$\frac{1}{3}y_6^3$	1	$1 \times \frac{1}{3}y_6^3$
				Σ products

I = Σ products × ⅓ interval.

Moment of Inertia about End Ordinate

No. of Interval	Ordinate	Multiplier	Product	Square of No. interval	Products
0	y_0	1	y_0	0	0
1	y_1	4	$4y_1$	1	$1 \times 4y_1$
2	y_2	2	$2y_2$	4	$4 \times 2y_2$
3	y_3	4	$4y_3$	9	$9 \times 4y_3$
4	y_4	2	$2y_4$	16	$16 \times 2y_4$
5	y_5	4	$4y_5$	25	$25 \times 4y_5$
6	y_6	1	y_6	36	$36 \times y_6$

$I = \Sigma$ products $\times \frac{(\text{Int.})^3}{3}$. Σ products.

KINETICS

Newton's Laws of Motion. I. Every body persists in its state of rest or of uniform motion in a straight line, unless it is compelled by external forces to change that state. II. Change of momentum is proportional to the external moving force and takes place along the line in which that force acts. III. To every action there is an equal and contrary reaction.

Motion in a Straight Line—Rectilinear Motion

Uniform Motion. If a body is moving at constant velocity in a straight line, its momentum is $MV = m\frac{ds}{dt}$ and its kinetic energy is $\frac{1}{2}MV^2 = m\left(\frac{ds}{dt}\right)^2$.

Accelerated Motion. If the velocity of the moving body changes, the change of momentum for unit time, is equal to the force acting upon the body, hence $m\frac{dv}{dt} = m\frac{d^2s}{dt^2} = ma = F$, or if the force acts for a time t, $Ft = m(v_1 - v_0)$. Since $\frac{1}{2}(v_1^2 - v_0^2) = a(s_1 - s_0)$, $\frac{1}{2}m(v_1^2 - v_0^2) = ma(s_1 - s_0)_1 = F(s_1 - s_0)$, or, the change in kinetic energy is equal to the force times the distance through which it acts, hence change in kinetic energy is equal to the work done.

Simple Harmonic Motion. If as in Fig. 1, the distance of a body from a fixed point is $s = r \sin \omega t$ and its velocity $v = \omega r \cos \omega t$, its momentum is $mv = m\omega r \cos \omega t$, and the force acting upon it is $ma = -m\omega^2 r \sin \omega t = -m\omega^2 s$, that is to say, to produce harmonic motion the external force must be directed toward the centre and be proportional to the distance of the body from the centre.

Motion in a Circle—Rotation. The acceleration and therefore the force acting upon a body moving in a circle may be resolved into components tangent and normal to the circle. The component tangent to the circle is $mr\frac{d^2\theta}{dt^2}$, or since mr^2 is the moment of inertia about the centre, $F_t r = I\frac{d^2\theta}{dt^2}$ and if the angular velocity $\frac{d\theta}{dt} = \omega$, then $F_t r = I\frac{d\omega}{dt}$.

The force normal to the circle is $F_n = \frac{mv^2}{r}$ and is called the **centripetal force.** The equal and opposite force acting upon the axis is called the **centrifugal force.**

Since the tangential force F_t acts at a distance from the centre equal to the radius r it is equivalent to a couple or **Torque.** $H = F_t r$, therefore $H = I\frac{d\omega}{dt}$ or the torque is equal to the moment of inertia times the angular acceleration.

SECTION 2

NON-FERROUS AND FERROUS METALS AND HEAT

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CONTENTS

NON-FERROUS METALS AND ALLOYS

By G. H. CLAMER

	Page
Pure Metals	41
Bronzes	53
Brasses	56
Strength of Metals and Alloys at High Temperatures	68
Bearing Metals	68
White Metal Alloys	78

IRON AND STEEL

By D. J. McADAM, JR.

Ores of Iron	79
Extraction of Iron from Ore	80
Blast Furnace Products	81
Metallography	82
Chemistry of the Purification Processes	87
Wrought Iron	88
Steel	89
Ingots	97
Mechanical Treatment	98
Heat Treatment	98
Influence of Chemical Composition on Physical Properties	103
Case Carburization or Partial Cementation	109
Cast Iron	111
Ferro Alloys	119
Specifications for Wrought Iron	121
Specifications for Steel	122
Specifications for Cast Iron	148
Protective Coatings for Iron and Steel	153

SCREW THREADS, BOLTS AND NUTS

Screw Threads	158
Bolts and Nuts	161

OXY-ACETYLENE WELDING

By STUART PLUMLEY

Gases for Welding and Cutting	163
Burners	164
Expansion and Contraction	165
Welding Rods	166
Fluxes	167
Welding	168
Cutting	171

THE THERMIT WELDING PROCESS

By JOHN H. DEPPELER

	Page
Composition	172
Kinds of Thermit	172
Physical Properties	173
Large Welds	173
Crank Shaft Repairs	175
Pipe Welds	176

HEAT

By G. A. GOODENOUGH, M. E.

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Temperature Measurement	177
Expansion	178
Specific Heat	180
Freezing Mixtures	183
Melting Points of Solids	183
Freezing and Boiling Points	185
Heat of Fusion and Latent Heat	185
Transmission of Heat	186
Thermal Conductivities	189
Thermodynamics	195
Perfect Gases	200
Gas Mixtures	201
Expansion of Gases	202
Ideal Cycles with Perfect Gases	204
Air Compression	206
Vapors, Properties of	208
Steam Tables	209
Refrigerants	218
Expansion of Vapors	223
Mixture of Gases and Vapor	223
The Steam Engine	226
Refrigeration	227
Flow of Gases and Vapors	231
Throttling	237

NON-FERROUS METALS AND ALLOYS

BY

GUILLIAM HENRY CLAMER

REFERENCES: Roberts-Austen, "Introduction to the Study of Metallurgy," Lippincott. Fulton, "Principles of Metallurgy," McGraw-Hill. Hoffman, "General Metallurgy," McGraw-Hill. Law, "Alloys and Their Industrial Applications," Lippincott. Gulliver, "Metallic Alloys," Lippincott. Brannt, "The Metallic Alloys," Baird & Co.

Table 1. Physical Constants of Metals

Metal	Symbol	Atomic weight	Specific gravity	Specific heat	Melting point, deg. fahr.	10,000 × coefficient of linear expansion per deg. fahr.	Electrical conductivity (copper = 100)
Aluminum	Al	27.1	2.56	0.218	1216	0.1280	60.5
Antimony	Sb	120.2	6.71	0.051	1166	0.0583	4.4
Arsenic	As	75.0	5.67	0.081	1472	0.0306	.....
Barium	Ba	137.4	3.78	0.047	1562	.....	.....
Bismuth	Bi	208.0	9.80	0.031	518	0.0740	12.0
Cadmium	Cd	112.4	8.60	0.056	610	0.1700	15.6
Cæsium	Cs	132.8	1.87	0.048	79	.....	.....
Calcium	Ca	40.1	1.57	0.170	1481	.....	.....
Cerium	Ce	140.2	6.68	0.045	1152	.....	.....
Chromium	Cr	52.1	6.50	0.120	2741	.....	.....
Cobalt	Co	59.0	8.50	0.103	2714	0.0684	.....
Columbium	Cb	93.5	12.70	0.071	.....	.....	.....
Copper	Cu	63.6	8.93	0.093	1981	0.0928	100.0
Gallium	Ga	69.9	5.90	0.079	86	.....	.....
Glucinum	Gl	9.1	1.93	0.621	.....	.....	.....
Gold	Au	197.2	19.32	0.031	1945	0.0800	71.8
Indium	In	114.8	7.42	0.057	311	0.2320	.....
Iridium	Ir	193.1	22.42	0.033	4172	0.0389	.....
Iron	Fe	55.8	7.86	0.110	2768	0.0672	17.4
Lanthanum	La	139.0	6.20	0.045	1490	.....	.....
Lead	Pb	207.1	11.37	0.031	621	0.1624	7.8
Lithium	Li	6.9	0.54	0.941	367	.....	.....
Magnesium	Mg	24.3	1.74	0.250	1204	0.1495	35.8
Manganese	Mn	54.9	8.00	0.120	2237	.....	.....
Mercury	Hg	200.6	13.59	0.032	−38	0.3390	1.7
Molybdenum	Mo	96.0	8.60	0.072	4532	.....	.....
Nickel	Ni	58.7	8.80	0.108	2642	0.0706	12.6
Osmium	Os	190.9	22.48	0.031	4530	0.0361	.....
Palladium	Pd	106.7	11.50	0.059	2822	0.0651	15.4
Platinum	Pt	195.2	21.50	0.032	3191	0.0495	14.2
Potassium	K	39.1	0.86	0.170	144	0.4680	.....
Rhodium	Rh	102.9	12.10	0.058	3452	0.0472	.....
Rubidium	Rb	85.5	1.53	0.077	100	.....	.....
Ruthenium	Ru	101.7	12.26	0.061	3270	0.0534	.....
Silver	Ag	107.9	10.53	0.056	1762	0.1067	106.2
Sodium	Na	23.0	0.97	0.290	207	0.3950	.....
Strontium	Sr	87.6	2.54	.....	1472	.....	.....
Tantalum	Ta	181.5	10.80	0.036	5252	0.0439	.....
Tellurium	Te	127.5	6.25	0.049	825	0.0929	.....
Thallium	Tl	204.0	11.85	0.033	578	0.1680	.....
Thorium	Th	232.4	11.10	0.028	.....	.....	.....
Tin	Sn	119.0	7.29	0.055	450	0.1240	12.0
Titanium	Ti	48.1	3.54	0.130	3362	.....	.....
Tungsten	W	184.0	19.10	0.034	5432	.....	.....
Uranium	U	238.5	18.70	0.028	4352	.....	.....
Vanadium	V	51.0	5.50	0.125	3182	.....	.....
Yttrium	Yt	89.0	3.80	.....	.....	.....	.....
Zinc	Zn	65.4	7.15	0.094	786	0.1620	27.2
Zirconium	Zr	90.6	4.15	0.066	2700	.....	.....

By the courtesy of Mr. Lionel S. Marks much material for this section has been obtained from a similar section by the same Author in the Mechanical Engineers' Handbook, McGraw-Hill Book Co.

NON-FERROUS METALS

Copper

Copper is distinguished from all other metals by a peculiar **red color**, which is pinkish or yellowish on the fresh fracture of the pure metal, but inclines to purple in the case of copper containing cuprous oxide. The **fracture** of cast copper is hackly granular; in forged or rolled copper it is fibrous and shows a pale red silky luster. Copper crystallizes in the cubic system. **Specific gravity:** Pure crystalline copper, 8.940; electrodeposited copper, 8.914; cast copper, 8.921; rolled and hammered copper, 8.952. Ordinary commercial copper is more or less porous, and its specific gravity varies between 8.2 and 8.5.

Copper is **malleable** and **ductile.** It becomes hard by rolling and drawing, but is readily annealed. It is completely softened by being held at a temperature of 350 deg. fahr. for 72 hr., or almost instantly when raised to a temperature of 600 deg. fahr.

Mechanical Properties: E = 15,000,000 (about); tensile strength, lb. per sq. in., hammered, 38,400; drawn, 44,800; fire-box (rolled and annealed), 32,700; special stay-bolt (annealed), 38,400. Elongation, 35–38 per cent; reduction of area, 45–50 per cent. Shrinkage of castings, 0.1875 in. per ft. Compressive strength (cast), 40,000 lb. per sq. in.

There are **three standard grades** of copper on the market, namely, Lake, Electrolytic and Casting.

Lake Copper is obtained from the ores in the region principally bordering on Lake Michigan by wet concentrating methods. The concentrates are then smelted, and, after rabbling and poling, are cast directly into the form of copper ingots or bars. It is usually sold at a slight advance over electrolytic copper.

Electrolytic Copper is produced in great part from the Western United States sulphide ores, which are first matted and then blown in a Bessemer converter to eliminate most of the iron and sulphur; the metal is then cast into slabs, known as "blister" copper, which usually carry about 96 per cent of copper. The blister copper is melted in large reverberatory furnaces and cast into anodes; these are dissolved in an electrolytic bath and deposited on cathodes, which are melted in reverberatory furnaces; after refining to eliminate impurities taken up in melting, it is cast into wire bars, cakes, slabs, ingots and billets.

Casting Copper is obtained from ores which do not carry sufficient gold and silver to warrant the expense of electrolytic refining, and also from brass foundry by-products known to the trade as "copper-bearing material." It is produced by smelting and oxidizing the ore or by-products in reverberatory furnaces until the impurities have been largely eliminated. Casting copper may range from 99 to 99¾ per cent pure, depending on the raw material used and the degree of refining. It is considered inferior to electrolytic and lake copper, and sells at a slightly lower figure than electrolytic copper. **Secondary casting copper,** or that recovered from scrap copper, alloys and copper-bearing residues which occur as by-products in brass foundries and manufacturing plants where copper and copper alloys are used, runs from 98½ to 99¾ per cent pure, depending on the raw material and the degree of refining.

A. S. T. M. Specifications (adopted, 1913) for Copper Wire Bars, Cakes, Slabs, Billets, Ingots, and Ingot Bars (Abstract):

METAL CONTENTS: The copper in all shapes to have a purity of at least 99.88 per

cent, as determined by electrolytic assay, silver being counted as copper. (In the case of high-resistance Lake copper, silver and arsenic are counted as copper.)

Resistivity. Low-resistance Lake copper wire bars (ingots and ingot bars) to have a resistivity not to exceed 0.1553 (0.15694) international ohm per meter-gram at 20 deg. (0 deg.) cent. annealed. Electrolytic copper in wire bars (ingots and ingot bars) to have a resistivity not to exceed 0.15535 (0.15694) international ohm per meter-gram at 20 deg. cent. annealed. High-resistance Lake copper has a resistivity greater than 0:15694 international ohm per meter-gram at 20 deg. cent.

Wire, bars, cakes, slabs and billets to be substantially free from shrink holes, cold sets, pits, sloppy edges, concave tops and similar defects in set or casting. Five per cent variation in weight or ¼ in. variation in any dimension from refiner's published list or purchaser's specifications may be considered good delivery.

U. S. Navy Specifications (Adopted, 1917) for Ingot Copper (Abstract): There are two grades, the percentage chemical requirements of which are as follows: **Grade No. 1,** copper 99.9 (min.), bismuth and antimony, none; Arsenic and sulphur, each 0.0025 (max.). Only high grade lake copper or electro-copper, from ore of the best quality, will be accepted under this grade. **Grade No. 2,** copper, 99.75 (min.); bismuth, antimony, and sulphur, each 0.01 (max.); Arsenic, 0.03 (max.). Copper of this grade may be refined from the ore or reclaimed from scrap. Copper contents are determined by electrolytic assay, silver being counted as copper.

Marine Use.—Grade No. 1 is used in the manufacture of high grade bronzes and brasses; grade No. 2 for ordinary foundry purposes in compositions of commercial brass, cast naval brass, screw pipe fittings, and commercial rolled brass, and gun metal; phosphor bronze; and other compositions not requiring great strength.

U. S. Navy Specifications (Adopted, 1918) for Sheet Copper for Sheathing Wooden Craft require metal content of 99.5 per cent. pure copper. Sheets are furnished in standard sizes 14 in. by 48 in., hard or soft rolled.

A. S. T. M. Specifications (Adopted, 1911) for Hard-drawn Copper Wire (Abstract): Wire to be free from all surface imperfections not consistent with the best commercial practice. Necessary brazes to show at least 95 per cent of the tensile strength of unbrazed wire. Specific gravity assumed as 8.89.

Size to be expressed as the diameter of the wire in decimal parts of an inch (to three places). Permissible size variations: For wire 0.100 in. in diam. and larger, ±1 per cent; for wire less than 0.100 in. in diam., 1 mil (=0.001 in.)., Coils to be gaged at both ends and near the middle. If, two points being within limits, the third is off more than 2 per cent (for wire 0.064 and larger) or 3 per cent (for wire smaller than 0.064), coil may be rejected.

Wire to have a tensile strength and elongation at least equal to the values given in the following table, as determined on fair samples.

Table 2. Specifications for Hard-drawn Copper Wire

Diam., in.	Tensile strength, lb. per sq. in.	Elongation in 10 in., per cent.	Diam., in.	Tensile strength, lb. per sq. in.	Elongation in 60 in., per cent.	Diam., in.	Tensile strength, lb. per sq. in.	Elongation in 60 in., per cent.
0.460	49,000	3.75	0.165	62,000	1.140	0.081	65,700	0.92
0.410	51,000	3.25	0.162	62,100	1.090	0.080	65,700	0.91
0.365	52,800	2.80	0.144	63,000	1.176	0.072	65,900	0.90
0.325	54,500	2.40	0.134	63,400	1.020	0.065	66,200	0.89
0.289	56,100	2.17	0.128	63,700	1.000	0.064	66,200	0.87
0.258	57,600	1.98	0.114	64,300	1.000	0.057	66,400	0.86
0.229	59,000	1.79	0.104	64,800	0.970	0.051	66,600	0.85
.....	.....	in 60 in.	0.102	64,900	0.970	0.045	66,800	.....
0.204	60,100	1.24	0.092	65,400	0.950	0.040	67,000	.....
0.182	61,200	1.18	0.091	65,400	0.940	.....	.....	.....

Resistivity to be determined by resistance measurements on fair samples at 68 deg. fahr., and not to exceed 900.77 lb. per mile-ohm for wire from 0.460 to 0.325 in. in diam. or 910.15 lb. per mile-ohm for wire from 0.324 to 0.040 in. in diam.

U. S. Navy Specifications (Adopted, 1915) for Hard-drawn Copper Wire and Hard-drawn Flat Copper Material follow the above A. S. T. M. Specifications very closely. No brazes are allowed in the case of flat material.

Copper Castings. Sound copper castings may be made in sand or metal molds by adding about 1 per cent of boron suboxide flux (containing 0.08 to 0.10 per cent of boron suboxide) to the molten metal. Tensile strength of such castings, 24,000 lb. per sq. in.; elastic limit, 11,500 lb.; elongation, 48.5 per cent; reduction of area, 74.5 per cent; electrical conductivity, as high as 97 per cent.

U. S. Navy Specifications (Adopted, 1917) for Rolled Copper Rods, Bars, Shapes, and Plates (Abstract): Minimum pure copper contents, 99.5 per cent. Physical requirements for rods, bars, and shapes: tensile strength, lb. per sq. in. (elongation in 2 in., min.), soft, 30,000 (25 per cent); hard, up to ⅜ in., 50,000 (10 per cent); ⅜ in. to 1 in., 45,000 (12 per cent); 1 in. to 2 in., 40,000 (15 per cent); over 2 in., 35,000 (20 per cent). Physical requirements for sheets and plates; tensile strength, lb. per sq. in. (elongation in 2 in., min.), soft, 30,000 to 40,000 (25 per cent); hard, 35,000 (18 per cent). Sheets less than 0.072 in. thick need not be tested physically.

Marine Use. Copper pipe, shapes, receptacles, and general coppersmith work.

Weights and Dimensions of Brass and Copper Tubes are given on p. 1260. Weights and thicknesses of **brass and copper sheets and bars** are given in Tables 3 to 5. For properties of copper wire, see p. 43.

Table 3. Weights of Copper and Brass Sheets and Plates

B. & S. gage No.	Lb. per sq. ft.		B. & S. gage No.	Lb. per sq. ft.		B. & S. gage No.	Lb. per sq. ft.	
	Copper	Brass		Copper	Brass		Copper	Brass
0000	21.30	20.40	12	3.740	3.580	27	0.657	0.629
000	19.00	18.10	13	3.330	3.190	28	0.585	0.560
00	16.90	16.20	14	2.970	2.840	29	0.521	0.499
0	15.00	14.40	15	2.640	2.530	30	0.464	0.444
1	13.40	12.80	16	2.350	2.250	31	0.413	0.395
2	11.90	11.40	17	2.100	2.000	32	0.368	0.352
3	10.60	10.20	18	1.870	1.790	33	0.328	0.314
4	9.46	9.05	19	1.660	1.590	34	0.292	0.279
5	8.42	8.06	20	1.480	1.420	35	0.260	0.249
6	7.50	7.18	21	1.320	1.260	36	0.232	0.221
7	6.68	6.39	22	1.170	1.120	37	0.206	0.197
8	5.95	5.69	23	1.050	1.000	38	0.184	0.176
9	5.30	5.07	24	0.931	0.890	39	0.164	0.156
10	4.72	4.51	25	0.829	0.793	40	0.146	0.139
11	4.20	4.02	26	0.738	0.706	.....	.....	.....
Thickness, in.			Thickness, in.			Thickness, in.		
1/16	2.89	2.77	3/4	34.7	33.2	1 1/16	66.6	63.7
1/8	5.79	5.54	13/16	37.6	36.0	1 1/8	69.5	66.4
3/16	8.68	8.30	7/8	40.5	38.8	1 3/16	72.4	69.2
1/4	11.60	11.10	15/16	43.4	41.5	1 1/4	75.2	72.0
5/16	14.50	13.80	1	46.3	44.3	1 5/16	78.1	74.7
3/8	17.40	16.60	1 1/16	49.2	47.1	1 3/8	81.0	77.5
7/16	20.30	19.40	1 1/8	52.1	49.8	1 13/16	83.9	80.3
1/2	23.20	22.10	1 3/16	55.0	52.6	1 1/2	86.8	83.0
9/16	26.00	24.90	1 1/4	57.9	55.4	1 15/16	89.7	85.8
5/8	28.90	27.70	1 5/16	60.8	58.1	2	92.6	88.6
1 1/16	31.80	30.40	1 3/8	63.7	60.9	.....	.....	.....

Table 4. Thicknesses of Standard Copper Sheets Rolled to Weight
(American Brass Co.)

Lb. per sq. ft.	Thickness, in.	Lb. per sq. ft.	Thickness, in.	Lb. per sq. ft.	Thickness, in.	Lb. per sq. ft.	Thickness, in.	Lb. per sq. ft.	Thickness, in.	Lb. per sq. ft.	Thickness, in.
16	0.3456	9½	0.2052	6½	0.1404	3½	0.0756	1¾	0.0378	¾	0.0162
15	0.3240	9	0.1944	6	0.1296	3	0.0648	1½	0.0324	⅝	0.0135
14	0.3024	8½	0.1836	5½	0.1188	2¾	0.0594	1¼	0.0270	½	0.0108
13	0.2808	8	0.1728	5	0.1080	2½	0.0540	1⅛	0.0243	⅜	0.0081
12	0.2592	7½	0.1620	4½	0.0972	2¼	0.0486	1	0.0216	¼	0.0054
11	0.2376	7	0.1512	4	0.0864	2	0.0432	⅞	0.0189	⅛	0.0027
10	0.2160										

Table 5. Weights of Drawn Copper Bars, Standard Rectangular Sizes
(Weights in lb. per lin. ft.)

Size, in.	Lb.	Size, in.	Lb.	Size, in.	Lb.	Size, in.	Lb.
1/16 × 1/2	0.121	3/16 × 2¾	1.990	3/8 × 5	7.24	3/4 × 3½	10.10
1/16 × 5/8	0.151	3/16 × 3	2.170	3/8 × 1	1.93	3/4 × 3¾	10.90
1/16 × 3/4	0.181			3/8 × 1¼	2.41	3/4 × 4	11.60
1/16 × 7/8	0.211	1/4 × 1/2	0.482	3/8 × 1½	2.89	3/4 × 4¼	12.30
1/16 × 1	0.241	1/4 × 5/8	0.603	3/8 × 1¾	3.38	3/4 × 4½	13.00
1/16 × 1¼	0.301	1/4 × 3/4	0.724	3/8 × 2	3.86	3/4 × 4¾	13.80
1/16 × 1½	0.362	1/4 × 7/8	0.844	3/8 × 2¼	4.34	3/4 × 5	14.50
		1/4 × 1	0.965	3/8 × 2½	4.82	3/4 × 5¼	15.20
1/8 × 1/2	0.241	1/4 × 1¼	1.210	3/8 × 2¾	5.31	3/4 × 5½	15.90
1/8 × 5/8	0.301	1/4 × 1½	1.450	3/8 × 3	5.79	3/4 × 5¾	17.40
1/8 × 3/4	0.362	1/4 × 1¾	1.690	3/8 × 3¼	6.27	3/4 × 6	16.60
1/8 × 7/8	0.422	1/4 × 2	1.930	1/2 × 3½	6.75		
1/8 × 1	0.482	1/4 × 2¼	2.170	1/2 × 3¾	7.24	1 × 1	3.86
1/8 × 1¼	0.603	1/4 × 2½	2.410	1/2 × 4	7.72	1 × 1¼	4.82
1/8 × 1½	0.724	1/4 × 2¾	2.650	1/2 × 4¼	8.20	1 × 1½	5.79
1/8 × 1¾	0.844	1/4 × 3	2.890	1/2 × 4½	8.68	1 × 1¾	6.75
1/8 × 2	0.965			1/2 × 4¾	9.16	1 × 2	7.72
1/8 × 2¼	1.090	3/8 × 1	1.450	1/2 × 5	9.65	1 × 2¼	8.68
1/8 × 2½	1.210	3/8 × 1¼	1.810	1/2 × 5¼	10.10	1 × 2½	9.65
1/8 × 2¾	1.330	3/8 × 1½	2.170	1/2 × 5½	10.60	1 × 2¾	10.60
1/8 × 3	1.450	3/8 × 1¾	2.530	1/2 × 5¾	11.10	1 × 3	11.60
		3/8 × 2	2.890	1/2 × 6	11.60	1 × 3¼	12.50
5/16 × 1/2	0.362	3/8 × 2¼	3.260			1 × 3½	13.50
5/16 × 5/8	0.452	3/8 × 2½	3.620	3/4 × 1	2.89	1 × 3¾	14.50
5/16 × 3/4	0.543	3/8 × 2¾	3.980	3/4 × 1¼	3.62	1 × 4	15.40
5/16 × 7/8	0.633	3/8 × 3	4.340	3/4 × 1½	4.34	1 × 4¼	16.40
5/16 × 1	0.724	3/8 × 3¼	4.700	3/4 × 1¾	5.06	1 × 4½	17.40
5/16 × 1¼	0.904	3/8 × 3½	5.060	3/4 × 2	5.79	1 × 4¾	18.30
5/16 × 1½	1.090	3/8 × 3¾	5.430	3/4 × 2¼	6.51	1 × 5	19.30
5/16 × 1¾	1.270	3/8 × 4	5.790	3/4 × 2½	7.24	1 × 5¼	20.30
5/16 × 2	1.450	3/8 × 4¼	6.150	3/4 × 2¾	7.96	1 × 5½	21.20
5/16 × 2¼	1.630	3/8 × 4½	6.510	3/4 × 3	8.68	1 × 5¾	22.20
5/16 × 2½	1.810	3/8 × 4¾	6.870	3/4 × 3¼	9.41	1 × 6	23.20

Table 6. Strength of Sheet Copper at Various Temperatures
(In per cent of the strength at about + 50 deg. fahr.)

Temperature, deg. fahr.	122	212	392	482	545	693	844	1033
Tensile strength	98	95	91	85	79	75	66	51

U. S. Navy Specifications (Adopted 1917) for (1) Phosphor Copper, and for (2) Silicon Copper requires the following percentage chemical compositions:

Phosphor Copper: Phosphorus, 10 (min.); copper, 99.5 per cent pure, 90.

Silicon Copper: Silicon, 10 (min.); copper, remainder; iron, 0.5 (max.); aluminum,

0.1 (max.); tin and zinc, trace (max.). The sum of copper, silicon, and iron contents shall not be less than 99.8 per cent; analysis shall be made from each heat or furnace charge.

Marine Use. For deoxidizing foundry alloys.

Lead

Sources. Practically all the lead on the market is obtained from the **sulphide of lead** ore, known as **galena.** Practically all sulphide ores carry gold and silver, and crude lead is therefore subjected to further treatment for the extraction of the precious metals, and finally to an oxidizing refining treatment which renders the lead extremely pure.

Lead possesses a characteristic bluish-gray color and a dull metallic luster, which is lost on exposure to the air, the surface becoming dull gray. It is soft enough to be indented with the nail and can be cut with a knife, the softness of the metal increasing with its purity. It can be rolled into thin sheets, and, by previous heating, can be "squirted" into rods and tubes; it cannot, however, be drawn into fine wire on account of its want of ductility and its low tensile strength. The **specific gravity,** between 11.254 and 11.395, is only very slightly increased by rolling (according to Knab, 11.352 for cast lead and 11.358 for rolled lead). The specific gravity is lowered by the presence of other metals, so that it furnishes an indication of the softness and purity of the lead.

Lead crystallizes in the cubical system. At a bright red heat, in the presence of air, it volatilizes perceptibly, but only to a very slight extent if air be excluded; at a white heat (2900–3560 deg. fahr.) it boils. Antimony, arsenic, copper and zinc diminish the softness of lead when present to any considerable extent, antimony and arsenic rendering it brittle, hard and easily fusible.

Mechanical Properties: E in lb. per sq. in. = 700,000 for rolled or cast lead (= 1,000,000 for lead wire). Tensile strength in lb. per sq. in. = 1780 for rolled or cast lead (= 3130 for hard and 2420 for soft wire). Shrinkage of castings, 0.3125 in. per ft.

Table 7. Composition of American Pig Leads

a—Southeast Missouri Undesilverised. *d*—Ordinary Common.
b—Southeast Missouri Desilverised. *e*—Ordinary Corroding—or Refined.
c—Southwest Missouri Undesilverised.

	a	*b*	*c*	*d*	*e*
	Per cent	Per cent	Per cent	Per cent	Per cent
Silver	0.0070	0.0004	0.0005	0.0005	0.0005
Arsenic and zinc	trace	trace	trace	trace	trace
Antimony	0.0030	0.0030	0.0020	0.0100	0.0050
Bismuth	0.0030	0.0030	0.0030	0.0800	0.0500
Copper	0.0600	0.0003	0.0190	0.0006	0.0006
Iron	0.0015	0.0015	0.0015	0.0015	0.0015
Cobalt and nickel	0.0080	none	0.0018	none	none
Lead	99.9175	99.9918	99.9722	99.9074	99.9424

U. S. Navy Specifications (Adopted, 1917) for Pig Lead require for **No. 1 grade** not less than 99.9 per cent of metallic lead to be the product of new ore only; **Grade No. 2,** metallic contents not specified, may be either old or new lead.

Marine Use. Grade No. 1 pig lead is used for foundry alloys and compositions; grade No. 2, for weights, ballasts, etc.

Pennsylvania R. R. Specifications (Adopted, 1908) for Pig Lead specify that No. 1 grade shall contain not less than 99½ per cent of metallic lead, and No. 2 grade (used only for counterbalancing) not less than 97½ per cent.

Chemical Lead. The lead sold in the United States under this name is obtained in the Flat River District in southeast Missouri, from disseminated ores which are contaminated with copper, cobalt, nickel and other impurities. This lead is alloyed naturally with certain percentages of other ingredients which render it more impervious to acid attacks than other brands of lead. The Hoyt Metal Co., of St. Louis, manufacture a standardized chemical lead known as **"Hoyt process" lead**, which is furnished in the form of sheet and pipe. **Hoyt metal** is a lead carrying from 6 to 10 per cent of antimony, and in the form of sheets and pipes is extensively used in chemical works and other industries where acids are made or employed. The 10 per cent Hoyt metal weighs 8 per cent less than chemical lead, its tensile strength is double that of the latter, and it is much more rigid. It is claimed that it will not buckle, creep, stretch, tear or sag in use, and that it is much superior to chemical lead in acid-resisting qualities.

Lead Pipe. For tables of sizes and weights of lead pipe and tubing, see pp. 1260 and 1261.

Table 8. Maximum Sizes in Which Sheet Lead Can Be Furnished

Weight per sq. ft., lb.	Max. size in ft.	Weight per sq. ft., lb.	Max. size in ft.
1	8 × 20	10	11½ × 40
1½	8 × 20	12	11 × 40 or 11½ × 35
2	7 × 45	14	11½ × 40 or 11¾ × 30
2½	9 × 45	16	11½ × 40 or 11¾ × 30
3	10 × 45	20	11½ × 40 or 11¾ × 38
3½	10 × 45	20	11¾ × 36
4	10 × 45	24	11¾ × 30
5	10 × 43	24	11 × 34 or 11 ½ × 32
6	10 × 43 or 11 × 40	30	11 × 27 or 11½ × 25½
6	11½ × 30 or 11¾ × 25	30	11½ × 24½ or 12 × 16
8	10 × 40 or 11½ × 35	40	11 × 24 or 12 × 16
10	11½ × 30 or 11 × 40 or 10 × 48	60	12 × 12

Zinc

Zinc is a bluish-white metal having a **specific gravity** of about 7.1. It boils at about 930 deg. cent., so that it can be readily distilled; there is always a sensible loss when it is used in the manufacture of alloys. The metal in fine shavings or vapor burns readily with an intense bluish-white flame, forming dense clouds of white zinc oxide (philosophers' wool). It is malleable and ductile through a limited range of temperature only. It oxidizes only slightly on exposure to the air, with the formation of a basic carbonate.

Zinc is marketed in the form of **rolled sheets**, and also in cast cakes of about 1 in. thick, known as **spelter**. The cakes are very brittle, and break with a more or less crystalline fracture. If the metal be nearly pure the crystal faces are large, bright and smooth; if there be a small quantity of iron present dull spots appear on the crystal faces; with an increased quantity of iron, as in dross spelter, the **fracture** becomes granular. The amount of iron present can be fairly judged from the appearance of the fracture. Its **tensile strength** is low: 27,000 lb. per sq. in. along the grain; 36,000 across grain, for rolled sheets; spelter, 4000 (coarsely crystalline) to 14,000 (fine-grained). Modulus of rupture, 8000 to 22,000 lb. per sq. in., increasing with fineness of grain. Compressive strength (cast), 20,000 lb. per sq. in. E (average) 13,700,000; elongation, 12 to 38 per cent; reduction of area, 23 to 56 per cent.

Zinc casts well, contracts but little on solidifying, and is largely used for the manufacture of statuettes and other ornamental castings which are usually

coated with bronze or brass by electrodeposition. Shrinkage of castings, 0.3125 in. per ft. Its **chief uses** are in galvanizing and in the manufacture of brass and other alloys.

Zinc is never pure, the principal **impurities** being iron, lead, tin, copper, arsenic, and cadmium. When zinc is used for galvanizing, a hard zinc which contains several per cent. of iron accumulates in the vats. Good commercial spelter should not contain more than 0.05 per cent of iron, and this is about the maximum allowable for alloy making.

Lead is invariably present in spelter in larger or smaller quantity. The other impurities are rarely present in objectionable quantities.

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Table 9. Typical Analyses of Various Grades of Virgin Spelter

Grade	Iron %	Lead %	Cadmium %	Zinc %
High grade	0.02(0.03)	0.05(0.07)	(0.05)	99.930
Intermediate	0.025(0.03)	0.15(0.20)	(0.50)	99.825
Brass special	0.025(0.04)	0.30–0.60(0.75)	0–0.50(0.75)	balance
Prime Western	0.08(0.08)	1.50(1.50)	0.50	97.920
Sheet zinc[1]	0.015	0.27	0.29	99.400

[1]Matthiessen & Hegeler Zinc Co.

High-grade spelter is used mainly for brass for spinning and drawing; for the highest grades of alloys, such as manganese bronze, and for galvanizing telegraph and telephone wires that have to withstand sharp bending; also for artistic castings. The **intermediate** grade is used in brasses and bronzes where the very highest quality is not essential, and also for the better grades of casting-brass. The **brass special** grade is used for the better grades of alloys where high ductility is not required. **Prime Western** spelter is used mainly for the ordinary galvanizing of sheet, wire and miscellaneous articles, also in red-brass alloys and free-cutting yellow brass.

Secondary Spelter is recovered by refining zinc dross—an iron-zinc alloy formed in the process of galvanizing. It carries from 3 to 5 per cent of iron and from 2 to 3 per cent of lead. This is refined either by distillation or by a liquating method. If refined by the former method a product of fairly high purity results, and if refined by the latter method a very inferior quality results, carrying a high percentage of iron and lead. This latter product is given a marketable appearance by the addition of a very small percentage of aluminum.

A. S. T. M. Specifications (Adopted, 1911) for Spelter. Limiting percentages of impurities are specified for the four grades of virgin spelter (see values in parentheses in Table 9), methods of sampling, analysing, etc. High, intermediate and brass special grades are to be free from aluminum, and their combined contents of iron, lead and cadmium are not to exceed 0.10, 0.5, and 1.20 per cent, respectively.

U. S. Navy Specifications (Adopted, 1918) for Spelter designate three grades: (1) **Grade A, virgin spelter,** that is, spelter made from *ore* by a process of direct reduction and distillation, electrolysis, or a combination of the above; (2) **Grade B, high-grade spelter** may include refined zinc, from short ends, etc., but not scrap metal; **Grade C,** which may include metal produced by refining by-products resulting from the manufacture of nonferrous alloys or similar metals.

The Chemical Requirements of Grades A, B, and C, conform closely to the A. S. T. M. Specifications for High grade, Intermediate, and Brass Special, respectively.

except that U. S. Navy Specifications require a min. pure zinc content of 99.85 per cent, 99.35 per cent, and 98 per cent respectively.

Marine Use. Grade A is suitable for the manufacture of manganese bronze, and other high-grade nonferrous alloys; Grade B, for the manufacture of all other nonferrous alloys; and Grade C, for galvanising purposes.

Table 10. Weight of Sheet Zinc

(Matthiessen & Hegeler Zinc Co.)

Zinc gage No.	Thickness, in.	Lb. per sq. ft.	Zinc gage No.	Thickness, in.	Lb. per sq. ft.	Zinc gage No.	Thickness, in.	Lb. per sq. ft.
3	0.006	0.22	12	0.028	1.05	21	0.080	3.00
4	0.008	0.30	13	0.032	1.20	22	0.090	3.37
5	0.010	0.37	14	0.036	1.35	23	0.100	3.75
6	0.012	0.45	15	0.040	1.50	24	0.125	4.70
7	0.014	0.52	16	0.045	1.68	25	0.250	9.40
8	0.016	0.60	17	0.050	1.87	26	0.375	14.00
9	0.018	0.67	18	0.055	2.06	27	0.500	18.75
10	0.020	0.75	19	0.060	2.25	28	1.000	37.50
11	0.024	0.90	20	0.070	2.62	.....	.....	.....

U. S. Navy Specifications (Adopted, 1917) for Rolled Zinc Plates (Abstract): Metal content, not less than 98.5 per cent pure zinc, nor more than 0.08 per cent iron, and must be thoroughly compressed by rolling to make a solid homogeneous slab with smooth surfaces free from all defects. The plates must be able to stand bending through of 45 deg. over a round surface whose diameter is 1 in. without breaks or crack at a temperature not exceeding 100 deg. fahr.

Hull Zincs. Standard size, 12 in. by 6 in. by ½ in. Other sizes for circular openings are given in the table below:

Thickness	Standard size	Extreme size	Thickness	Standard size	Extreme size
Inch	*Inches*	*Inches*	*Inch*	*Inches*	*Inches*
1/8	36 by 84	60 by 96	5/8	24 by 48	24 by 48
1/4	36 by 84	36 by 84	3/4	24 by 36	24 by 48
3/8	24 by 48	24 by 72	7/8	24 by 36	24 by 36
1/2	24 by 48	24 by 72	1	24 by 36	24 by 36

Zincs for Boilers, Salt-water Piping, etc. Standard size, 12 in. by 6 in. by ½ in., with one central hole ¾ in. in diam. Tolerance of 10 per cent over or under weight will be permitted, the weight of 1 cu. in. of rolled sheet zinc being taken as 0.2605 lb.

Tin

Sources. The principal **tin ore** is cassiterite, or tin stone—an oxide of tin. The lowest grades of pig tin come from China and Bolivia, and contain as little as 94 per cent metallic tin. **Banca tin** has the highest reputation of any on the market, and next to this is Straits tin, from Malacca. Tin is now recovered from tin-plate clippings by an electrolytic or "detinning" process in the form of a fine powder, which is then smelted in a reverberatory furnace and cast into pigs. This **by-product tin** usually runs from 97 to 99 per cent of tin and carries lead, copper and antimony. For the production of the finest grades of solder, Babbitt metal and high-grade alloys subject to physical tests, Banca, Straits or similar grades of tin should be used. For all other uses the lower grades of tin are suitable, due consideration being given to their actual tin content.

Tin has an almost silvery whiteness with a slightly bluish tinge and a brilliant luster which depends largely on the pouring temperature; if this is too high the surface will show iridescent colors, while if too low the surface will be dull. The admixture of small quantities of lead, arsenic, antimony, iron, or bismuth also diminishes the luster of tin.

The structure of tin is distinctly crystalline. **Specific gravity:** Cast tin, 7.291; rolled tin, 7.299; electrically deposited tin, from 7.143 to 7.178.

Mechanical Properties. Tin is harder than lead but softer than gold. At ordinary temperatures it can be beaten and rolled into thin leaves (sheet tin and tinfoil). **At higher temperatures** its extensibility diminishes, until at 392 deg. fahr. it is so **brittle** that it breaks to pieces when hammered, and can be powdered. If the temperature at which tin is cast be either too high or too low, it will be "short," *i.e.*, brittle. The addition of 1 to 2 per cent of copper or lead increases its hardness and tenacity. Tin is **ductile** but possesses little tenacity (T.S. = 5000 to 5700 lb. per sq. in.; E = 5,700,000); it is most ductile at about 212 deg. fahr.; a wire 0.08 in. in diam. breaks under a load of 54 lb. For cast tin, tensile strength is 3500 lb. per sq. in. and compressive strength 6000 lb. per sq. in.

Impurities and Their Effects. Iron, if present in considerable quantities, makes the tin hard and brittle; arsenic, antimony and bismuth, if present to the extent of 0.5 per cent, reduce its tenacity; copper and lead (1 to 2 per cent) make it harder and increase its strength, but make it less malleable; tungsten and molybdenum render it less easily fusible; stannous oxide reduces its tenacity; sulphur is said to render it "short."

Table 11. Typical Percentage Analyses of Various Grades of Pig Tin

Grade	Tin	Antimony	Arsenic	Lead	Bismuth	Copper	Iron	Silver	Sulphur
Banca	99.95	0.007	.....	trace	.....	0.018	0.045	.....	trace
Penang	99.939	trace	0.013	trace	.....	0.016	0.028	.....	.....
Singapore	99.87	0.008	0.045	0.034	0.003	0.052	0.003	0.006	0.005
Range of impurities in other grades.	.....	0 to 0.569	0 to 0.065	0 to 4.00 (China)	0 to 0.055	0.018 to 0.445	0 to 0.028	traces	0 to 0.013

U. S. Navy Specifications (Adopted, 1917) for Ingot Tin require that the metal shall be free from scrap or remelted metal, in commercial and branded ingots of prime quality tin of the following percentage chemical composition: Pure tin, 99.75; with the following maximum impurities: Lead, antimony, arsenic, and copper, each 0.1; sulphur, 0.01. The sum of all impurities shall not exceed 0.25.

Marine Use. Tin is used in making various bronze and bearing metal alloys, for coating condenser tubes, etc., and for making safety plugs for boilers.

U. S. Navy Specifications (Adopted, 1915) for Phosphor Tin call for the following percentage chemical composition: Phosphorus 3.5 (min.); impurities, 0.5 (max.); remainder, tin of at least 99.5 per cent purity.

Marine Use. For introducing phosphorus into foundry alloys.

Antimony

Sources. Antimony glance (known also as gray antimony ore, antimonite and stibnite) is the most important ore of antimony. It is a sulphide ore (containing 71.77 per cent of antimony in the pure state), and is commonly associated with quartz, calc-spar, heavy spar, and spathic iron ore. Zinc blende and galena frequently occur intimately mixed with it.

Antimony is characterized by its great brilliancy and by its color—silver white with a slight tinge of blue, the latter being increased by the presence of impurities, such as sulphur, arsenic, lead, copper and iron.

When pure molten antimony is allowed to solidify slowly and without disturbance under a layer of slag, a fern-like arrangement of raised lines radiating from the center appears on the surface of the metal (the so-called "antimony star"). This is generally regarded as an indication of the purity of the metal.

Production. When the ores exist as pure sulphide of 90 per cent or over, they are usually roasted direct to form an oxide, which is afterward reduced by carbonaceous matter. If the ores are lean the antimony sulphide is liquated to free it of the gangue, and then roasted and reduced in the same manner. By another method the antimony sulphide is directly reduced by iron, which, having a greater affinity for sulphur than antimony, combines with the sulphur and liberates the antimony. The antimony produced by both methods must afterward be refined.

A typical analysis of commercial antimony shows the following percentage composition: Silver, 0.001; arsenic, 0.120; tin, 0.006; lead, 0.089; copper, 0.076; iron, trace; zinc, none; nickel and cobalt, 0.046; manganese, none; sulphur, 0.008; antimony (by diff.), 99.654.

Marine Use. Antimony is used in making Babbitts and other bearing metals, its presence in alloys conferring the properties of hardness and of expanding on solidification.

Nickel

Nickel possesses an almost silvery white luster with a steel-gray tinge and great brilliancy. With **great hardness** and capacity for taking polish it combines great **malleability**; it can be easily hammered, rolled, or drawn into wire. Sheets 0.0008 in. thick and wire 0.0004 in. in diameter may be made from it. Its tensile strength surpasses that of iron: for sheets of pure nickel 0.05 in. thick, rolled hard (annealed), 92,000 lb. (76,000 lb.) per sq. in. and 11 (35) per cent elongation in 2 in.; for 0.065-in. diam. hard-drawn wire, 150,000 to 160,000 lb. per sq. in.

Nickel is attracted by a magnet, and then becomes **magnetic** itself; it loses this property at 662 deg. fahr. It **can** not only be **welded** to itself at a white heat, but it can also be welded to iron and certain alloys. Fleitmann's method of manufacturing nickel-plated wares consists in welding pure nickel to iron and steel or alloys of copper and nickel, and then hammering or rolling into sheets. In welding, it is necessary to completely exclude the air from the surfaces to be welded.

Commercial nickel is exceedingly impure, containing substances of which even traces affect its valuable properties. The most harmful **impurities** are arsenic, sulphur, oxide of nickel and chlorine. Carbon is dissolved by molten nickel, and seems not to affect its good qualities so long as oxides are not present at the same time. It makes nickel slightly more fusible. According to Ledebur, nickel absorbs its own monoxide, and this injures its tenacity and malleability.

Nickel **does not tarnish** in dry air at ordinary temperatures. When heated in contact with air it is slightly oxidized into nickelo-nickelic oxide. At a red heat it decomposes steam very slightly. It is very little acted upon by hydrochloric or sulphuric acid in the cold, but is readily dissolved by dilute nitric acid and aqua regia.

Production. The chief **ores** of nickel are the silicate ores of New Caledonia, known as garnierite, and magnetic pyrites containing nickel and copper found in Ontario. The silicate ores can be treated by direct reduction, al-

though they are usually treated in connection with sulphide ores. The magnetic pyrite ores are first smelted to form matte. This matte is Bessemerized to oxidize the greater portion of the iron and sulphur, with the production of a nickel-copper matte which is afterward fused with sodium sulphate and coal, forming an easily fusible matte with the copper sulphide which is of less specific gravity than the nickel sulphide, and a fairly complete separation can be made because of the difference in the specific gravity of the two mattes. By repeating the process, practically pure nickel sulphide is produced, which is then roasted to form oxide and finally reduced to metallic nickel. Mond's method consists in roasting the Canadian mattes, eliminating the copper by sulphuric acid, and then exposing the resulting product to the reducing action of producer gas at about 660 deg. fahr. The reduced metal is then exposed to the action of CO at 176 deg. fahr. in a "volatiliser," the nickel carbonyl so formed being received in a chamber heated to about 390 deg. fahr.. where it decomposes, the nickel being deposited and the CO returned to the volatiliser.

Nickel as ordinarily reduced by carbon is brittle, but by the addition of a small percentage of magnesium or manganese, it can be made ductile.

A typical percentage composition after the addition of magnesium is as follows: Nickel, 98.24; cobalt, 1.09; iron, 0.36; copper, 0.10; silicon, 0.06; magnesium, 0.11.

Marine Use. In nickel steels, plating, in the preparation of German silver and other alloys, resistance alloy wires, anodes, etc.

Aluminum*

Aluminum (aluminium)† is a white metal closely resembling tin in color, produced electrolytically in large quantities from alumina prepared from the mineral bauxite—a hydrated oxide of the metal. Its **specific gravity** at 62 deg. fahr. ranges as follows: Pure aluminum, 2.56; pure aluminum sheets and wire unannealed, 2.68; pure aluminum sheets and wire annealed, 2.66; aluminum alloy sheets and wire unannealed or annealed (3 S), 2.75; aluminum casting alloys, 2.82 to 2.99. The alloys contain small percentages of Cu, Ni, W, Mn, Cr, Ti, Zn or Sn to produce hardness, rigidity and strength.

Ordinary commercial aluminum has about the **hardness** of copper. By pressing, rolling, forging, stamping or other similar working it may be made very hard and rigid in the finished shapes, whereas the soft annealed metal would be too weak for the purpose intended. This is especially true of aluminum alloyed with other metals in amounts not to exceed 5 or 6 per cent Greater percentages render aluminum base alloys non-malleable.

Aluminum is very **ductile.** It can be rolled into sheets as thin as 0.0007 in. and then be beaten into leaf; drawn into wire as small as 0.004 in diameter; spun, stamped or extruded in various shapes. It is susceptible of a high degree of finish by polishing or burnishing. It becomes **hardened by working,** and sheets for stamping or spinning must be annealed.

At ordinary temperatures aluminum is **highly** resistant to all inorganic **acids** except hydrochloric. Solutions of caustic alkalines, chlorine, bromine, iodine and hydrofluoric acid rapidly corrode the metal. A clean surface tarnishes in damp air, an almost invisible coating of oxide being formed, which, however, is very permanent and prevents further attack.

As the common metals are electronegative to aluminum in a voltaic couple, care should be taken that aluminum exposed to water or aqueous solutions shall not come in contact with any other metal that will cause **galvanic**

* Much of this information has been obtained from the publications of the Aluminum Co. of America.

† Trade usage in North America sanctions the use of the word aluminum.

action to be set up. For example, tubing can be thoroughly insulated by the use of rubber gaskets, washers or nuts. Aluminum sheets and shapes can be insulated from other metal by the liberal use of good heavy paint between the joints.

Commercial aluminum is sold in three **grades**: No. 1 (pure), No. 2, and extra pure. No. 1 has approximately the following composition: Silicon, 0.30 per cent; iron, 0.15 per cent; aluminum, 99.55 per cent. For No. 2 the percentages are respectively 2, 2 and 96. Silicon and iron are the only **impurities** commonly found in aluminum.

The **tensile and compressive strengths** of aluminum (99 per cent pure) average as follows:

	Tension		
	Elastic limit, lb. per sq. in.	Ultimate strength, lb. per sq. in.	Reduction of area, per cent
Castings	8,500	12,000–14,000	15
Sheet	12,500–25,000	24,000–40,000	20–30
Wire	16,000–33,000	25,000–55,000	40–60
Bars	14,000–23,000	28,000–40,000	30–40
	Compression		
Short cast columns (height = 2 × diam.).	3,500	12,000	

E for cast aluminum = 9,000,000.

Pure aluminum in castings is a metal somewhat open in texture, and when used for cylinders to stand pressure an increase in thickness over that calculated by ordinary formulæ should be made to allow for its **porosity**. Shrinkage of castings, 0.2031 in. per ft.

Under transverse tests the metal will bend nearly double before breaking. The strength of aluminum is greatly improved by forging or pressing the ingots at a temperature of about 600 deg. fahr.

U. S. Navy Specifications (Adopted, 1913) for Ingot Aluminum: Aluminum to be manufactured from bauxite or other high-grade ore, metal reclaimed from scrap not being acceptable. Percentage chemical requirements: Aluminum (min.), 99.4; iron and silicon each (max.), 0.6; other impurities (max.), 0.1. The sum of iron, silicon, and other impurities not to exceed 0.6. Ingots are to be of commercial standard form and shape, to be clean, free from adhering dirt, slag or foreign matter, and shall have normal and uniform shrinkages.

Marine Use. Suitable for the manufacture of castings and ingots for forging purposes of aluminum, aluminum bronze, manganese bronze, etc.

Extruded Aluminum can be obtained in shapes possible to manufacture no other way, and in any continuous lengths desired. It is possible to extrude shapes in compositions ranging from pure aluminum to very hard alloys, some of which cannot be rolled. The tools used are much less expensive than are the rolls necessary for rolled shapes, making it commercially possible to furnish smaller quantities of special shapes than rolling-mill costs would justify. The maximum sectional dimension of an extruded shaped should not exceed 6 in., and in general, walls under $\frac{1}{8}$ in. in thickness should not be specified.

BRASSES AND BRONZES

Copper-Tin Alloys (Bronzes)

The term **bronze** should be applied only to those alloys of copper in which tin is the predominating metal affecting or modifying the properties of the copper. A **true bronze** is an **alloy of copper and tin** only, in which copper is in predominating proportion, the most useful alloys being those containing from 4 to 25 per cent of tin. After 12 per cent of tin is exceeded, the ductility

of the alloys is so diminished that they are subject to breakage by shock. Modifications are often made by the addition of relatively small proportions of lead and zinc, and occasionally of other metals.

Table 12. Properties of Various Copper-Tin Alloys

(From Report of U. S. Test Board, 1879–1881)

Composition by analysis		Specific gravity	Color	Fracture	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Modulus of rupture, lb. per sq. in.	Elongation in 8 in., per cent	Deflection, in bar 1 in. sq., 22 in. long	Crushing strength, lb. per sq. in.	Max. torsion moment, ft.-lb.	Angle of torsion, deg.
Copper	Tin											
100.00	0.00	8.791	Copper red...	Fibrous...	27800	14000	29848	6.47	bent	42000	143	153.0
97.89	1.90	8.564	Red.....	Vesicular..	24580	10000	.....	13.33	....	34000	150	317.0
96.06	3.76	8.649	Reddish-yellow	Vesicular..	32000	16000	53232	14.29	bent	42048	157	247.0
90.27	9.58	8.669	Grayish-yellow	Earthy....	26860	15750	49400	3.66	bent	38000	175	114.0
87.15	12.73	8.681	Mottled.....	Finely vesicular	29430	20000	34531	3.33	4.00	53000	182	100.0
80.95	18.84	8.740	Reddish-gray	Finely granular	32980	.....	56715	0.04	0.49	78000	190	16.0
76.64	23.24	8.565	.....	Smooth....	22010	22010	32210	0.00	0.19	114000	122	3.4
69.94	29.88	8.932	White.....	Conchoidal	5585	5585	12076	0.00	0.06	147000	18	1.5
68.58	31.26	8.938	White.....	Conchoidal	1620	.....	9152	0.00	0.04	.....	.....	.....
65.34	34.47	8.947	Bluish-gray..	.....	2201	2201	4776	0.00	0.02	84700	16	1.0
56.70	43.17	8.682	Light gray....	Stony.....	1455	1455	2126	0.00	0.02	.....	.....	.....
44.52	55.28	8.312	Grayish-white	.....	3010	3010	4776	0.00	0.03	35800	23	1.0
34.20	65.80	8.013	Grayish-white	Crystalline	3371	3371	5384	0.00	0.04	19600	17	2.0
23.35	76.29	7.835	Grayish-white	Finely crystalline	6775	6775	12408	0.00	0.27	.....	.....	.....
15.08	84.62	7.657	Grayish-white	Crystalline	6520	.....	9063	.....	0.86	6500	23	25.0
11.49	88.47	7.552	Grayish-white	Crystalline	6380	3500	10706	4.10	5.85	10100	23	62.0
8.57	91.39	7.490	Grayish-white	Granular..	6450	3500	5305	6.87	bent	9800	23	132.0
3.72	96.31	7.360	Grayish-white	Granular..	4780	2750	6925	12.32	bent	9800	23	220.0
0.00	100.00	7.293	Grayish-white	Fibrous...	3505	.....	3740	35.51	bent	6400	12	557.0

Test pieces for compression tests were 2 in. long, ⅝ in. diam.
Test pieces for torsion were 1 in. long, ⅝ in. diam.

Gun Metal is a copper-tin alloy (8 to 11 per cent of tin) used before the introduction of steel in the manufacture ordnance. Occasionally it contains small percentages of iron, zinc and lead.

U. S. Navy Specifications (Adopted, 1919) for Gun Metal (Abstracts): Percentage chemical requirements: Copper, 88; tin, 10; zinc, 2. A variation of 1 per cent is allowed above or below. Iron not to exceed 0.06 per cent; lead not over 0.2 per cent.

Minimum physical properties: Tensile strength, 30,000 lb.; yield point, 15,000 lb.; elongation in 2 in., 15 per cent. This alloy is susceptible to quite wide variation in its physical properties, due to the temperature at which it is poured, the speed at which it cools, the subsequent heat treatment which it has had, the manner of attaching test bar, gating and provision for shrinkage. It can also be modified by slight additions of deoxidizing agents.

Marine Use. All composition valves 4 in. in diameter and above; expansion joints, flanged pipe fittings, gear wheels, bolts and nuts, miscellaneous composition castings, all parts where strength is required of composition castings or where subjected to salt water, and for all purposes where no other alloy is specified.

Composition valves, safety and relief, feed check and stop, surface blow, drain, air, and water cocks, main stop, throttle, reducing, sea, safety, sluice, and manifolds at pumps.

Heads, shapes, and water chests for condensers, distillers, feed-water heaters, and oil coolers.

Pumps. Air-pump casing, valve seats, buckets, main circulating, water cylinders, valve boxes, water pistons, stuffing boxes, followers, glands, in general the water end of pumps complete except as specified.

Stuffing box glands and bushings.

Blower bearing boxes.

Miscellaneous. Grease extractors, steam strainers, separators, casing for stern tube and propeller shafts, propeller hub caps.

Bearings. Main, stern tube, strut, and spring.

U. S. Navy Specifications (Adopted, 1916) for Bronze Valve Castings call for the following percentage chemical composition; Copper (min.), 87; tin (min.), 7; zinc, remainder; iron (max.), 0.06; lead, (max.), 1. Scrap will not be used, except such as may result from the manufacture of articles of similar composition. Color of the fracture section and grains of the metal must be uniform throughout. Castings must be sound, clean, free from blowholes, porous places, cracks, or any other defects that materially affect their strength or appearance or which indicate an inferior quality of metal.

Marine Use. This material is suitable for valves below 4 in. for steam and general purposes for which the material is not otherwise specified, manifolds and cocks, relief valves, composition lug sockets, and pad eyes not requiring special strength; hose couplings and fittings.

U. S. Navy Specifications (Adopted, 1916) for Journal Bronze require the following percentage chemical composition: Copper, 82 to 84; tin, 12.5 to 14.5; zinc, 2.5 to 4.5; iron (max.), 0.06; lead (max.), 1.0; normal proportions, 83–13.5–3.5.

Marine Use. This material is suitable for the following purposes: Bearings, journal boxes, bushings, and sleeves, slides, slippers, guide gibs, wedges on water-tight doors, and all parts subject to considerable wear, for reciprocating engines in valve stem crosshead bottom brass, link block gibs, and suspension link brasses.

Table 13. Melting Points of Various Alloys

(From Technical Paper No. 60, U. S. Bureau of Mines)

Alloy	Composition by analysis				Melting point, deg. fahr.
	Cu	Zn	Sn	Pb	
	P. ct.	P. ct.	P. ct.	P. ct.	
Gun metal	.....	.....	.....	.....	1825
Leaded gun metal	85.4	1.9	9.7	3.0	1795
Red brass	.....	.....	.....	.....	1780
Low-grade red brass	81.5	10.4	3.1	5.0	1795
Leaded bronze	.....	.....	.....	.....	1735
Bronze with zinc	84.6	5.0	10.4	.....	1795
Half yellow, half red	75.0	20.0	2.0	3.0	1690
Cast yellow brass	66.9	30.8	.....	2.3	1645
Naval brass	61.7	36.9	1.4	.....	1570
Manganese bronze	.....	.....	.....	.....	1600

Copper-zinc Alloys			Copper-tin Alloys		
Parts by weight		Melting point	Parts by weight		Melting point
Copper	Zinc	Deg. fahr.	Copper	Tin	Deg. fahr.
			95	5	1920
			90	10	1840
			85	15	1760
95	5	1960	80	20	1635
90	10	1930			
85	15	1880			
80	20	1830	Copper-lead Alloys		
75	25	1795			
70	30	1725	Copper	Lead	Deg. fahr.
65	35	1660	95	5	1950
60	40	1635	90	10	1920
			85	15	1895

Effect of Heat Treatment. As these alloys contain different constituents which are stable only at certain elevated temperatures, the effect of quenching alloys at those temperatures at which the various constituents are stable will result in the properties of the alloys being modified by reason of containing such constituents. This accounts for the fact that bronze acts in a manner reverse to steel when quenched above 930 deg. fahr. in water. If quenched at this temperature the alloy is more malleable and stronger, for the reason that the hard and brittle constituent Cu_4Sn has been prevented from forming. This constituent does not form unless the alloy is cooled in the air to below 930 deg. fahr. This important fact is taken advantage of commercially in modifying the properties of copper-tin alloys in a way analogous to the annealing treatments of steel.

Strength of Bronze at Various Temperatures in per cent of the strength at 68 deg. fahr. averages about as follows:

Temperature, deg. fahr.	212	392	572	752	932
Tensile strength (relative)	101	94	57	26	18

Bell Metal alloys are alloys of copper and tin containing from 15 to 25 per cent of tin. The higher the tin content the more brittle the alloys become, and the higher the note which they will produce.

Copper-Zinc Alloys (Brasses)

The term **brass** is properly applied to alloys of copper, the properties of which are modified chiefly by the addition of zinc up to approximately 42 per cent. Many of the commercial brasses contain small percentages of tin and lead, and certain well-known alloys belonging under the general classification of brass have their properties modified by the addition of iron, manganese and aluminum. Alloys containing approximately 50 per cent each of copper and zinc are known as **brazing solders.**

Industrial Brass may be divided into two classes, namely, cast brass and wrought brass.

Cast Brass varies in its zinc content from 30 to 40 per cent. The most desirable mixture contains 65 per cent copper and 35 per cent zinc. The alloy is given hardness by the addition of a small percentage of tin. The straight alloy is soft and ductile and drags severely under the tool, but the addition of 1 to 2 per cent of lead allows it to be machined freely, producing short chips.

There are two classes of **yellow brass** ingots made from scrap on the market intended **for casting purposes:**

1. Ingots produced from rod chips, which when melted down make a very satisfactory casting brass; it contains little tin, has the necessary hardness and fluidity in casting, enough lead to insure free machining, and is practically free of iron. A typical analysis is 63 per cent copper, 2 per cent lead, no tin, balance zinc.

2. Ingots made from miscellaneous yellow brass scrap and containing from about ½ to 2 per cent of iron, and often lead and tin in excess of the amounts desirable for good working; used only for cheap classes of work, such as plumbers' ferrules, etc.

U. S. Navy Specifications (Adopted, 1917) for Naval Brass Castings. Castings shall be made of raw materials of a high grade; clean high-grade scrap may be used provided resultant alloy is in accordance with percentage chemical requirements, which are: Copper, 60–65; tin, 0.5–1.5; zinc, remainder; iron and aluminum each (max.) 0.5; lead (max.), 1.0; normal proportions, 62–1–37.

Table 14. Properties of Various Copper-Zinc Alloys

(From Report of U. S. Test Board, 1879–1881)

Composition by analysis		Specific gravity	Color	Fracture	Tensile strength, lb. per sq. in.	Elastic limit, per cent of breaking load	Modulus of rupture, lb. per sq. in.	Elongation in 5 in., per cent	Deflection, in in. of 1-in. bar 22 in. long		Crushing strength, lb. per sq. in.	Max. torsion moment, ft.-lb.	Angle of torsion, deg.
Copper	Zinc												
97.83	1.88	8.791	Yellow-red..	Vesicular.	27,240	.....	.....	.....	.....		.....	130	357
82.93	16.98	8.633	Red-yellow..	Earthy...	32,600	26.1	23,197	26.7	Bent		.....	155	329
81.91	17.90	8.598	Yellow.....	Earthy...	32,670	30.6	21,193	31.4	Bent		.....	166	345
77.39	22.45	8.574	Yellow.....	Earthy...	35,630	20.0	25,374	35.5	Bent		.....	169	311
76.65	23.08	8.528	Yellow.....	Earthy...	30,520	24.6	22,325	35.8	Bent		42,000	165	267
73.20	26.47	8.465	Yellow.....	Earthy...	31,580	23.7	25,894	38.5	Bent		.....	168	293
71.20	28.54	8.444	Yellow.....	Earthy...	30,510	29.5	24,468	29.2	Bent		.....	164	269
69.74	30.06	8.384	Yellow.....	Earthy...	28,120	28.7	26,930	20.7	Bent		.....	143	202
66.27	33.50	8.371	Red-yellow..	Earthy...	37,800	25.1	28,459	37.7	Bent		.....	176	257
63.44	36.36	8.411	Red-yellow..	Earthy...	48,300	32.8	43,216	31.7	Bent		.....	202	230
60.94	38.65	8.405	Red-yellow..	Earthy...	41,065	40.1	38,968	20.7	Bent		75,000	194	202
58.49	41.10	8.363	Red-yellow..	Earthy...	50,450	54.4	63,304	10.1	Bent		.....	227	93
55.15	44.44	8.283	Red-yellow..	Earthy...	44,280	44.0	42,463	15.3	Bent		78,000	209	109
54.86	44.78	8.301	Red-yellow..	Coarsely granular.	46,400	53.9	47,955	8.0	Bent		.....	223	72
49.66	50.14	8.291	Red-yellow..	Coarsely granular.	30,990	54.5	33,467	5.0	1.26		117,400	172	38
48.95	50.82	8.216	Pinkish-gray	Coarsely granular.	26,050	100.0	40,189	0.8	0.61		.....	176	16
47.56	52.28	.....	Pinkish-gray	Coarsely granular.	24,150	100.0	48,471	0.8	1.17		121,000	155	13
43.36	56.22	8.035	Pinkish-gray	Finely granular.	9,170	100.0	17,691	....	0.10		.....	88	2
41.30	58.12	8.061	Silver white.	Conchoidal	3,727	100.0	7,761	....	0.04		.....	18	2
32.94	66.23	7.811	Silver white.	Conchoidal	1,774	100.0	8,296	....	0 04		.....	29	1
29.20	70.80	7.766	Light gray...	Vitreous.	6,414	100.0	16,579	....	0 04		.....	40	2
20.81	77.63	7.418	Bluish-gray.	Finely granular.	9,000	100.0	22,972	0.2	0.13		52,152	65	1
12.12	86.67	7.238	Bluish-gray.	Finely granular.	12,413	100.0	35,026	0.4	0.31		.....	82	3
4.35	94.57	7.108	Bluish-gray.	Finely granular.	18,065	100.0	26,162	0.5	0.46		.....	81	22
0.00	100.00	7.143	Bluish-white	Crystalline	5,400	75.0	7,539	0.7	0.12		22,000	37	142

Test pieces for compression tests were 2 in. long and ⅝ in. diam.
Test pieces for torsion tests were 1 in. long and ¾ in. diam.

General Requirement for Alloy Castings. **Test coupons** shall be provided for each casting or lot of castings, a lot being construed as comprising all the castings poured from one ladle or crucible of metal. At least two test coupons shall be cast attached to each casting weighing 250 pounds or more as delivered from the foundry. The dimensions of these test coupons shall be 3 inches by 1 inch by 8 inches long, and shall be cast as a part of the casting and not gated thereto.

When castings weigh less than 250 pounds each as delivered from the foundry, coupons shall be cast attached to the first and to the last casting poured from each lot of not over 500 pounds. These coupons shall be ⅞ inch in diameter by 6 inches long.

Workmanship. Castings shall be in accordance with detail specifications and shall be sound, clean, and free from blowholes, porous places, cracks, segregation to such an extent as to result in a spotted surface after machining, or other injurious defects.

Fracture. The color of the fracture and the grain of the metal shall be uniform throughout.

Marine Use. **Naval Brass** is suitable for: Hatch-frames, hatch-cover frames, door-frames, scuttle-frames, fittings for mess tables and benches; skylight and chest hinges

and fittings; rail and ladder stanchions; brackets, clips, etc., for stowage purposes; fittings for canopy frames; all brass valves and fittings of ventilation system (except working parts); belaying pins, tarpaulin hooks, brass hatch and door fittings, brass-pipe flanges. Valve handwheels, handrail fittings, ornamental and miscellaneous castings and valves in water chests of condensers.

U. S. Navy Specifications (Adopted, 1917) for Commercial Brass Castings: The percentage chemical requirements are: Copper (min.), 62; zinc (min.), 30; iron (Max.), 0.75; lead (limits), 1.5–3; nickel, tin, or both, remainder.

Marine Use. Material is suitable for name and number plates, cases for instruments, oil cups, distribution boxes, etc.

Wrought Brass. There is some confusion of nomenclature in relation to wrought brass. The names in the table on p. 57 are the usual commercial names in the U. S. In addition an alloy of 85 copper and 15 zinc is called **rich low brass.** The nomenclature is entirely different from that in use in England.

Hot-working Brass. Alloys containing approximately 56 to 62 per cent copper and the balance zinc become plastic when heated to redness and are capable of hot working. The most widely used and known alloy within these specifications is **Muntz metal** (60 per cent copper and 40 per cent zinc), which was formerly largely used for ship sheathing, it being claimed that this alloy is corroded just sufficiently to prevent the attachment of barnacles. The alloy is also used for many purposes in which a hard sheet brass is desirable. According to Bengough and Hudson, it should not be rolled or extruded at a temperature below 1100 deg fahr. It is hardened by quenching. Tensile strength, about 55,000 lb. per sq. in.

U. S. Navy Specifications (Adopted, 1916) for Cast Muntz Metal: Percentage chemical requirements: Copper, 59–62; zinc, 38–41; lead (max.), 0.6. No physical requirements are specified.

U. S. Navy Specifications (Adopted, 1918) for Rolled Muntz Metal Bars, Plates, Sheets, and Shapes: Chemical requirements the same as Cast Muntz Metal except iron (max.) 0.2 per cent. Physical requirements: tensile strength, lb. per sq. in., 45,000; elastic limit, lb. per sq. in., 25,000; elongation in 2 in., min., 25 per cent. Tolerances same as brass sheet.

Marine Use. Rolled muntz metal is suitable for bolts and nuts not subject to the action of salt water, sheathing bottoms of wooden boats, etc.

U. S. Navy Specifications (Adopted, 1916) for Brazing Metal (More properly called Flange Metal) call for the following percentage chemical composition: Copper, 84 to 86; zinc, remainder. Iron must not exceed 0.06 per cent and lead 0.3 per cent.

Marine Use. This material is suitable for flanges for copper pipe and other fittings that are to be brazed.

U. S. Navy Specifications (Adopted, 1918) for Rolled Naval Brass Bars, Shapes, Plates, and Rods (Abridged):

Scrap shall not be used in the manufacture of rolled naval brass, except such as may accumulate in the manufacturers' plants from material of the same composition of their own make. *Note: This clause appears on all Naval Rolled Alloy Specifications.*

The chemical and physical requirements are: Copper 59–63 per cent; tin 0.5–1.5 per cent; zinc, remainder, iron (max.), 0.06 per cent; lead (max.), 0.2 per cent.

Rods

Diameter or thickness Inch	Minimum tensile strength Lb. per sq. in.	Minimum yield point Lb. per sq. in.	Minimum elongation in 2 inches, per cent	Bend 120 deg. cold.
0 to ½	60,000	27,000	35	Radius equal to thickness.
Over ½ to 1	53,000	26,000	40	Do.
Over 1	54,000	25,000	40	Do.

Shapes

Diameter or thickness	Minimum tensile strength Lb. per sq. in.	Minimum yield point Lb. per sq. in.	Minimum elongation in 2 inches, per cent	Bend 120 deg. cold
All	56,000	40 per cent. of the tensile strength.	30	Radius equal to thickness.

Plates

0 to ½ in. up to 30 in. width	56,000	28,000	30	Radius equal to thickness.
0 to ½ in. above 30 in. width	54,000	27,000	35	Do.
Over ½ in	56,000	25,000	35	Do.

Tubing

Wall thickness Inch				
0 to ⅛	60,000	36,000	22	
Over ⅛ to ¼	55,000	33,000	25	
Over ¼	50,000	30,000	27	

Tolerances. (*a*) No excess weight will be paid for, and no single piece that weighs more than 5 per cent above the calculated weight will be accepted. (*b*) Plates and sheets shall not vary throughout their length or width more than 5 per cent for sheets or plates under 48 in., 7 per cent between 48 and 60 in., and 8 per cent when over 60 in. wide.

Marine Use. The material is suitable for parts especially subject to corrosion as by salt water: **Bolts, studs, nuts, and turnbuckles:** rolled rounds, used principally for propeller blade bolts, air pump and condenser bolts and parts requiring strength and incorrodibility; **and pump rods, tube sheets, supporting plates, and shafts for valves** in water heads.

U. S. Navy Specifications (Adopted, 1917) for Commercial Brass for Rods, Bars, Shapes, Sheets, and Plates require Copper. 59–63 per cent; tin (max.), 1.5 per cent; zinc, 32–41 per cent; lead (max.), 3 per cent; iron (max.), 0.2 per cent. The material shall be clean, smooth, of uniform quality, color, and size, and shall be free from all injurious defects. Physical tests of this material are not required.

Marine Use. This material is suitable for liners, trim, etc. (sheet brass); hand rails, distributing oil tubes and water pipes (brass pipe), for trim and purposes where strength and incorrodibility are not required (rod).

U. S. Navy Specifications (Adopted, 1917) for Electrical Appliances (Castings for) require copper, 80–88 per cent; tin (max.), 2 per cent; zinc, remainder; iron (max.), 0.25 per cent; lead (max.), 2 per cent.

Marine Use. This material is suitable for electrical fittings such as junction boxes, switches, distribution boxes, connection boxes, water-tight bells and buzzers, etc.

Modified Hot-working Brass is brass the properties of which have been somewhat altered by the addition of tin and iron. Two alloys of this nature which are largely used are Tobin bronze and Delta metal.

Tobin Bronze is a proprietary alloy of the American Brass Co., containing, according to published analyses, 58 to 60 per cent of copper and about 40 per cent of zinc, the remainder being iron, tin and lead. It is made in the form of sheets and plates ($\frac{1}{16}$ to 2 in. thick), rods (up to 7 in. diam.), rectangu-

lar bars and seamless tubes. At a cherry-red heat is can be **readily forged.** It is especially **resistant to corrosion,** and is used largely in naval work. **Tensile strength** of rods **larger than** 1 in. in diam.. 60,000 lb. per sq. in. (62,000 for smaller rods); minimum **yield point,** ½ × tensile strength; **elongation** in 2 in., rods larger than 1 in., 28 per cent (25 per cent for smaller rods); compressive strength, 170,000 to 180,000 lb. per sq. in. These properties make it adaptable for a wide variety of engineering purposes where a strong, reliable material is required. **Weights of sheets and rods** are 1¼ per cent lower than the weights for brass rods and sheets in Tables 3 and 5.

Delta Metal is similar in composition and properties to Tobin bronze, except that it carries from 1 to 2 per cent of iron. It is manufactured by the Phosphor Bronze Smelting Co., and is furnished in sheets and rods.

Cold-working Brass. For tubes, cartridge cases, wire drawing, etc., the standard alloy contains 70 per cent copper and 30 per cent zinc; this alloy possesses considerable strength and great ductility; higher-copper alloys have still greater ductility. This alloy **can be worked cold,** and when intended for rolling or drawing is cast into a mold of such size and shape that the alloy may be formed into its finished shape with as little working as possible. In wire drawing, ingots of suitable size are first rolled into rods, these being finally drawn into wire. **Spring brass** has the composition copper 72, zinc 28, or copper 66⅔, zinc 33⅓. This alloy **hardens under** the effect of **working** and must be frequently annealed in order to correct this condition and allow further reduction. After each **annealing** process the brass must be washed with acid to rid the surface of oxide. The **temperature of annealing** is an **important** matter and must be carefully observed. Annealing below 536 deg. fahr. has practically no effect. There is a particular temperature at which the best results are obtained for each alloy, depending on the amount of hardening the alloy has undergone. At 620–650 deg. fahr. there is a marked softening of 70–30 brass. The maximum effect of annealing is reached **at 1400 deg. fahr.** Above this temperature the brass is burnt. Alloys **containing** small percentages of tin and lead can be raised to higher temperatures than those not containing these metals without being burnt.

Lead and tin are frequently added to cold-working brass alloys, in order to confer certain properties. **Lead** is **added to facilitate machining,** chips of a leaded brass leaving the tool easily, as they are short and brittle. Lead is mechanically mixed in brass and therefore breaks up the continuity of the structure. In brass containing an excess of lead, the lead is present in solid solution, rendering the alloy more brittle. Turnings from brass which does not contain lead come off in long, tenacious curls. Leaded brass must be rolled cold because it is "hot-short." The best alloy is one which contains 60 per cent copper, 38 per cent zinc and 2 per cent lead. This has an average **tensile strength** in the rolled form of about 60,000 lb.; elongation, 15 to 20 per cent; reduction of area, over 50 per cent. Tin is often added to brass to **increase the hardness and to add to the resistance of the metal to the** corrosive **action of sea water.** Antimony and bismuth are detrimental impurities. Arsenic is considered beneficial by some makers.

U. S. Navy Specifications (Adopted, 1918) for Admiralty Metal call for the following percentage chemical composition: copper, 70; tin, 1; zinc, remainder; iron (max.), 0.06; lead (max.), 0.075.

Marine Use. This material is suitable for condenser tubes, distiller tubes, feed water heater tubes, and evaporator tubes.

For **weights of brass sheets and plates,** see p. 44. For **weights of round and square brass rods,** see p. 61.

Table 15. Weights of Round and Square Brass Rods

(Lb. per lin. ft.)

Diam., in.	Round	Square	Diam., in.	Round	Square	Diam., in.	Round	Square	Diam., in.	Round	Square
1/16	0.0113	0.0144	1 1/16	1.37	1.74	1 5/16	4.99	6.36	1 15/16	10.9	13.9
1/8	0.0458	0.0576	1 1/8	1.63	2.08	1 3/8	5.48	6.97	2	11.6	14.8
3/16	0.1020	0.1300	1 3/16	1.91	2.44	1 7/16	5.99	7.62	2 1/16	13.1	16.7
1/4	0.1810	0.2300	1 1/4	2.22	2.83	1 1/2	6.52	8.30	2 1/8	14.7	18.7
5/16	0.2830	0.3600	1 5/16	2.55	3.24	1 9/16	7.07	9.01	2 3/16	16.3	20.8
3/8	0.4070	0.5190	1	2.90	3.69	1 5/8	7.65	9.74	2 1/4	18.1	23.1
7/16	0.5550	0.7060	1 1/16	3.27	4.16	1 11/16	8.25	10.50	2 5/16	20.0	25.4
1/2	0.7240	0.9220	1 1/8	3.67	4.67	1 3/4	8.87	11.30	2 3/8	21.9	27.9
9/16	0.9170	1.1700	1 5/16	4.09	5.20	1 13/16	9.52	12.10	2 7/16	24.0	30.5
5/8	1.1300	1.4400	1 1/4	4.53	5.76	1 7/8	10.20	13.00	3	26.0	33.2

Table 16. Approximate Weight of Brass Wire

(Lb. per 1000 Ft.)

B. & S. gage No.	Weight	B. & S. gage No.	Weight	B. & S. gage No.	Weight	B. & S. gage No.	Weight
0000	610.0	8	47.60	19	3.710	30	0.2900
000	483.0	9	37.70	20	2.940	31	0.2300
00	383.0	10	29.90	21	2.330	32	0.1820
0	304.0	11	23.70	22	1.850	33	0.1440
1	241.0	12	18.80	23	1.470	34	0.1150
2	191.0	13	14.90	24	1.160	35	0.0908
3	152.0	14	11.80	25	0.923	36	0.0720
4	120.0	15	9.38	26	0.732	37	0.0571
5	95.4	16	7.44	27	0.581	38	0.0453
6	75.6	17	5.90	28	0.460	39	0.0359
7	60.0	18	4.68	29	0.365	40	0.0285

Heat-treatment of Brass

The physical properties of annealed brass vary according to the temperature, and to some extent to the time in which the annealing has been done. When annealed at 650 to 750 deg. fahr. under ordinary annealing conditions, brass stripe containing two parts copper and one part zinc may have a tensile strength of about 55,000 lb. per sq. in., with elongation in 2 in. of 40 per cent. When annealed at 1300 to 1400 deg. fahr., under similar conditions, they may have a tensile strength of about 40,000 lb. per sq. in., with elongation in 2 in. of 55 per cent. The process of annealing relieves those internal strains which result from cold-working the material. Annealing of cold-worked brass is accompanied by a re-distribution of the molecules resulting in the formation of a new crystalline structure. The average minimum temperature required to effect re-crystallization of the brasses in one-half hour is 750 deg. fahr. The size of the grain is roughly proportional to the temperature within the safe annealing range (750–1400 deg. fahr.), and for a given time, temperature and alloy the grain size is constant. Under these conditions the grain size has a definite relation to the physical properties; also, grain size has considerable influence on resistance to corrosive action. Impurities such as iron, nickel, etc., retard the growth of the grain size and render necessary the use of a higher temperature to secure a given grain size than with pure brass.

Annealing removes, to a very great extent, the liability of brass to crack or disintegrate on prolonged exposure under unfavorable conditions. For this reason the best practice requires that in structural work all material for tubes, rods, bolts, etc., shall be lightly annealed.

Special Bronzes and Brasses

Casting Manganese Bronze. Manganese bronze is in reality a brass, consisting largely of copper and zinc. The term bronze, however, has been generally applied to it for many years. It has a light yellow or golden color. It is very **much stronger than** the **ordinary brasses.** Manganese is simply used as a deoxidizing agent, and the finished alloy very often contains but mere traces of it.

Manganese bronze is now one of the standard alloys on the market and is highly **recommended for propellers** which are used in salt water, both because of the resistance of the metal to the action of salt water, and its remarkable strength and ductility. It is also a favorite alloy for many castings used in automobile construction. In casting, large risers are necessary, due to the high shrinkage of the alloy.

The A. S. T. M. Specifications for Manganese Bronze Ingots having notched flat bottoms approx. 3×2¾ in. wide by 12 in. long, call for the following percentage chemical composition: Copper, 53 to 62; zinc, 36 to 45; aluminum, 0.05 to 0.5; lead; not over 0.15. Ultimate tensile strength not to be less than 70,000 lb. per. sq. in., elongation in 2 in., not less than 20 per cent. Test specimen to be turned to 0.5 in. diam. and 2 in. gage length, from a piece cut from one corner near bottom of ingot. The compressive strength is about 120,000 lb. per sq. in.

The U. S. Navy Specifications (Adopted, 1917) for Manganese Bronze Ingots from 9 in. to 12 in. long call for the following percentage chemical composition: copper, 55 to 62; zinc, 38 to 42; tin and aluminum, each, not over 1.5; manganese, not over 3.5; iron, not over 2, and lead, not over 0.2. The sum of the specified elements shall equal 99.9 per cent. Tensile strength not to be less than 70,000 lb. per sq. in.; elongation in 2 in., not less than 20 per cent.

Marine Use. This material is suitable for propeller hubs; propeller blades; engine framing; composition castings requiring great strength, such as main gearing in steering engine, worm wheels in windlass and where great strength and resistance to corrosion are required.

U. S. Navy Specifications (Adopted, 1917) for Cast Manganese Bronze call for the same elongation and chemical requirements as Manganese Bronze ingots except that the sum of the specified elements shall equal 99.8 per cent. The minimum tensile strength allowed, 65,000 lb. per sq. in. Two grades are specified, Grade A, made from virgin materials, and Grade B, from materials not specified.

Marine Use. Grade A is suitable for propeller blades, propeller hubs, engine framing, and in general for all castings requiring great strength and purity of material. Grade B material is suitable for all manganese bronze castings where Grade A is not specifically designated.

U. S. Navy Specifications (Adopted, 1917) for Rolled Manganese Bronze call for the following percentage chemical composition: Copper, 57 to 60; tin, 0.5 to 1.5; zinc, 40 to 37; iron, 0.8 to 2; lead (max.), 0.2; manganese (max.), 0.3. Tensile strength not to be less than 72,000 (70,000) for stock 1 in. and below (above 1 in.) and the elongation in 2 in. 30 per cent; yield point is 50 per cent of tensile strength. Bars must stand being bent cold through an angle of 120 deg. and to a radius equal to the diameter or thickness of the test bar.

Marine Use. Material is suitable for rolled round bars requiring great strength where subject to corrosion and salt water, valve stems, propeller blade bolts, air pump and condenser bolts, and parts requiring strength and incorrodibility.

Wrought Manganese Bronze. Manganese bronze is capable of being wrought similar to Muntz metal, Delta metal and Tobin bronze, except that it extrudes with great difficulty.

Tensilite, a high-strength bronze manufactured by the American Manganese Bronze Co., is stated to have the following physical properties: Tensile strength, rolled or forged, 120,000 lb. (cast, 105,000); elastic limit, 75,000 (60,000); elongation in 2 in., 18 per cent (15); reduction of area, 22

per cent (20); elastic limit in compression, 70,000 (60,000); permanent set under a load of 100,000 lb. per sq. in., 0.015 in. (0.020).

Aluminum Bronze is an alloy of copper and aluminum, containing up to 11 per cent of the latter metal. Some years ago it was a strong competitor of manganese bronze, and in general it may be said that 11 per cent aluminum bronze (the strongest mixture) has a somewhat higher tensile strength than manganese bronze, but a lower percentage of elongation. It was found that manganese bronze was more easily handled in the foundry and could be better depended upon; and as it could be produced more cheaply than aluminum bronze, it naturally became the more popular alloy for resistance to severe stresses. By the addition of titanium good solid castings of aluminum bronze can be obtained. Aluminum beyond 11 per cent makes a brittle bronze which is hard to work. Aluminum bronze can be readily soldered.

Dagger's tests on copper-aluminum alloys yielded the following results:

Aluminum, per cent	Specific gravity	Average tensile strength, lb. per sq. in.	Elongation, per cent
11.00	7.23	95,000	8
10.00	7.69	82,000	14
7.50	8.00	61,000	40
5.00	8.37	37,000	40
2.50	8.69	31,500	50
1.25	.....	27,000	55

Aluminum Brass is a copper-zinc alloy containing up to 3 per cent of aluminum, and a very valuable series of alloys is thus produced with a wide variation of tensile strength and elongation. The alloys are quite analogous to the so-called manganese bronzes, and are often substituted for them. Aluminum brass, as made by the Aluminum Co. of America, has a tensile strength of 40,000 to 50,000 lb. per sq. in. (elastic limit, 30,000), and an elongation of 3 to 10 per cent in 8 in.

Aluminum, when added in the form of aluminised zinc to brass in quantities up to 1 per cent, makes the brass flow freely, and the resulting castings have smooth surfaces and are free from blowholes. Added in larger quantities (up to 10 per cent) it imparts increased strength to the castings.

U. S. Navy Specifications (Adopted, 1918) for Aluminum Bronze Ingots and for Cast Aluminum Bronze require the following percentage chemical composition: Copper, 85 to 87; iron, 2.5 to 4.5; aluminum 7 to 9; tin (max.), 0.5; max. sum of impurities including lead, zinc, etc., 0.1. Only the best grades of virgin materials, or scrap of known and approved composition, shall be used in the manufacture of aluminum bronze castings and ingots.

Aluminum bronzes differing from the above composition in that they may contain nickel or deoxidising agents such as vanadium, titanium, etc., will be considered provided that a complete chemical analysis is given by the manufacturer and that other requirements are complied with. The sum of aluminum, copper and iron shall not be less than 99 per cent.

The following minimum physicalpr operties are required: tensile strength, 65,000 lb.; yield point, 35,000 lb.

Marine Use. Suitable for use where a non-corroding material possessing great strength and hardness similar to manganese bronze and where good bearing qualities combined with strength are desirable as for struts, rudder frames, propeller blades, worm wheels, gears, etc.

U. S. Navy Specifications (Adopted, 1918) for Rolled Aluminum Bronze call for the following percentage chemical composition: Copper, 85 to 87; iron, 2.5 to 4.5; tin (max.), 0.5; aluminum, 7 to 9; max. sum of impurities including lead, zinc, etc., 0.1. Only the best grades of virgin materials, or scrap of known and approved composition shall be used in the manufacture of rolled aluminum bronze.

Aluminum bronzes differing from the above composition in that they may contain deoxidizing agents such as vanadium, titanium, etc., or nickel, will be considered provided that a complete chemical analysis is given by the manufacturer and that the other chemical requirements are complied with. The sum of aluminum, copper and iron shall not be less than 99 per cent.

All aluminum bronze bars, sheets, plates and shapes shall be hot rolled, drawn or extruded, but may be finished cold.

The physical properties shown below are required:

U. S. Navy Physical Requirements for Rolled Aluminum Bronze

RODS AND BARS

Diameter or thickness, in.	Tensile strength, minimum lb. per sq. in.	Yield point, minimum lb. per sq. in.	Elongation in 2 in., minimum per cent	Cold bend, 120 deg.
Up to ½ in., inc.....	80,000	40,000	30	Radius equal to diameter or thickness.
Over ½ to 1 in., inc...	75,000	37,500	30	Do.
Over 1 in.....	72,000	35,000	30	Do.

SHAPES

Diameter or thickness, in.	Tensile strength, minimum lb. per sq. in.	Yield point, minimum lb. per sq. in.	Elongation in 2 in., maximum per cent	Elongation in 8 in., minimum per cent	Cold bend, 120 deg.
All.....	75,000	37,500	30	25	Radius = thickness.

SHEETS AND PLATES

0 to ½ in. (up to 30 in. width).....	80,000	40,000	30	25	Radius equal to thickness.
0 to ½ in. (over 30 in. width).....	78,000	39,000	30	25	Do.
Over ½ in.....	75,000	37,500	30	25	Do.

Marine Use. This material is suitable for rolled rods and bars requiring great strength where subject to corrosive action, valve stems, propeller blade bolts, air pump and condenser bolts, etc., and for purposes requiring great strength combined with good bearing qualities, such as slide liners, etc.

U. S. Navy Specifications (Adopted, 1916) for Vanadium Bronze Castings call for the following percentage chemical composition: Copper (min.), 61; tin, including vanadium, lead, bismuth, aluminum, and nickel (max.), 1; zinc 37 to 39. Tensile strength, 55,000 lb. per sq. in.; yield point 22,500 lb. per sq. in.; elongation in 2 in., 25 per cent.

Casting Red Brass. The term red brass is applied to all alloys containing a sufficient amount of copper to be red in color, with the exception of those reddish alloys of copper and zinc carrying approximately 45 per cent of zinc and 55 per cent of copper: manganese bronze, aluminum bronze and aluminum brass. Red brasses usually carry upward of 75 per cent of copper and varying amounts of tin, zinc and lead. These alloys, because of their ease of machining, ease of casting and red color, are used for steam fittings, ornamental cast-

ings, miscellaneous brass trimmings, hardware, etc. Different alloys for various purposes have the following percentage compositions:

	Copper	Tin	Lead	Zinc
Valve metal*	85	5	5	5
Steam metal	86	8	3	3
Gas cocks	76	2	6	16
Polishing metal	87	.....	1	12
Brazing metal	80	.....	.....	20

* S. A. E. Specification No. 27 for light castings.

Red Brass Ingots made from scrap, turnings, etc., are sold upon the market, subject to specifications with a variation of 1 per cent either way on the constituents, at a price below the cost of producing the same composition from new metals. **This material is not suitable for Marine use.**

Phosphor Bronze in the **wrought form** (bars and plates) consists of upward of 96 per cent of copper and up to 4 per cent of tin which has been deoxidised by sufficient phosphorus. A small percentage of phosphorus can be left in the alloy, and adds to its hardness. Its qualities of toughness and great tensile strength, combined with resistance to corrosion and crystallization, make it superior to German silver for exposed work. It has superior elasticity and can be furnished in wire and sheets to various tempers for use in making springs.

U. S. Navy Specifications (Adopted, 1916) for Rolled or Drawn Phosphor Bronze call for the following percentage chemical composition: Copper (min.), 94; tin (min.), 3.5; zinc (max.), 0.3; iron (max.), 0.1; lead (max.), 0.2; phosphorus, 0.05 to 0.5. The following physical properties are required for rods and bars: Tensile strength, lb. per sq. in., up to ½ in. diam., 80,000; ½ in. to 1 in., 60,000; over 1 in., 55,000. Yield point, lb. per sq. in., up to ½ in., 60,000; ½ in. to 1 in., 40,000; over 1 in., 30,000. Elongation in length of 4 times diam., up to ½ in., 12 per cent; ½ in. to 1 in., per cent in 2 in., 20; over 1 in., per cent in 2 in., 25. 120 deg. cold bend test on radius equal to thickness. Physical properties for sheets and plates, spring temper. Tensile strength, lb. per sq. in., 90,000; yield point and elongation not specified. For sheets and plates, medium temper: Tensile strength, 50,000; yield point, 25,000; elongation in 2 in., 25 per cent; 120 deg. cold bend test around radius equal thickness.

Marine Use. Rod material is suitable for bolt material, pump rods, valve stems, valves, and objects exposed to action of salt water. Medium temper sheets are suitable for valve discs, etc. Spring temper sheets are suitable for electric contacts.

Cast Phosphor Bronze (commercially known as "B" grade) contains from 88 to 92 per cent copper and from 8 to 12 per cent tin. To get the best results, just enough phosphorus is used to thoroughly cleanse the metal of oxides, none being retained in the finished product. The American Manganese Bronze Co. give for cast phosphor bronze, tensile strength, 25–40,000 lb. per sq. in.; elastic limit, 18–30,000 lb. per sq. in.; elongation in 2 in., 42 per cent; reduction of area, 32 per cent. To obtain the best results this alloy should contain no lead or zinc. It is a particularly desirable metal for gears, being hard and tough and having a low rate of wear.

U. S. Navy Specifications (Adopted, 1917) for Cast Phosphor Bronze specify two grades, Grade No. 1 having a tensile strength, of lb. per sq. in., of 45,000 and elongation in 2 in. of 20 per cent. Grade No. 2 has a tensile strength of 30,000 and elongation of 15 per cent. The percentage chemical compositions are copper, 85 to 90; tin, 6 to 11; Grade No. 1 (Grade No. 2), zinc, 4 (remainder); iron (max.), 0.06 (0.1); lead (max.), 0.2 (1.0); phosphorus (max.), 0.5. Proprietary bronzes differing from the above are acceptable when approved by the Bureau of Steam Engineering.

Marine Use. This material is suitable for fittings exposed to the action of salt water, for gears, driving and main nuts of steering gears, and castings for other parts which require strength combined with good bearing qualities and incorrodibility. Grade 1 should not be specified where Grade 2 may answer.

SPECIAL ALLOYS

Copper-Nickel Alloys

Cupro Nickel, or Benedict Nickel, is an alloy of 85 per cent copper and 15 per cent nickel. **U. S. Navy Specifications (Adopted, 1917)** allows 1 per cent variation from this formula. This material is suitable for tubes for condenser distillers and feed-water heaters. The addition of a small percentage of manganese or magnesium greatly facilitates the proper working of the alloy in ingots and under rolls. **Constantan** (60 per cent copper, 40 per cent nickel) has a very low temperature coefficient and is used for electrical resistance purposes.

German Silver is an alloy of copper, nickel and zinc. It is very **ductile** and can be rolled, hammered, stamped and drawn. At the same time it is **hard, tough** and not easily corroded, and above all possesses the valuable property of being **white.** It is softened by annealing.

Hiorns gives the following table showing the **composition** of various qualities of German silver made by the best makers in Birmingham, England. These different alloys are sold under the trade names indicated.

Name	Composition, per cent Copper	Nickel	Zinc	Name	Composition, per cent Copper	Nickel	Zinc
Extra white metal	50	30	20	Seconds	62	14	24
White metal	54	24	22	Thirds	56	12	32
Arguzoid	48½	20½	31	Special thirds	56½	11	32½
Best best	50	21	29	Fourths	55	10	35
Firsts or best	56	16	28	Fifths, for plated goods	57	7	36
Special firsts	57	17	27				

Hiorns states that alloys containing less than 16 per cent of nickel should contain 30 per cent of zinc in order to give the best results; while with alloys containing more than 16 per cent of nickel, the quantity of zinc should be less than 30 per cent. The **impurities** found in German silver are iron, lead and tin. Iron increases the strength, hardness and elasticity, and makes the alloy whiter. Tin renders the alloy brittle and unfit for rolling. It also gives the alloy a decidedly yellow color. Lead does not enter the alloy but simply remains mechanically mixed, and therefore should be added if the alloy is intended for casting and subsequent working.

U. S. Navy Specifications (Adopted, 1917) for German Silver call for the following percentage chemical composition: Copper 60 to 67; zinc, 18 to 22; nickel (min.) 15; iron, lead and sulphur, trace only. Fracture test may be made to determine soundness; the grain must be uniform throughout.

The **weight of German silver wire** made by the American Brass Co. and containing 18 per cent nickel is 2.5 per cent lower than that of copper wire of the same size and length; of 30 per cent nickel it is 1.8 per cent less.

Monel Metal is a natural alloy consisting of 68 to 70 per cent nickel, 1½ per cent iron, and the remainder copper. It is smelted and refined from ore mined in Canada without disturbing the proportions given. It is silver-white in color, takes a high polish and is **highly resistant to corrosion,** not being affected by atmospheric conditions, fresh or salt water, acid fumes or superheated steam. Its **specific gravity** (cast) is 8.87, and its **melting point** 2480 deg. fahr. **Tensile strength**: rods, 86,900 lb. per sq. in. (castings, 78,240); **elongation** in 2 in.: rods, 40 per cent (castings, 38.5 per cent).

U. S. Navy Specifications (Adopted, 1916) for Rolled Monel Metal Bars, Plates, Rods, Sheets, etc., call for the following percentage chemical composition: Copper 23; nickel, 60; with the following max. impurities: Iron, 3.5; aluminum, 0.5; man-

ganese, 3.5; and carbon and silicon combined, 0.8. The following tables give the physical requirements:

Rods and Bars

	Minimum yield point, lb. per sq. in.	Minimum ultimate tensile strength, lb. per sq. in.	Minimum elongation in 2 in., per cent
Rounds and squares:			
Up to and including 1 in	40,000	80,000	25
1 1/16 in. to and including 1 15/16 in	50,000	85,000	28
1 15/16 in. to and including 2 7/16 in	45,000	80,000	30
2 1/16 in. to and including 3 1/2 in	37,000	75,000	32
Over 3 1/2 in	40,000	75,000	32
Rectangles	40,000	75,000	32
Hexagons	40,000	80,000	32

NOTE.—No material less than 1/4 in. in thickness or diam. need be tested physically.

Sheets and Plates

Tensile strength, minimum, lb. per sq. in.	Yield point, minimum, lb. per sq. in.	Elongation, minimum in 2 in., per cent
65,000	30,000	15

Underweight Tolerances

NOTE.—Material shall not vary throughout its length or width more than the given tolerance.	Tolerance per cent.
Widths of sheets or plates:	
Up to 48 in	5
48 to 60 in	7
Over 60 in	8

Marine Use. This material is suitable for parts requiring strength or incorrodibility, such as propeller-blade bolts, air-pump and condenser bolts, and pump rods.

U. S. Navy Specifications (Adopted, 1916) for Cast Monel Metal call for the same chemical requirements as given above for Rolled Monel. The physical requirements are: Tensile strength, lb. per sq. in., 65,000; yield point, 32,500; elongation in 2 in., 25 per cent.

Marine Use. This material is suitable for valve fittings, plumbing fittings, boat fittings, propellers, propeller hubs, blades, engine framing, pump liners, valve seats, shaft nuts and caps, and composition castings requiring great strength.

Aluminum Alloys

Aluminum Alloys are extensively used in automobile construction. The S. A. E. Specification No. 23 for **aluminum-copper** alloy calls for 7 to 8 per cent of copper. This is a tough alloy having a tensile strength of from 15,000 to 20,000 lb. per sq. in., and is especially adaptable for castings to withstand severe shocks and stresses. Zinc produces the strongest alloys with aluminum, the tensile strength running as high as 30,000 to 35,000 lb. per sq. in. The S. A. E. Specifications Nos. 24 and 25 are for **aluminum-zinc** alloys, the former containing 12.5 per cent of zinc and 2.5 per cent of cop-

per, and the latter 35 per cent of zinc. The copper-carrying alloy is tough, well adapted to forging, and easily machined.

U. S. Navy Specifications (Adopted, 1915) for Aluminum or Light Alloy Castings call for the following percentage chemical composition: Aluminum (min.), 94; copper (max.), 6; iron and silicon (max.), each 0.5; manganese (max.), 3. The physical requirements are: Tensile strength, lb. per sq. in., 18,000; elongation in 2 in., 8 per cent.

Marine Use. This material is suitable for objects not liable to corrosion where lightness is desirable such as hand wheels, brackets, etc. It finds a special field in Ordnance work.

Magnalium is an alloy of aluminum and magnesium containing from 2 to 23 per cent of the latter metal. **Specific gravity,** 2.40 to 2.57; **melting point,** 1110 to 1290 deg. fahr. The alloy is lighter than pure aluminum, of much greater strength, and does not tarnish. **Tensile strength:** Cast in chills, 42,000 to 64,000 lb. per sq. in.; sand castings, 17,000 to 30,000 lb.; rolled, 28,000 to 35,000 lb.; stamped, 52,000 to 70,000 lb.

Effect of High Temperatures on the Physical Properties of Metals and Alloys

For a list of investigations on the effects of high temperatures on the strengths of metals and alloys, see M. Rudeloff, 1909, International Ass'n for Testing Materials. Most of the investigations have been limited in scope and have not dealt sufficiently with the cast and rolled metals at present in commercial use. An extensive series of tests by the Crane Co. on a wide range of metals and alloys is reported by I. M. Bregowsky and L. M. Spring, in *The Valve World*, Jan., 1913. The values given in Tables 17 and 18 are taken from this article, and are averages of from two to ten tests at each temperature.

ANTI-FRICTION ALLOYS (BEARING METALS)

Anti-friction Alloys are divided into **four** distinct **classes:** 1. Copper base, or those carrying over 50 per cent copper, usually from 65 to 80 per cent. 2. Tin base, or those containing over 50 per cent tin. 3. Lead base, or those containing over 50 per cent lead. 4. Zinc base, or those containing over 50 per cent zinc. Copper-base alloys are harder and stronger than white metals having tin, lead or zinc bases, and are used for bearings which are required to resist heavier pressures. Frequently, however, **compound metal bearings** are used, having a copper-base bearing metal for the backing and lines with an anti-friction metal usually of a tin or lead base.

The **copper-base metals** operate with a lower coefficient of friction because they are harder, but they are more liable to heat under abnormal conditions because they are lacking in the necessary plasticity to conform to irregularities of the journals or of foreign particles which may become lodged between the journal and the bearing surface. If bearings could be maintained in perfect adjustment, the harder alloys would give the most efficient service because of their low coefficient of friction and their lower operating temperature. The term anti-friction is generally applied to those alloys which have the highest coefficient of friction, namely, the soft white-metal alloys.

Table 17. Tensile Tests on Metals and Alloys Subjected to High Temperatures

[Tensile strength and elastic limit in lb. per sq. in.; elongation (in 2 in.) and reduction of area in per cent]

Materials	Temperatures, deg. fahr.							
	70	300	450	500	600	750	950	1000
CAST MATERIALS								
[1] Hard Metal:								
Tensile strength	33,735	34,280	31,810	.....	23,150	19,170	10,825	5,710
Elastic limit	25,035	19,630	20,845	.....	17,160	15,925	8,775	5,500
Elongation	7.4	10.7	7.8	.....	2.8	2.9	0.0	0.0
Reduction of area	9.4	12.8	9.2	.....	4.1	3.5	0.5	0.0
[2] Aluminum Bronze:								
Tensile strength	35,205	32,330	35,815	.....	24,265	6,425	.....	6,230
Elastic limit	27,740	.....	15,645	.....	18,420	6,425	.....	6,230
Elongation	56.3	78.1	65.7	.....	45.3	0.0	.....	0.0
Reduction of area	46.9	31.4	61.7	.....	38.4	0.0	.....	0.0
[3] Phosphor Bronze:								
Tensile strength	33,107	33,640	26,376	.....	16,537	15,775	.....	.....
Elongation	12.5	12.5	10.9	.....	4.7	0.0	.....	.....
Reduction of area	11.9	16.5	12.3	.....	0.8	0.0	.....	.....
Steam Metal:								(550°)
Tensile strength	31,780	26,370	21,900	20,260	12,180	10,280	6,630	12,230
Elastic limit	16,900	12,770	13,130	12,410	11,370	10,280	6,630	11,230
Elongation	21.1	18.8	10.9	8.9	1.1	0.0	0.0	0.0
Reduction of area	23.8	18.2	14.5	10.9	1.2	0.4	0.0	0.7
[4] Aluminum Bronze:								
Tensile strength	41,360	.....	42,308	.....	38,425	26,800	25,960	.....
Elongation	9.4	.....	12.5	.....	11.5	1.6	3.1	.....
Reduction of area	13.1	.....	4.8	.....	11.6	2.9	6.7	.....
[5] Cast Manganese Bronze:							(900°)	(925°)
Tensile strength	56,350	47,775	40,870	37,200	21,450	7,350	2,365	1,625
Elastic limit	31,875	26,900	25,580	22,200	13,525	5,230	1,500	1,625
Elongation	10.9	12.5	21.4	24.5	32.1	39.6	40.2	48.4
Reduction of area	15.8	17.1	25.3	33.7	41.0	54.0	61.3	63.0
Soft Cast Iron:							(860°)	
Tensile strength	22,060	23,260	20,730	.....	21,240	21,925	21,590	19,820
[6] Strong Cast Iron:							(900°)	
Tensile strength	32,692	33,290	33,400	.....	33,110	32,860	25,780	27,310
Malleable Iron:							(822°)	(950°)
Tensile strength	37,625	33,505	33,280	.....	34,000	34,055	31,830	27,110
Elongation	63.0	.....	.....	.....	6.3	6.3	.....	.....
Cast Steel:						(720°)	(750°)	
Tensile strength	73,325	76,570	81,167	.....	67,366	58,713	41,388	17,568
Elastic limit	39,817	34,020	34,267	.....	29,422	33,313	27,423	9,650
Elongation	27.0	15.6	19.8	.....	23.1	22.7	2.5	31.3
Reduction of area	35.0	24.5	23.4	.....	34.6	34.6	45.4	57.5
[7] Cast Nickel:							(900°)	
Tensile strength	38,029	40,850	36,667	.....	35,896	36,536	27,800	16,725
Elastic limit	23,798	25,123	25,137	.....	22,875	22,570	14,150	14,740
Elongation	5.7	9.6	6.5	.....	6.2	7.2	6.3	3.9
Reduction of area	6.1	14.6	10.3	.....	7.9	9.2	11.6	4.8
Monel Metal (cast):		(285°)	(435°)	(450°)				(1030°)
Tensile strength	52,870	53,130	54,100	47,200	39,450	41,787	.....	26,400
Elastic limit	30,088	24,910	23,850	22,350	21,901	21,700	.....	23,480
Elongation	16.6	20.3	25.8	15.6	18.2	14.1	.....	6.3
Reduction of area	19.8	19.9	23.6	29.5	22.0	16.9	.....	6.8
[8] U. S. Navy Brass:								
Tensile strength	28,930	27,000	25,600	20,900	13,130	9,025	.....	.....
Elastic limit	16,130	13,800	10,450	12,075	11,350	9,025	.....	.....
Elongation	19.3	18.8	19.5	17.2	1.0	0.0	.....	.....
Reduction of area	23.2	20.1	18.6	16.0	3.1	0.6	.....	.....
[9] U. S. N. Gun Bronze:								
Tensile strength	34,170	36,025	33,050	.....	21,380	19,640	9,650	.....
Elastic limit	25,650	21,900	19,650	.....	18,780	18,140	9,150	.....
Elongation	8.0	8.6	8.3	.....	3.7	0.0	0.0	.....
Reduction of area	7.3	7.1	10.0	.....	2.9	0.7	0.0	.....

Table 17. Tensile Tests on Metals and Alloys Subjected to High Temperatures—(*continued*)

Materials	Temperatures, deg. fahr.							
	70	300	450	500	600	750	950	1000
CAST MATERIALS								
[10] U. S. N. Valve Bronze:							(900°)	
Tensile strength	35,345	34,260	27,630	28,160	16,100	13,000	9,530	6,400
Elastic limit	18,740	13,210	15,220	14,520	15,225	12,275	9,530	6,400
Elongation	23.2	26.2	17.2	12.8	3.8	0.6	0.0	0.0
Reduction of area	24.9	26.0	21.0	17.4	4.3	1.1	1.7	0.0
ROLLED MATERIALS								
[11] Rod Brass:							(900°)	
Tensile strength	54,450	52,700	49,000	.....	35,050	18,740	10,170	.....
Elastic limit	45,000	43,200	39,100	.....	23,735	15,050	8,060	.....
Elongation	16.4	26.6	21.9	.....	14.9	17.2	21.9	.....
Reduction of area	18.0	34.2	26.0	.....	17.8	21.3	26.0	.....
[12] Nickel Steel:				(525°)				(1030°)
Tensile strength	99,498	97,000	84,950	83,000	69,575	45,650	.....	36,350
Elastic limit	39,850	31,700	32,250	26,200	25,650	21,100	.....	15,500
Elongation	51.2	64.1	62.5	59.4	56.3	43.0	.....	37.5
Reduction of area	59.8	65.0	65.0	66.8	72.6	59.4	.....	55.7
Monel Metal:				(525°)				(1030°)
Tensile strength	104,900	97,400	97,800	96,400	89,600	67,600	.....	47,200
Elastic limit	78,350	58,500	58,600	58,400	57,950	42,550	.....	26,800
Elongation	31.3	29.7	29.7	32.8	32.8	28.1	.....	28.1
Reduction of area	61.7	57.8	51.0	51.5	59.5	58.1	.....	60.7
[13] Cold-rolled Shafting:				(525°)		(795°)		
Tensile strength	82,800	91,850	96,083	96,250	88,525	59,500	39,250	.....
Elastic limit	76,800	77,100	72,850	75,300	54,275	53,200	30,400	.....
Elongation	21.9	21.9	21.9	18.8	25.0	25.0	35.2	.....
Reduction of area	49.5	39.1	38.7	37.5	44.2	58.5	78.0	.....

[1] A strong copper-tin bronze. [2] Containing 5 per cent of aluminum. [3] Highly resistant to acids; elastic limit not determined. [4] Containing 10 per cent of aluminum; elastic limit not determined. [5] At 550 deg.: 32,050, 19,300, 23.4 and 31.6. [6] Ferro-steel, used in extra heavy valves over 7 in. [7] Rendered malleable by special process and used for rings, disks and trimmings for iron and steel fittings for high temperatures. [8] For screw pipe fittings (Cu, 77–80; Sn, 4; Pb, 3; Zn, 13–19). [9] (Cu, 88; Sn, 10; Zn, 2.) [10] (Cu, 87; Sn, 7; Pb, 1; Zn, 5.) At 550 deg.: 17,200, 14,025, 4.4 and 6.2. [11] (Cu, 62.5; Pb, 2.5; Zn, 35.) [12] Rods, 30 per cent nickel. [13] Bessemer steel, for valve stems.

The rate of wear is greater in the hard alloys than in the soft alloys, because the particles of the hard alloys split off in service, whereas the particles of the soft alloys become compressed or flattened out before being torn off. The bearing should be just sufficiently hard and strong to carry its load. If the bearing is too soft it will roll out or be pounded out, and will probably hug the journal, squeeze out the oil film, become heated and be destroyed.

The physical properties of some of the bearing metals, as determined at the Penna. Ry. laboratories, are given in Table 19.

NOTES ON TABLE 18

[1] Bessemer steel, 0.093 per cent carbon. [2] Cumberland cold-rolled shafting, 0.083 per cent carbon. [3] Open-hearth steel, 0.084 per cent carbon. [4] (Ni, 3.25; V, 0.45; C, 0.365), oil tempered. [5] 25 per cent nickel. [6] 30 per cent nickel; turned from 1 in. to 0.8 in. diam. [7] Turned to 0.75 in. diam. from a ⅞-in. bar. [8] Cumberland shafting, 0.375 per cent carbon. [9] Annealed (Cr, 0.49; V, 0.145; C, 0.722). [10] Rolled rod (Cu, 62.5; Zn, 35; Pb 2.5). [11] Elephant (phosphor) bronze (Cu, 95.5; Sn, 4; P, 0.31).

Table 18. Torsional Tests on Metals and Alloys Subjected to High Temperatures

(Torsional strength and elastic limit in in.-lb. per sq. in. Total twist in number of turns, with number of degrees excess. Test bars turned to 0.855 in. diam. from 1½-in. rods.) See notes at foot of opposite page.

Materials	Temperatures, deg. fahr.			
	70	385	600	800
[1] Cold-rolled Shafting:				
Torsional strength	72,400	82,350	41,175	12.350
Elastic limit	41,050	32,800	26,350	6,550
Twist	3 and 152°	1 and 265°	16	8 and 132°
[2] Cold-rolled Shafting:				
Torsional strength	68,990	77,210	33,680	17,250
Elastic limit	42,710	29,430	25,460	9,034
Twist	3 and 180°	1 and 345°	9 and 115°	12
[3] Machinery Steel:				
Torsional strength	59,590	49,260	33,061	26,530
Elastic limit	24,530	9,830	7,310	2,450
Twist	7 and 130°	3 and 190°	12 and 120°	61 and 260°
[4] Nickel-vanadium Steel:				
Torsional strength	101,200	82,500	19,590	14,340
Elastic limit	57,200	36,200	13,060	6,560
Twist	3 and 140°	3 and 10°	3 and 110°	8 and 35°
[5] Nickel Steel:				
Torsional strength	91,900	64,490	41,300	.....
Elastic limit	17,250	8,150	6,560	6,560
Twist	9 and 30°	8 and 115°	8 and 40°	8 and 132°
[6] Nickel Steel:				
Torsional strength	104,800	78,550	53,170	25,350
Elastic limit	21,800	15,700	10,900	6,040
Twist	11 and 100°	6 and 330°	6 and 195°	9 and 210°
Rolled Monel Metal:				
Torsional strength	91,990	78,030	54,210	38,600
Elastic limit	37,780	36,140	19,800	10,680
Twist	11 and 150°	4 and 320°	4 and 90°	7 and 50°
[7] Rolled Monel Metal:				
Torsional strength	94,610	83,030	72,290	40,610
Elastic limit	45,510	33,940	31,300	10,910
Twist	12 and 150°	5	5 and 205°	6 and 240°
[8] Cold-rolled Shafting:				
Torsional strength	83,840	76,650	15,920	7,180
Elastic limit	42,540	36,800	2,040	1,630
Twist	2 and 280°	2 and 30°	8 and 230°	39 and 195°
[9] Vanadium Tool Steel:				
Torsional strength	137,295	124,080	67,710	.....
Elastic limit	54,100	37,750	17,820	.....
Twist	345°	1 and 250°	320°	.....
[10] Rod Brass:				
Torsional strength	51,200	43,780	14,190	.....
Elastic limit	32,600	22,180	4,100	.....
Twist	2 and 225°	2 and 185°	5 and 40°	.....
Tobin Bronze:				
Torsional strength	61,200	36,560	8,860	.....
Elastic limit	26,850	9,800	1,630	.....
Twist	2 and 155°	4	3 and 255°	.....
[11] Phosphor Bronze:				
Torsional strength	70,000	51,020	19,920	.....
Elastic limit	34,250	21,240	6,560	.....
Twist	12 and 95°	1 and 215°	15 and 200°	.....
Delta Metal:				
Torsional strength	61,630	42,860	3,265	.....
Elastic limit	26,940	14,600	1,360	.....
Twist	180°	3 and 235°	4 and 300°	.....
[12] Manganese Bronze:				
Torsional strength	61,630	37,630	8,980	.....
Elastic limit	22,040	13,060	3,260	.....
Twist	2 and 5°	3 and 110°	2 and 10°	.....

Table 19. Physical Properties of Bearing Metals

Kind of material	Composition in per cent						Tensile test			Brinell test 500-kg. load, 10-mm. ball		Compression test Original section	
	Copper	Lead	Tin	Antimony	Nickel	Zinc	Tensile strength, lb. per sq. in.	Elongation, per cent in 2 in.	Per cent reduction of area	Hardness number Natural	Hardness number ¼" comp.	lb. per sq. in.	Per cent
Phosphor bronze.	79.70	9.50	10.00	.....	...	.....	23,920	2.875 in 8"	.....	55.00	....	.....	....
Cyprus bronze.	64.75	30.00	5.00	.....	...	.....	17,380	6.5 in 8"	.....	41.25	....	.....	....
Plastic bronze.	64.00	30.00	5.00	.....	1.0	.....	21,340	10.10	8.20	45.00	53.0	64,490	44.2
Demo bronze.	60.67	32.97	4.60	.....	2.1	.....	18,170	3.00	0.35	52.00	59.0	56,220	30.4
Plumbic bronze.	50.00	50.00	.....	.....	...	.....	.....	.....	.....	21.80	....	.....	....
Standard Babbitt.	3.70	.....	88.89	7.41	...	.....	11,780	5.00	3.14	27.70	29.0	29,150	59.0
German Babbitt.	5.55	.....	88.33	11.11	...	.....	8,726	.....	.....	21.10	22.5	33,895	59.2
Souther Babbitt.	7.00	.....	84.00	9.00	...	.....	10,000	1.25	.....	34.80	48.0	28,410	44.1
Parsons white brass.	2.25	0.15	64.90	.....	...	32.93	21,416	11.25	18.72	18.00	19.0	75,460	73.8
Shonberg M. metal.	2.50	0.25	58.38	.....	...	38.93	11,710	4.00	5.76	25.10	31.5	36,755	54.8

Copper-base Bearing Metals

The copper-base alloys (usually known as bronzes) consist of the following classes: 1. Copper and tin. 2. Copper, tin and zinc. 3. Copper, tin and lead. 4. Copper, tin, lead and zinc. Occasionally there are added small proportions of other metals, which modify the general properties of the alloys.

Copper-Tin Bearing Alloys. About 30 years ago these alloys were in popular use, not only for machinery bearings but also for car and locomotive bearings, the alloy of 7 copper and 1 tin being the one most commonly used. This alloy has a high compressive strength, is quite hard and should be used only in cases where heavy pressures are encountered. Their use in other bearings is not justifiable, because the alloys have very little plasticity and therefore become heated upon little provocation, and have a high rate of wear.

Bell metal is a still harder metal than the 7-to-1 bronze, and carries from 16 to 25 per cent of tin, the balance being copper. Such bearings have been tried because of the false idea that the harder metals will better resist wear. Their use has always met with failure.

Copper-Tin-Zinc Bearing Alloys. The addition of zinc to the copper-tin alloys improves their casting qualities. Such alloys are very rarely used to-day, except for backing a softer lining metal. The housings or backs on marine bearings are usually of this alloy. It is used by the United States Government and to some extent in the automobile trade, the bearings usually being lined with a high-tin-base Babbitt metal. This is a very expensive form of bearing and its use in many cases is not justified, as more satisfactory service could be rendered by cheaper alloys.

Copper-Tin-Lead Bearing Alloys. This class of alloys is the standard bronze alloy in use to-day for bearings. The standard phosphor-bronze bearing metal is an alloy of these three metals containing a small percentage of phosphorus. It was found about 25 years ago that the addition of lead to the then standard copper-tin alloy made the alloy more plastic and therefore better able to conform to irregularities of the service without heating, and diminished the rate of wear.

Dr. C. B. Dudley as a result of many investigations found that the rate of wear in a bearing metal and its tendency to become heated in service diminish with the increase of lead and with the diminution of tin in the alloy.

The following series of alloys were experimented upon by the Pennsylvania Railroad under the direction by Dr. Dudley:

Metal Tested	Percentage composition					Relative wear
	Copper	Tin	Lead	Phosphorus	Arsenic	
Phosphor bronze, standard.	79.70	10.00	9.60	0.80	.....	1.00
Ordinary bronze.....	87.50	12.50	.....	.....	.....	1.49
Arsenic bronze, "A".....	89.20	10.00	.....	.....	0.80	1.42
Arsenic bronze, "B".....	82.20	10.00	7.00	.....	0.80	1.15
Arsenic bronze, "C".....	79.70	10.00	9.50	.....	0.80	1.01
Bronze, "K".....	77.00	10.50	12.50	.....	.....	0.92
	77.00	8.00	15.00	.....	.....	0.86

An alloy of 85 per cent copper, 10 per cent tin, and 5 per cent lead will withstand compression of approximately 24,000 lb. on a 1-in. cube at the yield point; a 1-in. cube will compress 26 per cent under a load of 100,000 lb. without fracture. The compressive strength is sufficiently high for it to be used in severe service; it is an admirable alloy for connecting-rod brasses.

An alloy of 80 per cent copper, 10 per cent tin, and 10 per cent lead, as compared with the alloy carrying but 5 per cent of lead, shows a diminished resistance to compression, namely, 23,000 lb. on a 1-in. cube at the yield point, and a compression under 100,000 lb. of 29 per cent. This alloy is very largely **used for locomotive, car, and general machinery bearings.** A small percentage of phosphorus is often added and the alloy is then known as standard phosphor-bronze bearing metal. (S. A. E. Specification No. 26—0.05 to 0.25 per cent P.) The phosphorus adds somewhat to the resistance to compression and to the fluidity of the metal, but it has not been proved that it benefits in any way the quality of the alloy from the standpoint of service as a bearing. In fact, the limited data at hand tend to show that it acts adversely in service, because of the production of the brittle phosphide of copper in the alloy which frequently tends to cause heating. This alloy, nevertheless, is called for in the specifications of many prominent consumers.

The alloy of 77 per cent copper, 8 per cent tin, and 15 per cent lead is extensively used, and together with a small percentage of phosphorus (0.2 per cent) is known on the Pennsylvania Railroad as Ex. B. metal. This is the alloy of highest lead and lowest tin content which Dr. Dudley was able to produce, owing to foundry difficulties. Alloys of lower tin and higher lead segregated, the lead liquating to the bottom of the casting, due probably to the presence of phosphorus which lowers the solidifying point of the alloy and therefore maintains the alloy in a liquid condition for a longer period. This alloy will support a load of 21,000 lb. on a 1-in. cube without distortion. It has a slow rate of wear and is less liable to heat in service than either of the two preceding alloys.

Alloys containing from 4 to 7 per cent of tin, and 20 to 30 per cent of lead, are known in the trade as **Ajax plastic bronze** alloys, and are manufactured by the Ajax Metal Co. The patent controlling these alloys is based upon the fact that lead which is only mechanically held in the alloy can be prevented from liquating to the bottom of the casting if the copper and tin matrix which holds it can be made to solidify quickly. Quick solidification of the matrix will result if the copper and tin are in correct proportions, that is, in such proportions that only a solid solution of the two metals is formed. The following table gives the results of tests made in the author's laboratory.

Copper, per cent	Tin, per cent	Lead, per cent	Wear, in grams	Copper, per cent	Tin, per cent	Lead, per cent	Wear, in grams
85.76	14.90	.....	0.2800	81.27	5.17	14.14	0.0327
90.67	9.45	.....	0.1768	75?	5?	20?	0.0277
95.01	4.95	.....	0.0776	68.71	5.24	26.67	0.0204
90.82	4.62	4.82	0.0542	64.34	4.70	31.32	0.0130
85.12	4.64	10.65	0.0380				

It was found that the addition of a small percentage of nickel adds to the compressive strength of the above series of alloys, and also elevates the temperature of solidification of the matrix. Ajax plastic bronzes of various grades, having compositions within the limits above set forth and also with the addition of 1 per cent nickel are in use very extensively for locomotive and car bearings, cold-roll neck mill bearings, etc.

Parsons Bronze. An excellent metal for the thrust blocks of Mitchell and Kingsbury Thrust Bearings is Parsons Bronze. Its percentage composition is copper, 80.75 to 84.75; tin, 12.8 to 14.8; lead, 2.45 to 4.45; material not specified (impurities), not to exceed 1. Where high pressures are encountered and sudden burning out of a bearing may endanger the turbine blading it is advisable to use this hard metal. The lubrication must be of the forced type.

Copper-Tin-Lead-Zinc Bearing Alloys are the cheapest form of bronze bearing-metal alloys on the market. Zinc is an adulterant which usually enters the alloy through the use of miscellaneous scrap. The following tests made in the author's laboratory show the effect on the rate of wear of increasing amounts of zinc.

Percentages				
Copper	Tin	Lead	Zinc	Wear, in grams
85.12	4.64	10.64	.....	0.0380
82.27	5.28	10.25	2.07	0.0415
79.84	4.71	10.30	5.44	0.0466
77.38	5.62	11.42	6.54	0.0672
74.28	4.68	10.61	11.04	0.0848

Zinc adds to the brittleness of the alloys, and this probably accounts for the more rapid rate of wear due to zinc content. The carrying of considerable zinc in the presence of fairly high percentages of lead has the further detrimental effect of causing the alloys to become somewhat weakened under the influence of elevated temperatures. The best bearing bronzes do not contain zinc, and its presence is restricted to below 1 per cent in the specifications of the most prominent consumers. The Society of Automobile Engineers specification (No. 27) for **red brass** is copper, 85; tin, lead and zinc, each 5 per cent. This is said to machine well and to be an excellent bearing metal where speed and pressure are not excessive.

Tin-base Bearing Metals

Tin-base bearing metals (tin over 50 per cent) come within the classification commonly known as **Babbitt metals**, and are divided into three classes: 1. Tin, antimony and copper alloys. 2. Tin, antimony, copper and lead alloys. 3. Tin, copper and zinc alloys. They possess in general a higher resistance to compression than the lead-base metals. The resistance to compression, however, is due in a very large measure to the relative percentages of copper and antimony which the alloys contain. Copper and antimony are the hardening constituents in such alloys, and, although they may also contain a certain percentage of lead and infrequently small percentages of other metals, the essential properties are derived from these two metals.

The name "Babbitt" is derived from that of the inventor (Isaac Babbitt) of soft-metal-lined bearings. The term "babbitting" has been applied to the process of applying soft anti-friction metals inside of a harder shell for the purpose of producing bearings. The best-known soft-metal alloy at the time of the invention was the high-grade **pewter** alloy then used, the composition of which was copper, 3.70 per cent; antimony, 7.40 per cent, and tin, 88.90 per cent. This particular alloy became known as "Genuine Babbitt," although the term is now more generally applied and includes all tin-antimony-copper alloys in which tin is the base metal, and which do not contain lead or zinc.

Soft Genuine Babbitt is usually made to the formula above given for Isaac Babbitt's original metal. Sometimes, however, this composition is slightly varied between 91 per cent tin, 4½ per cent lead, and 4½ per cent antimony, and 88 per cent tin, 4 per cent copper and 8 per cent antimony. The latter composition, as quoted by Charpy on a test piece 0.6 in. in height and 0.0155 sq. in. in sectional area, under a load of 1929 lb. showed a compression of 0.008 in., and under a load of 4960 lb. showed a compression corresponding to 0.3 in. This metal is free-flowing, tough, and gives excellent service in bearings where the loads are not heavy.

U. S. Navy Specifications (Adopted, 1917) for Anti-friction Metal calls for the following percentage chemical composition: Tin, 88 to 89.5; antimony, 6 to 8; copper, 3 to 4.5.

Marine Use. This metal is suitable for white metal liner bearings, and bearing surfaces.

The Society of Automobile Engineers Specification for Babbitt Metal No. 1 is: tin, 89 per cent; antimony, 7 per cent, and copper, 4 per cent. A variation of 2 per cent either way is permissible in the tin content, and a variation of 1 per cent either way in the antimony and copper. Only traces of other metals are permissible.

Hard Genuine Babbitt. The composition of this alloy may range from 83 per cent tin, 8½ per cent antimony, 8½ per cent copper, to 80 per cent tin, 10 per cent antimony and 10 per cent copper. For the former Charpy gives the following figures: A test piece 0.6 in. in height and 0.0155 sq. in. in sectional area, under a load of 2646 lb., gave a compression of 0.008 in., and under a load of 5622 lb. a compression of 0.3 in. The metal of the latter formula would sustain a higher load with equal compression.

The method of casting and rate of cooling have a very important influence upon the structure and service which these metals will give. The rate of cooling affects the hardness of the matrix, and the size and number of the tin-antimony crystals. The proper temperature for pouring is approximately 810 deg. fahr., and the metal should preferably be cast against a chill heated

to a temperature of approximately 212 deg. fahr., so that the rate of cooling is not too rapid.

The specification of the Society of Automobile Engineers for Babbitt Metal No. 2 is: tin, 84 per cent; antimony, 9 per cent; copper, 7 per cent. A variation of 2 per cent either way is permissible in the tin, and a variation of 1 per cent either way in the antimony and copper; only traces of other metals. Specifications Nos. 1 and 2, above given, are also used in making **die-cast bearings**. An alloy of 80 tin, 10 copper and 10 antimony is used on the high-speed bearings of De Laval turbines.

Tin-base Babbitts Containing Lead. Lead has the property of increasing the fusibility of the tin-base alloys by forming a eutectic of low melting point, but the tin present should preferably be over 50 per cent, the antimony not less than 12 per cent, and the copper not less than 3 per cent for the best results. The alloys may be made within a wide variation of composition, the hardness depending on the increasing percentages of copper and antimony.

Tin-base Babbitts Containing Zinc. The most important alloy of this class is (Parsons') white brass, which consists, approximately, of 62 per cent tin, 35 per cent zinc, and 3 per cent copper. (S. A. E. Specification: Cu, 3 to 6; Sn > 65; Zn, 28 to 30.) It is largely **used for marine bearings and** to some extent for **automobile bearings.** Such bearings should be well lubricated. It is hard, tough, is rather sluggish to pour, and cannot for this reason be cast into thin sections after the manner of using the ordinary Babbitt metals. It must be poured at a fairly high temperature, approaching redness, and is preferably peined after casting to compress the metal and so add to its durability under the influence of wear.

Lead-base Bearing Metals (Babbitts)

Lead-base bearing metals (lead over 50 per cent) are divided into three classes: (1) Lead-antimony alloys; (2) lead-antimony-tin alloys; (3) lead-antimony-tin-copper alloys. The useful alloys of lead and antimony for bearings contain from 7 to 20 per cent of antimony. Below 7 per cent the alloys are too soft, and above 20 per cent too brittle, for practical uses. Lead and antimony are insoluble in each other in the cold, and the solidified alloys therefore consist of lead and eutectic alloy if antimony is present below 12.8 per cent, and of antimony and eutectic alloy if antimony is present in excess of 12.8 per cent. This is the **cheapest class of anti-friction metal** which can be produced, as lead sells at the lowest price of any of the metals, and antimony usually sells at only two or three times the price of lead. The alloy commercially known as No. 4 Babbitt consists of 88 to 90 per cent of lead and 10 to 12 per cent of antimony. **Magnolia metal** is a lead-tin-antimony alloy, together with traces of copper, zinc, iron and possibly bismuth. A recent analysis shows lead 79, tin 5, antimony 16.

Charpy has determined the compressive strength of several of these alloys and his results are given in the following table (test pieces as given under "Soft Genuine Babbitt"):

Composition of alloy, per cent		Load corresponding to a permanent set of 0.008 in.,	Load corresponding to a permanent set of 0.3 in.,	Remarks
Lead	Antimony	lb.	lb.	
100	0.0	220	1100	
90	10	1433	2866	
82.5	17.5	1433	3197	Broke at 2756 lb.
80	20	1676	....	

Lead-Tin-Antimony Alloys are very largely in use as Babbitt metals because of their low price, and because they perform very satisfactorily if

used in the proper service and under the proper conditions. Many of the widely advertised brands of Babbitt metal come within this class.

Charpy states that the alloys of lead, tin and antimony are similar to those of lead and antimony; but the addition of tin diminishes the hardness and brittleness of the hard grains and also increases the compressive strength of the eutectic alloy. For this reason the alloys of lead, tin and antimony are superior to those of lead and antimony alone. He states that the tin must be present to the extent of more than 10 per cent, but not necessarily more than 20 per cent, and the antimony may vary between 10 and 18 per cent. The following table compiled by him shows the compressive strength of this series of alloys, test pieces as given under "Soft Genuine Babbitt," p. 75.

Composition		Load corresponding to a compression of		Composition			Load corresponding to a compression of	
Lead	Tin	0.008 in.	0.3 in.	Lead	Tin	Antimony	0.008 in.	0.3 in.
.....	100	661	2337	10	80	10	2425	5952
20	80	1323	3858	20	60	20	2976	4850
40	60	1433	3252	40	40	20	2535	4023
60	40	1323	3086	60	20	20	2315	3748
80	20	1047	2535	80	10	10	1764	3913

Lead-Antimony-Tin-Copper Alloys. The addition of copper to the alloy of lead, tin and antimony increases the hardness and causes an increased sluggishness in pouring, depending on the amount of copper present in proportion to the tin.

In the high-tin-carrying alloys the percentage of copper can be increased without increasing the sluggishness. With 10 per cent of tin, not over ½ per cent of copper should be added; with 20 per cent of tin, not over 1 per cent; with 30 per cent of tin, not over 1½ per cent; with 40 per cent of tin, not over 2 per cent, and with 50 per cent of tin, not over 3 per cent copper. A great many Babbitt metals are on the market having these four constituent metals in their composition, which are known in the trade as "copper-hardened Babbitt," and are designated by grades according to the amount of tin which they contain.

Zinc-base Bearing Metals

Lumen Metal consists of 85 per cent zinc, 10 per cent copper, and 5 per cent aluminum. This alloy has a **low coefficient of friction,** a low specific gravity, and is capable of being cast in sand after the manner of brass and bronze castings. It is **easily machined,** and because of its low price and low specific gravity has found considerable favor in certain classes of service such as motor bearings, crane bearings, etc. It should be used, however, in bearings which can be maintained in fairly good adjustment, because, owing to the high zinc content of the alloy, it becomes brittle under the effects of heating, and in the absence of an oil film is quite liable to grip the journal. It is an **excellent** metal **for high-speed bearings carrying little weight. Fenton's alloy** contains 16 per cent tin, 5 per cent copper, and 79 per cent zinc.

Alloys for Die Casting. Castings in which the molten metal is forced under pressure into accurately cut metal dies are now largely used in the place of small sand castings formerly made of brass and machined to size. It is possible in this way to produce parts of more or less intricate shape that are very accurate in size and smoothly finished. Babbitt metals are used in

making die-cast bearings, while other machine parts are made from zinc-base alloys, the two most extensively used having the following compositions:

	I	II
Zinc	89.00	82.00
Aluminum	0.25	0.25
Tin	7.00	14.50
Copper	3.25	3.25

These and similar zinc alloys have a tensile strength not exceeding 18,000 lb. per sq. in., and an exceedingly low elongation and reduction of area. They compare favorably in strength with cast iron. They are corroded by aqueous solutions and should not be used for food containers or conveyors. Commercial gasoline when in direct and constant contact will also corrode them; copper plating, however, will aid in resisting its attack.

Miscellaneous White-metal Alloys

Alloys for Metallic Packing. For casting metallic packing rings the following inexpensive alloy has been found to give the best results: Lead, 83½ per cent; tin, 8½ per cent; antimony, 8½ per cent. Metallic packing for use with superheated steam should not fuse under the temperature produced by the combined effect of the temperature of the steam and the friction on the rod, should have sufficient plasticity to produce a tight joint, and should show but a slight amount of wear. A composition fulfilling these requirements, when only a small amount of superheat is used, consists of 80 per cent lead and 20 per cent antimony.

Fusible Alloys. By the addition of varying percentages of bismuth or cadimum to lead-tin alloys of different compositions, the melting points may be greatly lowered. Such alloys expand on cooling and are little used for marine work. Some of these compositions (parts by weight), together with their melting points are as follows:

Alloy	Tin	Lead	Bismuth	Cadmium	Melting point, deg. fahr.
Lipowitz's	4	8	15	3	140
Wood's	4	8	15	4	158
Darcet's	25	25	50	.....	203
Cliché metal	2	2	5	.....	221
Rose's	24.6	28.1	50	.....	230
Bismuth solder	24.8	22.1	53.1	.....	250

Solders and Brazing Metals. Soft solders are used for joining tin plate and other metal sheets. A 90-tin-10-lead solder melts at 410 deg. fahr., and a 70-tin-30-lead alloy at 384 deg. fahr. **Plumber's solder** consists of 2 parts lead and 1 part tin. **Brazing solder,** or brazing spelter, as it is commercially termed, consists of 50 to 55 per cent of zinc, the remainder being copper. This is cast in ingots and granulated under a drop hammer into grades known as "long grain," "short grain," "fine grain," etc. The alloy mainly used as **brazing metal** consists of 80 per cent copper and 20 per cent zinc. Brazing solder and metal are used to unite brass, copper, iron and steel in strong joints.

U. S. Navy Specifications (Adopted, 1917) for Spelter Solder designate two grades: **Long-grain spelter solder** which consists of not less than 52 per cent copper, not more than 0.2 per cent lead, not more than 0.1 per cent iron, and the remainder zinc; **Grey spelter solder** which consists of 49 to 52 per cent copper, 3 to 3.5 per cent tin, not more than 0.5 per cent lead, and the remainder zinc.

Fluxes for soft solders, powdered rosin, hydrochloric acid "killed" by the addition of zinc scraps, and tallow (by plummers). Sal ammoniac is sometimes used in brazing copper.

IRON AND STEEL

BY

D. J. McADAM, JR.

REFERENCES: Harbord and Hall, "The Metallurgy of Steel," Griffin & Co. Bradley Stoughton, "The Metallurgy of Iron and Steel," McGraw-Hill Book Co. Campbell, "The Manufacture and Properties of Iron and Steel," McGraw-Hill Book Co. Becker, "High Speed Steel," McGraw-Hill Book Co. Hall, "The Steel Foundry," McGraw-Hill Book Co. Howe, "Metallography of Steel and Cast Iron," McGraw-Hill Book Co. Johnson, "Blast Furnace Operation and Construction in America," McGraw-Hill Book Co. Moldenke, "Principles of Iron Founding," McGraw-Hill Book Co. Giolitti, "Cementation of Iron and Steel," McGraw-Hill Book Co. Sauveur, "Metallography and Heat Treatment of Iron and Steel," Sauveur and Boylston. Hatfield, "Cast Iron in the Light of Recent Research," Griffin & Co. Forsythe, "Blast Furnace and the Manufacture of Pig Iron," David Williams Co.

ORES OF IRON

Composition. Commercial ores of iron consist of compounds, usually oxides, mixed with various non-ferrous minerals known as "gangue." The most important commercial ores of the United States are hematite, limonite and magnetite.

Hematite. More than three-fourths of the iron ore mined in the United States consists of hematite, Fe_2O_3, which is identical in composition with red iron rust. This mineral when pure contains about 70 per cent iron. It is either deep red or black in color and gives a red streak when rubbed on unglazed porcelain. The **red** varieties vary from compact, columnar or botryoidal masses to loose earthy material. **Black** hematite occurs in brilliant scaly masses known as specular iron ore.

Limonite, $Fe_2O_3 \cdot nH_2O$, is essentially a hydrated hematite and is sometimes called brown hematite. When pure it contains about 60 per cent iron. It varies in color from yellow to dark brown and may be soft and pulverulent or hard and compact, but is never crystalline. It gives a yellow streak on unglazed porcelain.

Magnetite, Fe_3O_4, is a heavy hard, black mineral frequently occurring in octahedral crystals. It gives a black streak on unglazed porcelain. When pure it contains a higher percentage of iron than any other mineral. In the United States, it frequently occurs mixed with non-ferrous elements that make its extraction difficult or that make the ore unsuitable for use in steel making without previous concentration. In Sweden, however, this ore occurs in a high degree of purity.

Siderite, $FeCO_3$, though rarely used as an ore in the United States, is used in certain districts of Europe, especially in the Cleveland district of England. It occurs as rhombohedral crystals which when pure contain only 48.3 per cent of iron. Before use it is always calcined to remove the carbon dioxide.

Location of Ores in the United States. Immense deposits of hematite in the Lake Superior district furnish more than ¾ of the iron ore mined in the United States. Brown hematite or limonite, together with red hematite, occurs in the Appalachian region from New York to Alabama, and is mined throughout this region, but especially in Alabama. Limonite is also mined in Colorado. Magnetite ores are mined in eastern New York, northern New Jersey, Pennsylvania and Colorado.

The Value of an Iron Ore depends on the percentage of iron that it contains. The greater part of the iron ores mined in the United States contain from 50 to 60 per cent of iron and are used without previous concentration. When an ore contains less than 40 per cent of iron, however, it must be concentrated either by washing or by magnetic separation. In some cases also the ores must be prepared by calcination, roasting or agglomeration.

The value of an ore depends also on the character of the **gangue** and the quantity of certain elements that it contains. Of all the elements occurring in the gangue **phosphorus** and **sulphur** are the most objectionable. The phosphorus usually occurs as calcium phosphate and the sulphur as iron pyrites. Unless the sulphur percentage in the ore is unusually high the percentage in the iron obtained will not be excessive since about $\frac{15}{16}$ of the sulphur in the ore and fuel is removed in the smelting process. Practically all the phosphorus in the ore, however, will be found in the iron after smelting.

The percentage of phosphorus is the basis of the classification of all iron ores as either **Bessemer or non-Bessemer** according as they can or cannot be used in the Bessemer process of steel making. If an ore contains more than 0.001 as much phosphorus as iron, it cannot be used in the Bessemer process and is therefore called a non-Bessemer ore. Within each of these two great classes, there are further subdivisions also based on the phosphorus content of the ore.

EXTRACTION OF IRON FROM ORE

Processes of making wrought iron and steel directly from the ore date back to prehistoric times. Such processes are now practically obsolete. They have been unable to compete with the blast furnace process, which produces pig iron, from which all other iron products are made.

The Blast Furnace. This is a vertical shaft furnace, 80 to 100 feet in height and circular in section. Successive layers of fuel, ore and flux are continually charged into the top of this furnace and liquid iron and slag are withdrawn at intervals at the bottom. The fuel is burned by a blast of preheated air which enters at the bottom of the shaft. There is thus a continuous, slow downward movement of the solid charge and a continuous, upward rush of heated gases. The combustible gases leaving the top of the shaft are utilized to preheat the air blast and to furnish power for auxiliary machinery. In addition to the furnace with its hoists for handling the charges and its ladles for handling the liquid products, other essential parts of the equipment are blowing engines, hot blast stoves and water pumps for cooling coils.

At the bottom of the shaft is the crucible or **hearth** about 17 feet in diameter and about 8 feet in height. Above the hearth the shaft widens out gradually through a vertical distance of 14 feet, until the diameter is about 22 feet forming what is called the **"bosh."** Above the bosh the walls converge until a diameter of 17 feet is reached again near the top of the shaft. The blast enters through a series of 10 to 15 circular openings called **"tuyeres,"** situated near the top of the hearth and arranged at regular intervals around the circumference.

The **fuel** in most general use is coke, though a few furnaces use charcoal or anthracite coal. The **flux**, usually limestone, is added for the purpose of forming a fusible slag with the gangue of the ore and the ash of the fuel. In the hot blast, the fuel burns to carbon monoxide and this gas mixed with the oxygen of the air ascends through the shaft.

The **process of iron smelting** in the blast furnace consists of two distinct stages. In the shaft above the bosh the reduction of iron oxide to solid spongy metallic iron is gradually accomplished, chiefly by the chemical action of the carbon monoxide. By the time the descending charge has reached the upper part of the bosh, the reduction of the iron is practically complete, and the lime and the gangue have united to form a viscous slag.

At the top of the bosh, the iron melts and together with the slag trickles over the burning fuel which is the only remaining solid. The iron rapidly absorbs carbon and other constituents at this stage and falls into the hearth as molten cast iron. The slag floats on the top of the molten iron and is drawn off through the slag hole. The iron is then

drawn off through a hole at the bottom level of the hearth and either cast into "pigs" or used in the molten condition for one of the steel making processes.

The productive capacity of a modern blast furnace is about 500 to 550 tons per day. For each ton of iron produced there are required about 2 tons of ore, ½ ton of flux, 1 ton of fuel and about 4 tons of air. For each ton of iron, there are produced about 5½ tons of waste gases and ½ ton of slag.

Blast Furnace Products

Control of Composition. Regulation of the proportions of the various materials of the charge is not only necessary for the successful operation of the furnace, but also is the chief means for control of chemical composition of the products. As stated above, the percentage of phosphorus in the pig iron cannot be controlled by the blast furnace operation, but depends entirely on the quantity of phosphorus in the charge. Of all the elements whose percentage can be controlled by the blast furnace process, silicon and sulphur are the most important in their effect on the quality of the iron.

Silicon. For the control of **silicon**, the chief factors are hearth temperature and slag composition. At the high temperature of the hearth, silica is reduced by the solid carbon of the fuel and the silicon is absorbed by the iron. The higher the temperature, the larger the proportion of silicon formed. On the other hand, the greater the ratio of lime and magnesia to the silica in the slag, the less silica is reduced to silicon. By varying the proportions of fuel and flux in the charge, therefore, the hearth temperature and the slag composition can be varied so as to regulate the percentage of silicon in the iron within certain limits. The upper limit which is about 6 per cent in pig iron, depends on considerations of fuel economy, since the required heat energy increases rapidly with increase in the percentage of silicon. It is extremely difficult to make iron with less than 0.2 per cent in the coke furnace without a consequent objectionable decrease in the percentage of sulphur. In the charcoal furnace iron can be produced with silicon as low as 0.1 per cent.

Sulphur is the element that indirectly produces most of the difficulties in the operation of the blast furnace. The maximum allowable percentage of this element in the iron is usually not above 0.04 or 0.05. Since coke usually contains at least 1 per cent of sulphur, and since about a ton of fuel is required for each ton of iron produced, evidently about $\frac{15}{16}$ of the sulphur in the charge must be kept out of the iron in the operation of the coke fired blast furnace. Since sulphur is carried in the slag as calcium sulphide and in the iron as iron sulphide, the proportion kept out of the iron can be increased by increasing the basicity and volume of the slag. The absorption of sulphur by the slag is also facilitated by increasing the fluidity and temperature of the slag. Since some of the factors may conflict, a proper balance of all these factors will produce the best results.

In general, high silicon in the iron tends to be accompanied by low sulphur and *vice versa*. It is difficult to obtain iron low in both silicon and sulphur from the coke fired blast furnace. Since the charcoal fired furnace has the advantage of use of a fuel almost free from sulphur, such furnaces can produce iron low in both silicon and sulphur.

The percentage of carbon in pig iron rarely rises above 4.3 or falls below 3.5. In iron from the cold blast charcoal furnace, however, the percentage may fall as low as 3.0. High hearth temperature or high percentage of manganese tends to increase while high percentage of silicon tends to decrease the absorption of carbon by the iron.

Manganese cannot be so readily controlled as can some of the other elements. Ordinarily from ½ to ¾ of the maganese in the charge goes into the iron. High temperature and basic slag favor the release of manganese from the slag and its absorption by the iron. On account of the affinity of manganese for sulphur and oxygen, it is desirable to have 0.50 to 0.75 per cent of manganese in pig iron.

Classification of Blast Furnace Products According to Uses. From the products of the blast furnace originate all these countless objects of iron or steel so essential to our civilisation. These products in the order of their importance may be divided into the following groups: steel-making irons, foundry irons, puddling irons and ferro-alloys. The second group includes all these irons which are to be used without further change except remelting, the other three groups include the pig-iron and ferro-alloys that are to be subjected to other metallurgical processes.

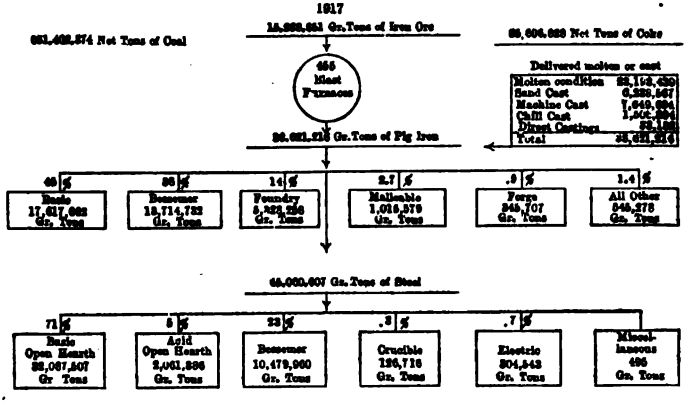


FIG. 1.

Fig. 1, by courtesy of the *Secretary, Iron and Steel Institute*, classifies all blast furnace products according to the various metallurgical processes in which they are to be utilized, and gives the approximate percentages utilized in each process. The percentages apply to the year 1917, and have doubtless been modified since that date. The duplex and triplex processes are not included in the diagram.

METALLOGRAPHY

The products of the blast furnace, as well as all iron and steel made from them, can be classified according to their microstructures as interpreted by the science of metallography. Metallography or physical metallurgy, comprises the study of the microstructures of metals in relation to chemical composition, processes of manufacture, and physical properties. In the light of this science will be discussed as far as possible all the blast furnace products, the metallurgical processes to which they are subjected, and the iron and steel

thus produced. It will be necessary, however, first to describe briefly the types of microstructure observed in metals and alloys.

Examination of the polished and etched surface of a metal or alloy usually reveals one or more of the following types of micro-constituents: Grains of pure metal, solid solutions, definite compounds, eutectics, non-metallic inclusions. The microstructures of all slowly cooled metals are made up of one or more of such constituents.

Pure Metals. The microstructure of a pure metal consists of grains of irregular shape appearing in section as irregular polygons with narrow dark lines as boundaries. Each grain is a crystal with definite orientation of axes and planes of symmetry. The boundaries of a grain, however, depend not on crystal symmetry, but on limitation of space due to surrounding grains. Since the orientation of each grain differs from that of the surrounding grains there are variations in the degree of attack of the etching liquid and consequently of the degree of shading this produced.

Solid Solutions. Many metals are more or less soluble in each other in the solid state. Some metals are mutually soluble in the solid state in all proportions. When such an alloy as cast is examined microscopically, it shows irregular grains similar to those of a pure metal except that each grain has a core of a dendritic or pine tree structure. Each core represents a crystallite that formed in the molten liquid constituting a nucleus upon which was deposited layers of gradually changing composition until the grain was complete. Throughout each grain of alloy as cast, the composition, therefore, varies gradually from the dendritic core to the boundary. When such an alloy is annealed, diffusion takes place establishing uniformity of composition and making the microstructure indistinguishable from that of a pure metal.

Compounds appear under the microscope as crystals bounded by definite crystal faces.

Eutectics. When in any alloy the elements are in such proportions that it has the lowest melting point of any alloy of the same elements, it is called an eutectic alloy. The microstructure of such an alloy consists of alternate particles of two or more of the above described micro-constituents. During the solidification of the alloy, fine particles of two or more constituents crystallise simultaneously from the molten mass thus forming a fine duplex or triplex structure of characteristic appearance. This characteristic structure, due to the simultaneous formation of two or more kinds of crystal, may occur in alloys which have not a lower melting point than any other alloy of the same elements; nevertheless, all such micro-structures should be designated either eutectic or eutectoid structures.

Non-metallic Inclusions. In addition to the above described essential constituents of alloys, non-essential particles are visible in practically all metals and alloys. These foreign particles or non-metallic inclusions consist of oxides, sulphides, silicates, gas bubbles, etc.

Physical Properties. Pure metals are relatively soft and ductile, compounds are hard and brittle, and eutectics and solid solutions have intermediate properties. The physical properties of an alloy, therefore, depend largely on the types, proportions, and grouping of its micro-constituents.

Micro-structures of Iron-carbon Alloys

The ordinary iron-carbon alloys range in composition from pure iron to an alloy containing about 6.7 per cent carbon, and include wrought iron, steel, and white and gray cast iron. When classified according to micro-

structure, steel and white cast iron form a continuous series. White and gray cast iron also form a continuous series, which will be considered later.

Steel-White Cast Iron Series. Alloys of this series, after slow cooling, consist of only two ultimate micro-constituents in various proportions and grouped in various ways. One constituent, known as **ferrite**, is practically pure iron; it is soft (H = 3.6, Moh's), malleable and ductile. The other constituent is iron carbide, Fe_3C, known as **cementite**; it is hard (H = 6) and brittle. Between the molten condition and room temperature, however, a number of transformations take place. All these micro-structures and their inter-relations may be best described with the aid of diagrams such as Figs. 2 to 5 taken with slight changes from "*The Metallography of Steel and Cast Iron*" by Henry M. Howe.

Iron-carbon Equilibrium Diagram. In the diagram, Fig. 2, the lines *ABD* and *AEBC* are called respectively the "**liquidus**" and "**solidus.**"

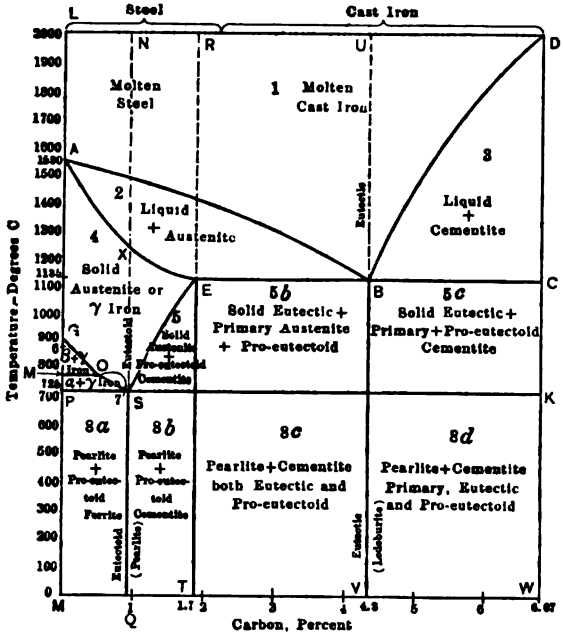


FIG. 2.

All the area above the liquidus represents liquid alloys, the area below the solidus represents solid alloys, and the area between the liquidus and solidus represents alloys that are partly liquid and partly solid. The vertical line LM represents pure iron and the line DW represents cementite. The line UV represents an eutectic alloy, containing about 4.3 per cent carbon and known as **Ledeburite.** When a molten alloy of this composition is cooled

below the temperature represented by the point *B*, it forms a typical eutectic structure composed of cementite and austenite. **Austenite**, a solid solution of carbon (or iron carbide) in iron, has a range of temperature and composition represented by area 4.

Solid alloys, represented by the diagram to the right of the line *RT*, contain more or less of the eutectic, while in solid alloys represented by the diagram to the left of *RT* the eutectic is absent. The line *RT*, therefore, is the boundary between the eutectiferous and non-eutectiferous alloys. It represents approximately, also, the boundary between steel and white cast iron, though there are some commercial steels having over 2 per cent carbon.

Molten alloys of the eutectiferous series, on solidifying, deposit first either cementite or austenite according as the carbon content is greater or less than that of the eutectic. Finally the remaining liquid reaches the composition and temperature represented by the point *B*, when on further cooling the eutectic structure is formed. **Hypo-** or **hyper-eutectiferous** alloys, therefore, just below the solidification temperature consist of the eutectic and excess of austenite or cementite respectively; the proportion of eutectic decreases as the total composition of the alloy departs from that represented by the point *B*. On further cooling, changes take place which will be described after first describing the non-eutectiferous series.

In all alloys represented by the diagram to the left of *RT*, austenite is formed on solidification. Austenite of the composition represented by the line *XS*, on cooling below the temperature represented by the point *S*, is transformed into **pearlite**, an **"eutectoid"** structure composed of ferrite and cementite. According as the alloy has more or less carbon than the eutectoid composition, the austenite on cooling to the lower boundary of area 4 deposits cementite or ferrite respectively until the remainder of the austenite is at the temperature and composition represented by the point *S*; this eutectoid austenite on further cooling is transformed into pearlite.

Not only the austenite of the non-eutectiferous series, however, but also the austenite of the eutectiferous series, is transformed on cooling from the temperature of the solidus to room temperature. The alloys represented by areas 5*b* and 5*c* consist of eutectic and **primary austenite** or **primary cementite** respectively. Both the primary austenite and the austenite of the eutectic have the composition represented by the point *E* when they are at the corresponding temperature. On further cooling, however, this austenite whether it is primary or part of the eutectic, behaves as does the austenite of the non-eutectiferous series; it deposits cementite until the remaining austenite is at the composition and temperature represented by the point *S*, when on further cooling the residual austenite is transformed into pearlite. At room temperature, therefore, the micro-structures of the eutectiferous series are as indicated in areas 8*c* and 8*d*. In such micro-structures the aggregates formed from the primary and eutectic austenite preserve to a large extent their original boundaries and are thus distinguishable. The transformation products of primary and eutectic austenite are, therefore, designated respectively, **primaustenoid** and **eutectic-austenoid.**

Supplementary Diagrams and Table. Fig. 3 represents in diagrammatic form the micro-structures of the steel-white cast iron se asries observed at ordinary temperatures after slow cooling. Fig. 4 represents the percentages of pearlite and free ferrite and cementite, as well as of the various aggregates such as primaustenoid, eutectic, etc.

The constitution of an alloy of any given carbon content is measured by the

intercepts of the various areas on the corresponding ordinate. Table 1 gives the percentages of these constituents and also of the ultimate micro-constituents ferrite and cementite.

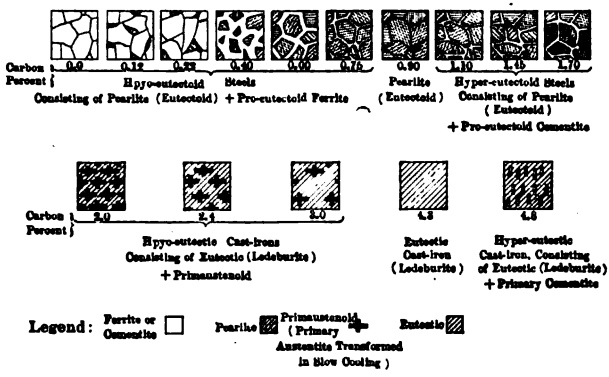


FIG. 3.

Gray Cast Iron Series. Cementite has a tendency to decompose into iron and graphitic carbon according to the equation $Fe_3C = 3Fe + C$. This tendency increases with increase in temperature and increase in the percentages of carbon and silicon; the tendency is decreased by the presence of

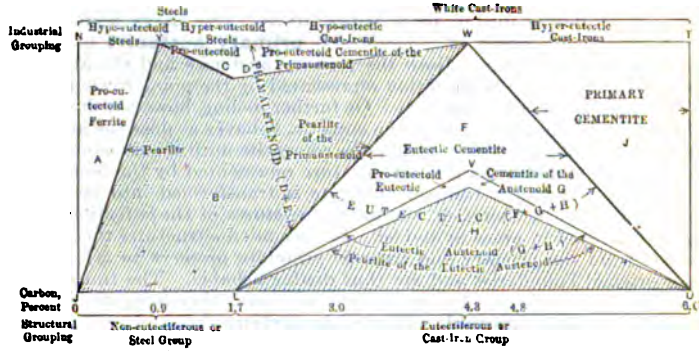


FIG. 4.

sulphur and usually by that of manganese. The carbon set free by this decomposition may take the form of plates or of compact rounded particles, the shape being influenced by the temperature of formation or by the presence of other elements. Details of this series are considered under cast iron, p. 112.

Outline of the Constitution of Iron

Table 1. Theoretical Constitution of the Pearlitiferous Series, Slowly Cooled Carbon Steel, and White Cast Iron

Name		Proximate composition, per cent										
		Carbon, per cent.	Total pearlite	Pro-eutectoid ferrite	Pro-eutectoid cementite	Primaustenoid	Eutectic	Eutectic austenoid	Eutectic cementite	Primary cementite	Total ferrite	Total cementite
Steel	Low carbon	0.0	0	100	0		0	0	0	0	100.0	0
		0.10	11	89	0		0	0	0	0	98.5	1.5
		0.20	22	78	0		0	0	0	0	97.0	3.0
	Medium carbon	0.30	33	67	0		0	0	0	0	95.5	4.5
		0.40	44	56	0		0	0	0	0	94.0	6.0
		0.50	56	44	0		0	0	0	0	92.5	7.5
		0.60	67	33	0		0	0	0	0	91.0	9.0
		0.70	78	22	0		0	0	0	0	89.5	10.5
	High carbon	0.80	89	11	0	100	0	0	0	0	88.0	12.0
		0.90	100	0	0		0	0	0	0	86.5	13.5
		1.00	98	0	2		0	0	0	0	85.0	15.0
		1.10	97	0	3		0	0	0	0	83.5	16.5
		1.20	95	0	5		0	0	0	0	82.0	18.0
		1.30	93	0	7		0	0	0	0	80.5	19.5
		1.40	91	0	9		0	0	0	0	79.0	21.0
		1.50	90	0	10		0	0	0	0	77.5	22.5
		1.60	88	0	12		0	0	0	0	76.0	24.0
		1.70	86	0	13.9	100	0	0	0	0	74.5	25.5
		1.80	84	0	13.6	96	4	2	2	0	73.0	27.0
		1.90	83	0	13.3	92	8	4	4	0	71.5	28.5
		2.00	81	0	13.1	89	11	5	6	0	70.0	30.0
		2.10	79	0	12.8	85	15	7	8	0	68.5	31.5
		2.20	77	0	12.5	81	19	9	10	0	67.0	33.0
White cast iron		2.30	76	0	12.2	77	23	11	12	0	65.5	34.5
		2.40	74	0	11.9	73	27	13	14	0	64.0	36.0
		2.50	72	0	11.7	69	31	15	16	0	62.5	37.5
		2.75	68	0	11.0	60	40	19	21	0	58.75	41.25
		3.00	64	0	10.3	50	50	24	26	0	55.0	45.0
		3.25	59	0	9.6	40	60	29	31	0	51.25	48.75
		3.50	55	0	8.9	30	70	34	36	0	47.5	52.5
		3.75	51	0	8.2	21	79	38	41	0	43.75	56.25
		4.00	46	0	7.5	12	88	42	46	0	40.0	60.0
		4.30	41	0	6.6	0	100	48	52	0	35.5	64.5
		4.50	38	0	6.1	0	92	44	48	8	32.5	67.5
		4.80	32	0	5.2	0	79	39	41	21	28.0	72.0
		6.67	0	0	0.0	0	0	0	0	100	0.0	100.0

CHEMISTRY OF THE PURIFICATION OR CONVERSION PROCESSES

In the various processes for the conversion of pig iron into wrought iron or steel, certain chemical principles are of general application. Pig iron is converted to wrought iron or steel by the removal of some or all of five elements, carbon, silicon, manganese, sulphur and phosphorus. Percentages of these elements which are unobjectionable in foundry irons are detrimental in wrought iron or steel.

In all of these processes, the removal of objectionable elements is effected by their oxidation and by removal of the resulting oxides by vola-

tilisation or by absorption in the slag. The oxidising agent in all cases, is oxide of iron. Though in some cases, an air blast furnishes the necessary oxygen, its action produces oxide of iron, which reacts with the elements that are to be removed, oxidising them and being itself reduced. The essential differences between the various purification processes are differences in the temperature and duration of the reaction and in the composition of the slag.

The temperature of the purification process has a great effect on the relative affinity of oxygen for the elements that are to be removed. At relatively low temperatures oxygen combines with silicon, manganese and phosphorus in preference to carbon. As the temperature rises, however, the affinity of carbon for oxygen increases until it exceeds that of the other elements; at high temperatures, carbon can reduce the oxides of nearly all elements, especially in the presence of a metal such as iron which dissolves the reduced elements.

The composition of the slag greatly influences removal of some of these elements, especially sulphur and phosphorus; the sulphur and phosphorus are readily taken up by a basic but not by an acid slag.

WROUGHT IRON

In contrast to pig iron, which is brittle at all temperatures up to the melting point, wrought iron when heated to redness **can be worked** into almost any desired shape. This difference in properties is due to the greater purity of the wrought iron, and especially to its lower carbon content. While pig iron has about 4 per cent carbon, wrought iron usually has not more than 0.15 per cent. From ancient to modern times wrought iron has been made directly from the ore. As the original low Catalan forges or bloomeries were gradually replaced by shafts of increasing height, manufacture of direct wrought iron gradually ceased. Nearly all wrought iron in the United States is now made from pig iron by the puddling process. A small amount of wrought iron in this country and a large amount in some European countries is made from pig iron by one of the charcoal hearth processes.

Puddling Process. In this process, the pig iron generally used is **"gray forge"** iron. Its composition is: Si, 0.75 to 1.25 per cent; S, 0.04 to 0.1 per cent; P, 0.3 to 0.4 per cent; Mn 0.5 to 1.0 per cent. The **furnace** is of the reverberatory type. The hearth is usually about 4 by 6 ft. and holds about 500 lbs. pig iron, though some furnaces have a capacity of about 1500 lbs. The hearth is lined with oxide of iron in the form of mill cinder, ore or scale, which furnishes some of the required oxygen. The purification of the iron takes place in three stages. In the **melting stage**, about ½ hour, most of the silicon and manganese and some of the phosphorus are removed. In the **clearing stage**, about 10 minutes, iron oxide in the form of scale or high-grade ore is added and the charge is stirred and cooled. At the lowered temperature, the highly basic slag oxidizes and absorbs the other impurities in preference to the carbon. Carbon then begins to be oxidised with liberation of carbon monoxide, causing the "boil." During the **boil** which lasts about 25 minutes, the mass is stirred or **"rabbled"** with a hoe shaped implement known as the "rabble." With removal of the carbon, the melting point rises as indicated in Fig. 2, until finally a pasty mass is formed when the iron is said to have **"come to nature."**

The pasty iron is now collected into balls weighing about 80 lbs. and either squeezed in a rotary squeezer or hammered to squeeze out the excess slag. The **blooms** are then rolled out into bars about 3 to 6 in. wide, ¾ in. thick

and 15 to 30 ft. long, known as "**muck bar.**" For further removal of slag, the muck bars are cut into pieces about 3½ ft. long, wired together in **fagots**, reheated to welding heat, and rolled into bars, sheets, plates, etc. It is now known as "**merchant iron**"or "**single refined iron.**" Sometimes "**merchant bars**" are cut, piled, reheated and rolled to form "**double refined iron.**"

Wrought iron made from the product of the charcoal blast furnace is known as "**charcoal iron.**" It is of better quality than iron from coke pig, which is known as "**common iron.**" In the United States much wrought iron is made by piling wrought iron scrap, heating to welding temperature, and rolling into bars called "**puddle bar.**" Sometimes small pieces of wrought iron and steel known as "**busheled scrap**" are rolled with larger pieces to form bars of an inferior grade of wrought iron.

Charcoal Hearth Processes. In the Styrian, Walloon, Lancashire and other processes, charcoal pig iron is heated in a low hearth with charcoal as fuel. The pig iron, though it rests on charcoal, is surrounded by an oxidising atmosphere due to the air blast from one or two tuyeres. During the gradual melting the impurities are largely removed by oxidation and the iron covered with slag falls into the hearth where further purification takes place during the stirring of the pasty mass. It is then raised above the tuyeres, remelted and collected in the hearth. Since most of the slag is now on the surface, the mass is not stirred unless it is to be remelted a second time above the tuyeres. Wrought iron made by a charcoal hearth process contains less slag than puddled iron. The best quality of wrought iron, such as the well-known **Swedish iron** and the "**knobbled charcoal iron,**" is made by charcoal hearth processes.

Properties of Wrought Iron. Wrought iron **melts at** approximately 2700 deg. F. At white heat it is readily **welded.** It has the following **physical properties:** Tensile and Compressive strength 40,000 to 54,000; elastic limit in tension about 30,000, in torsion about 20,000; modulus of elasticity in tension 28,000,000, in torsion 12,800,000; elongation in 8 in., 20 to 28 per cent; reduction of area 30 to 50 per cent. It has a well defined yield point and **fibrous fracture.** Since it **resists corrosion** better than steel, it is much used in machines or structures where corrosion is of importance. When polished wrought iron is examined microscopically, many particles of slag are visible. In addition to the slag particles, the etched surface also shows the **micro-structure** of low carbon steel as illustrated in Fig. 3.

Marine Uses. Wrought iron was formerly used in moving parts of machinery requiring strength and toughness. It has now been almost entirely replaced by steel. It is still used to some extent in boiler staybolts and braces, rivets, lapwelded boiler tubes, etc. It is still used for shafting of many merchant vessels. Wrought iron is used for plates which require ability to resist corrosion. It is also used for the manufacture of the best tool steel by the crucible process. Until the recent development of cast steel anchor chains. these were made of wrought iron.

STEEL.

The Bessemer Process

Until the invention of this process in 1856, practically all steel was made by the recarburisation of wrought iron by either the cementation or crucible process. Departing from the chronological order, however, a description will first be given of the Bessemer process. In this process, molten pig-iron is

subjected to the action of an air blast which oxidises the impurities. Although the melting point rises about 400 deg. C. in the course of the purification, the heat of combustion of the impurities is sufficient to keep the metal molten.

The molten iron is contained in a **"converter,"** a pear-shaped vessel open at the smaller end and supported on trunnions perpendicular to its long axis. In the large end, or sometimes in the side, are **tuyères** through which the air blast enters. In **"blowing,"** the converter is turned so that the small end is upward; in **charging** it is turned so that the molten metal does not cover the tuyères before the blast is turned on. The lining is siliceous or basic according as the **acid Bessemer** or the **basic Bessemer** process is to be used.

Acid Bessemer Process. In this process, the converter is lined with siliceous material, which is unaffected by the acid silicate slag produced by the oxidation of the silicon and other impurities of the pig-iron. Since, under these conditions, no phosphorus and sulphur are removed, "Bessemer pig-iron" must not contain over 0.09 per cent phosphorus and 0.05 per cent sulphur. Since the oxidation of silicon is the chief source of heat, the iron must contain not less than about 1 per cent of this element; it may contain as much as 2.5 per cent. Though the requirements for manganese are not very strict, the usual maximum in American and English practice is about 0.8 per cent, while in Swedish practice about 2.5 per cent may be used.

The "Blow." In the Bessemer process, usually no attempt is made to stop the blow before practically all the impurities have been removed. The order of oxidation of the various impurities, depends not only on their relative chemical affinities, but also on their relative quantities and on the temperature. Since, the element iron is the chief chemical constituent of pig-iron, iron oxide is the chief oxide formed by the direct action of the air blast. As explained on p. 88, the iron oxide acts as a carrier of oxygen to the other elements, oxidizing them and being itself reduced. In the first five minutes of the blow, known as the **slag forming period,** most of the silicon and manganese are thus oxidized and unite with some iron oxide to form a slag. The oxidation of the carbon then begins, carbon monoxide being formed in large quantities and burning at the mouth of the converter. In this period, known as the **"boil,"** the flame, which is at first short and orange colored, lengthens and becomes white and dazzling. When the carbon, with the remainder of the manganese and silicon, have thus been removed, the **flame "drops."** The converter is then turned down and the blast is stopped.

Control of Temperature. If, during the blow, the metal gets too hot, the boil becomes too active, causing troublesome spattering and over-oxidation; if the temperature falls too low, the metal cannot be poured readily into moulds. **To lower the temperature,** solid scrap may be added, or steam may be blown into the converter. **To raise the temperature,** the converter may be tilted until some of the tuyères are above the surface of the metal; this results in oxidation of iron and carbon monoxide and the production of additional heat within the vessel. In America, though pig-iron containing relatively low percentages of silicon and manganese is used, the temperature is kept up by running the heats in such rapid succession that the loss of heat from the converter is minimized.

Deoxidation. "Blown metal" is **"red short"** on account of excess of iron oxide. It evolves gases in such quantities that it cannot produce sound ingots or castings. To remove the excess oxygen, therefore, **deoxidizers** are added. Of these, the most important are **ferro-manganese** and **spiegel-eisen.** Ferro-manganese, since it contains about four times as much

manganese and about the same amount of carbon as spiegel-eisen, is used when low carbon steel is to be made. Since spiegel-eisen, must be used in relatively large quantities, it is melted before addition. **Ferro-silicon** and **silico-spiegel** are also frequently used as deoxidants. Though occasionally **aluminum** is used, its use is limited, since its oxide tends to remain in the metal as small, hard particles which weaken the metal.

Recarburization. Some of the above mentioned deoxidants, especially ferro-manganese and spiegel-eisen serve also to raise the carbon content of the metal to the required point. For this reason, they are usually known as **"recarburizers."** If additional carbon is needed, low phosphorous pig-iron, coke or anthracite may be added.

Basic Bessemer Process. In this process, lime is added to the molten pig-iron in order to form a basic slag. To prevent corrosion, the converter lining must also be basic. The basic slag in the presence of iron oxide is able to remove most of the phosphorus as calcium phosphate. Though part of the sulphur also is removed, its removal is less certain than that of phosphorus. As in the blast furnace process, the sulphur appears to be removed in the form of calcium sulphide. To prevent corrosion of the basic lining by silica, the percentage of silicon in the pig-iron should be as low as possible, and not more than 1 per cent. To compensate for the loss of this heat forming element, the percentage of phosphorus in basic Bessemer pig-iron must be high, from 2.5–3.0 per cent. Since there are no low-silicon, high-phosphorus ores in the United States, the basic Bessemer process is not used in this country. Germany is the only country that makes extensive use of the process.

The "blow" is similar to that of the acid process except that it is continued for three or four minutes after the drop of the flame. In this "after-blow" most of the phosphorus and some sulphur are removed. The end of the "after-blow" is determined by the appearance of the fracture of test specimens taken from the converter. The recarburization is similar to that of the acid process except that the recarburiser is added not to the converter but to the ladle.

Bottom and Side-blown Converters. In the manufacture of steel to supply rolling mills, a bottom-blown converter holding about 20 tons is used; this is able to take care of the product of two blast furnaces. Molten pig-iron direct from the blast furnace is placed in a large vessel called a **"mixer;"** from there it is drawn as needed for the converter. For making steel castings, a converter holding one or two tons is used. Though sometimes this **"baby Bessemer" converter** is bottom blown, it is more usually side blown. To supply these converters, pig-iron is melted with coke in a cupola.

In the bottom-blown converter, the air bubbles through the metal, stirring it and oxidising it uniformly throughout. In the side-blown converter, the air blast impinges on the surface of the slag and oxidation of the metal occurs chiefly near the top. For this reason, the course of the "blow" is not so regular in the side-blown as in the bottom-blown converter; there may be several periods of "boil" and "drop of the flame" before purification is complete.

Uses of Bessemer Steel. It is generally recognized that Bessemer steel is inferior in quality to open-hearth steel. For this reason its use in steel rails is rapidly decreasing. Though used in structural shapes for buildings it is no longer permitted in structural shapes for bridges and ships. In the Navy, its use is limited to steel castings, material for boiler plates, bolts and rivets, etc., where great strength is not required.

The Open Hearth Process

In this process, the metal is purified in the shallow hearth of a reverberatory furnace of the regenerative type. At one end of the furnace are two chambers filled with checker work of fire brick, in which the air and gas are preheated on their way to the furnace. At the other end are similar chambers in which the products of combustion give up a large part of their heat on the way to the stack. By frequent reversal of the flow of gases, the temperature of the entering gases is kept at about 1800 deg. F. In some plants, natural gas or oil are used as fuel; in that case, only the air is preheated. Recently also powdered coal has been successfully used as fuel.

The oxygen for the oxidation of the impurities in the metal comes partly from the air and partly from the iron ore. The purification depends also largely on the character of the slag. In the basic process, since a basic lining is used, the slag may be made so strongly basic that it can absorb nearly all the phosphorus and a large part of the sulphur. In the acid process, since the lining and consequently the slag are acidic, neither of these elements is removed.

The standard open hearth furnace has a capacity of 50 to 80 tons. In a 50 ton furnace, the hearth is about 30 to 35 ft. long, 12 to 15 ft. wide, and 18 to 24 in. deep. For making steel castings and special steels smaller furnaces may be used. Though 15 tons is the smallest economical size, furnaces of only 5 tons capacity are used. Some furnaces are so constructed that the hearth can be tilted enough to pour out the finished metal; all trouble with a tapping hole are thus avoided.

Acid Process. In the United States, the charge consists of two-thirds to three-fourths scrap and the rest pig iron. The carbon, silicon and manganese in the charge may vary within wide limits. Since no phosphorus and sulphur are removed in this process, the charge must not contain more than about 0.935 per cent phosphorus and 0.04 per cent sulphur. The percentages of these two elements will increase somewhat during the purification on account of the removal of silicon, manganese, etc. The percentage of sulphur is also increased slightly by absorption from the fuel, if producer gas is used.

If the charge has been properly proportioned, most of the silicon and manganese and about two-thirds of the carbon will have been removed during the melting. The oxides of silicon and manganese, together with some iron oxide, combine to form a slag which floats on the surface of the metal. To reduce the percentage of carbon to the required value, and to remove the remainder of the silicon and manganese, ore may now be added. This oxidises the carbon to carbon monoxide which bubbles through the molten mass and escapes. The reaction is allowed to proceed until the carbon is slightly below the required value. No ore is added during the last hour or two lest the metal be over oxidised.

As in the Bessemer process, the steel at the end of the purification contains excess of oxides which must be removed by the addition of deoxidants. The carbon and other constituents must also be raised to the desired percentages. Both these reactions are effected by the addition of recarburizers as in the Bessemer process.

Basic Process. In this process, limestone or burnt lime is added to form a basic slag. If the basicity is kept high enough, it is thus possible to remove nearly all the phosphorus even when the percentage in the charge is as high as 2 per cent. Since the removal of sulphur is less complete than that of phosphorus the charge should not contain more than 0.05 or 0.06 per cent of this element. The silicon in the charge should not be higher than 1 per

cent, otherwise an excessive quantity of lime will be required. It is advisable to have at least 1 per cent of manganese in the charge, since this element tends to prevent over-oxidation and aids in the removal of sulphur. The proportions of pig iron and scrap vary greatly.

If the quantity of lime is large enough in the beginning, the greater part of the phosphorus will be removed during the melting. After the melting is complete, ore is added as required to remove carbon and the remaining impurities. The removal of the phosphorus is usually incomplete until the carbon has reached a low percentage. In this respect the open hearth process occupies an intermediate position between that of the puddling and basic Bessemer processes. Even when high carbon steel is to be made, therefore, it is usually necessary to reduce the carbon percentage to a low value and then add sufficient recarburiser to bring the carbon up to the required percentage. Sulphur, in the open hearth as in the blast furnace process, is removed chiefly as calcium sulphide. **The more basic and the more liquid the slag, the more efficient it is in the removal of sulphur.** For this reason, compounds such as calcium chloride or calcium fluoride are sometimes added to increase the fluidity of the slag without increasing its basicity.

The usual range of slag composition in the acid and basic open hearth processes is shown in Table 2.

Table 2.—Slag Composition

	Acid open hearth, per cent	Basic open hearth, per cent
SiO_2	48 to 55	10 to 25
P_2O_5	.....	5 to 15
FeO	23 to 34	10 to 25
MnO	13 to 21	.....
CaO	.....	35 to 50
MgO	.....	6 to 10
Al_2O_3	.....	.....

In the acid slag, the percentage of silica normally never varies far from 50; iron and manganese oxides together make up nearly all the remainder.

The **time required for the melting and purification** of the metal is from 10 to 16 hours. This time may be shortened by increasing the ore additions and, at the end of the purification, adding an increased quantity of recarburiser. This hastening, however, is at the expense of the quality of the metal. To produce good results, time is required for the oxides and other impurities in dilute solution to react on each other. The long time required for this reaction to take place is the chief reason for the superiority of open hearth over Bessemer steel. Since the conditions in the acid process do not produce so oxidising a slag as in the basic process, acid steel is usually better than basic steel in spite of the fact that the basic steel is usually freer from phosphorus and sulphur. For example, the specifications of the A.S.T.M. for many kinds of steel allow 0.01 per cent more phosphorus in acid than in basic steel.

There are a number of modifications of the open hearth process. Of these the **Talbot process** is one of the most important. In this process, a basic lined, tilting furnace of over 200 tons capacity is used. The operation is continuous, the furnace being emptied only about once a week. After purification of the first charge as usual, the greater part of the slag is poured off; then about a third of the metal is poured into the ladle and recarburized. To the remaining metal, iron ore and limestone are added to form a basic, highly oxidised slag; pig iron is then added to replace the metal that has been removed.

Although the reaction is very vigorous, the mixture does not boil over, since the pig iron is considerably diluted by the previously purified metal. In this process, therefore, steel can be made from pig iron alone. The rapidity of the purification makes it possible to obtain three or four heats of about 75 tons each in 24 hours.

There are processes in which the metal is given a **preliminary treatment** to remove part of the impurities and is freed from the preliminary slag. It is then given **final purification** either in the same hearth or after transfer to a different hearth. Of these processes, the most important are the **Campbell No. 1, the Campbell No. 2, and the Bertrand-Thiel processes.** These will not be described.

Duplex Processes. When an ore has too high a percentage of silicon for use in the basic open hearth process, it is sometimes given a preliminary treatment in the acid Bessemer converter to remove the greater part of the silicon and manganese. The metal in the molten condition is now transferred to the basic open hearth and purified in the usual way. Although the cost of the duplex process is greater than that of either the Bessemer or open hearth processes alone, the increased cost is made necessary in some localities such as Alabama, where ores and coke are both of such quality that it is difficult to make pig iron that will conform to the requirements of the basic open hearth process. The duplex process makes it possible to use pig iron of high silicon content. As explained on p. 81, such pig iron, if properly made, has a low percentage of sulphur.

The Cementation Process

As used for several centuries, this process consists in heating wrought iron in contact with charcoal. The wrought iron in the form of flat bars is embedded in charcoal in cast iron boxes about 8 to 15 ft. long, and 2½ ft. wide and deep; the material is then heated in furnaces at about 1000 deg. C. The carbon of the charcoal gradually penetrates the iron to a depth depending on the temperature and time; for complete carbonization, 7 to 11 days are required. The progress of the reaction is observed by removing test pieces, breaking them and noting the appearance of the fracture. The bars are then allowed to cool in the furnace.

The bars now have a blistered appearance, on account of which the material is called **"blister bar"** or **"blister steel."** In order to make the carbon distribution more uniform, the bars are usually piled in fagots, reheated and rolled. The material is now called **"single shear steel."** Sometimes these bars are again piled, reheated and to form **"double shear steel."**

Since this process requires so long a time, and since even the "double shear steel" is not of uniform carbon content, the cementation process has been gradually replaced by the crucible process.

For surface carbonization or **"case hardening,"** however, the cementation process is still of great use. Details of this process will be given later.

The Crucible Process

This method has been in use in the Orient since ancient times. In England the process was first used by Huntsman in 1740. He melted blister bars in crucibles and thus produced a much more uniform material than could be produced by the cementation process alone. The process in this form is still in use at Sheffield, England. In the United States, however, the long preliminary cementation is omitted; the wrought iron is simply melted in crucibles with the requisite amount of charcoal or other carbonizing material.

The crucibles hold about 50 lb. each. In England, they are made of fire clay; in the United States, they are usually made of a mixture of fire clay and graphite. The fuel used in England is generally coke; in the United States coal or gas is commonly used, though there are some oil burning furnaces. If gas is used, the furnace is of the reverberatory type.

Since the crucibles are kept tightly covered, the atmosphere in contact with the steel, though at first oxidizing, soon becomes neutral or even reducing. One of the great advantages of this process, therefore, is that plenty of time may be given for the removal of oxides and other non-metallic inclusions. In other words, the reaction between iron oxide and carbon proceeds nearly to completion. Under these conditions, the carbon of the steel also reduces more or less silica to silicon, which is absorbed by the steel. Crucible steel, therefore, usually contains from 0.1 to 0.5 per cent silicon; this tends to prevent the formation of blow holes in the castings. If the "killing" of the steel is allowed to proceed too far, however, the steel becomes brittle from excess of silicon.

While in Sheffield **charcoal hearth iron** is used for the best grade of tool steel, in the United States **puddled iron** is generally used for this purpose. The use of basic open hearth scrap in the crucible process results in an inferior material. The reason for this inferiority is not clearly understood, since basic open hearth scrap may be apparently purer than wrought iron. Though open hearth scrap is not suitable for the manufacture of tool steel, it is good enough for the production of steel castings for many purposes. Crucible steel, if properly made, is superior to any except electric furnace steel.

The Electro-thermic Process

Electro-thermic heating has important **advantages** over other methods. In the electric furnace, the temperature can be brought high enough to maintain fluidity in a slag richer in lime than are the slags used in other steel making processes; this facilitates the removal of impurities from the metal. Moreover, since combustion is not the direct source of heat, the metal can be kept free from the contaminating effect of the products of combustion. An oxidizing atmosphere within the furnace can also be avoided.

The electric furnace has its **chief field of usefulness** in the production of steel from high grade scrap and pig iron, or in the final purification of steel that has been partially purified by one of the other steel making processes. Since electric energy as a direct source of heat usually costs more than fuel, the electric furnace is seldom used in the production of inferior grades of steel. In some localities, however, where electric power is cheap, steel may be produced from inferior grades of pig iron and scrap, and pig iron may even be produced from the ore.

Electric furnaces may be classified according to the method of application of the electric energy. The **heat may be supplied** entirely by radiation from the electric arc, or partly by radiation from the arc and partly by the passage of the electric current through the slag and metal. Or the heat may arise from a current induced in the metal itself.

In the **Stassano furnace**, the metal is heated by radiation from arcs passing between electrodes inserted diagonally through the sides of the furnace above the hearth. Generally three phase current is used, the three electrodes being symmetrically placed about the cylindrical furnace.

In the **Heroult furnace**, two or more electrodes are inserted vertically through the roof. The current passes by arc from one or more of these electrodes down through the slag to the metal beneath, thence through the

metal and by one or more arcs to the remaining electrodes. In this furnace, therefore, the current passes through two arcs in series.

In the **Girod furnace**, the current enters through one or more electrodes inserted vertically through the roof; thence it passes through one or more arcs in parallel to the surface of the metal and down through the metal to electrodes inserted through the bottom of the furnace.

In **induction furnaces** such as the **Kjellin**, no electrodes are used. A current is induced in a closed circuit of molten metal so arranged that it constitutes the secondary coil of a transformer.

This circuit of metal encloses an arm of the magnetic circuit about which is wound the primary coil carrying alternating current. As in ordinary transformers, the voltage of the induced current varies inversely, and the strength of the induced current varies directly as the number of turns in the primary coil. A disadvantage of this furnace is the ring-shaped hearth, which is unsuited for the purification of metals. For this reason the furnace is suited only for the melting of metal when little or no purification is required.

The disadvantages of the ordinary induction furnace have been overcome in the **Röchling-Rodenhauser furnace**. In this furnace each arm of the magnetic core is surrounded by a ring-shaped hearth; the two rings meet so as to form a central hearth large enough to permit of the purification of the metal therein. This central hearth receives additional heat from current induced in a secondary coil surrounding each arm of the magnetic circuit. The furnace is thus a combination induction and resistance furnace.

The **Gin furnace** consists of a long trough, usually of winding shape so as to permit of greater length, through which current passes and furnishes the required heat. On account of the unsuitability of this shape for metallurgical work, this form of furnace is little used in steel making.

Acid lined furnaces are used in simple remelting processes, in which no removal of phosphorus or sulphur is required. In such furnaces, scrap and pig iron of a high degree of purity are melted, brought to the required percentages of carbon, silicon, and alloy constituents, and held until deoxidation has been completed. The atmosphere of the electric furnace, unlike that of the open hearth, can be made neutral or even reducing in character, and the metal can be held in this atmosphere as long as desired.

Basic lined furnaces may be used for removing phosphorus and sulphur as well as other elements. For this purpose, it is better to use metal that has been treated by one of the other steel making processes so that the electric energy may be utilized only for the final purification of the metal. Preferably also the metal should be charged in the molten condition in order to conserve electric energy.

Chemistry of the Electro-thermic Process. The high available temperature and the ready control of the atmosphere within the furnace make the electro-thermic process of great value in the production of steel that is relatively free from phosphorus, oxides and sulphides. The conditions required for the removal of phosphorus differ entirely from those required for the removal of sulphur. While phosphorus is removed in the form of phosphates and therefore requires oxidizing conditions, sulphur is removed as calcium sulphide and therefore requires reducing conditions. For these reasons, it is necessary, after adjusting the conditions for the removal of one of these elements, to remove the slag containing this element and change the conditions so as to favor the removal of the other element.

Usually the phosphorus is removed first. If sulphur is to be removed first,

the metal must be deoxidised and freed from sulphur, reoxidised for the removal of phosphorus, and again deoxidised and recarburised. This procedure would require an excessive quantity of ferro-silicon. The prior removal of the sulphur is economical only when low silicon steel is to be made from metal that has been deoxidised by one of the other steel making processes.

Instead of treating the metal in the electric furnace with two or more slags as described above, the use of duplex or triplex processes is finding increasing favor. This procedure involves a preliminary treatment of the metal either in the basic open hearth or in the Bessemer converter followed by the basic open hearth. The greater part of the impurities are thus eliminated before the metal is transferred to the electric furnace.

Steel properly made by an electro-thermic process has better physical properties than that made by any other process. On account of its freedom from non-metallic inclusions its endurance when subjected to repeated or alternating stresses is especially good.

The Bell-Krupp Process

This is one of the most important processes for the partial purification of pig iron. In this process, molten pig iron is stirred in contact with iron oxide. The temperature is kept so low that the slag, rich in iron oxide, oxidises and removes from the metal most of the silicon and phosphorus without removing much of the carbon. The iron ore is heated in a slightly inclined revolving hearth furnace. Liquid pig iron is then added. After about five minutes, most of the silicon and phosphorus have been removed and the oxidation of the carbon begins. The furnace may be tapped at this point or, if more thorough removal of phosphorus is desired, the process may be continued for three to five minutes longer.

In the product, known as **"washed metal,"** the sulphur and phosphorus percentages do not usually exceed 0.03 and 0.025 respectively. Washed metal is used in crucible steel making and in foundry practice as a means of adjusting the carbon content in castings.

INGOTS

Pipes. Since steel on solidifying decreases in volume, and since the upper and inner portion of a top poured ingot is usually the last to solidify, the upper surface of the metal usually settles and forms a **"pipe."** If this downward flow is prevented, one or more closed cavities occur along the axis of the ingot. To diminish piping, many devices are used to delay the solidification of the upper part of the ingot and thus permit downward flow of the metal. The "pipe" is **diminished by** using a tapered ingot mold with large end up.

Blow holes in steel **are due to** the entrapping of gases evolved from the solidifying metal. These gases are usually due to the interaction of oxides with the carbon of the steel to form carbon monoxide. Though sometimes blow holes are purposely allowed to form in order to diminish piping, this method of prevention results in an inferior quality of steel. Usually blow holes are eliminated as much as possible by the addition of **deoxidising agents** such as ferro-manganese, ferro-silicon, ferro-titanium, aluminum, etc., as described under the various processes of manufacture.

Segregation. As illustrated by the iron-carbon equilibrium diagram, Fig. 2, during solidification the solid metal contains less carbon than the molten metal with which it is in equilibrium. The percentages of solid and liquid respectively at any temperature are represented by the abscissæ of isothermal points on the "solidus" and "liquidus." Since the residual

liquid becomes continually richer in carbon, there is segregation of this element in the regions near the top and center of the ingot. For similar reasons there is segregation in the same region of impurities such as phosphorus and sulphur. Since this region contains defective material, the top portion of the ingot is removed whenever possible by **"cropping."** Although the most defective metal is removed by this method, there still exists a certain amount of axial segregation resulting in deficiency in physical properties in that part of the ingot.

MECHANICAL TREATMENT

Ingots formed as described above are subjected while hot to various mechanical processes, such as **rolling, hammering or pressing,** which not only give the required shape to the metal but also improve its physical properties by breaking up the coarse crystallization of the original ingot. Steel objects of relatively simple shape, such as rails, when required in large quantities, are formed by rolling. When the required shape is complex, or when a superior quality of material is required, the metal is forged by hammer or press. **Pressing is more effective than hammering,** since it works the metal to a greater depth.

A minimum amount of working of metal is required in specifications for all important forgings. U. S. Naval Specifications require that the area of cross section of the rough forging shall be not more than one-fourth that of the original ingot. The amount of reduction of cross section, however, should be adapted to the use to which the forging is to be put; excessive reduction merely increases the physical properties in a longitudinal at the expense of those in a transverse direction.

The best **finishing temperature** for hot working is just above the transformation range as shown in Fig. 2. With increase in finishing temperature above the transformation range, there is a corresponding increase in the grain size of the finished product. On the other hand, if the forging process is continued below the transformation range of temperature, it results in grain distortion and other embrittling effects of cold working. For some purposes, such as wire, however, the metal is finished by **cold drawing** so as to produce high tensile strength even at the expense of considerable loss of ductility. At a "blue heat" (so named from the color of the oxide film), about 400 to 500 deg. C., steel is very brittle and cannot be forged without developing cracks.

As shown by the microscope, the grain distortion caused by cold working is made possible by the slipping of the metal along various planes bearing definite relation to the axes of crystal symmetry. These **slip lines** vary in number and prominence with the degree of distortion of the metal. According to a widely accepted theory, **amorphous metal** is formed along these slip lines. The presence of this amorphous metal, which is hard and brittle, accounts for the hardening and embrittling effect of cold working.

HEAT TREATMENT

Grain Refinement. The heat treatment of steel depends on the changes produced by heating the steel above the transition range into the austenite region as shown in Fig. 2, and on the reverse changes produced by varying the rates of cooling from the austenite into the pearlite range of temperature and composition. To remove the coarse structure of a casting or forging, the metal is heated until its temperature is such that it enters the austenite field. In passing through the transformation range into this field, the pearlite is changed into austenite and the excess ferrite or cementite is gradually dis-

solved. As soon as austenite is formed, the grains of this constituent begin to grow; if therefore, the temperature is carried too far or maintained too long above the transformation range, the austenite grain size, and the grains formed from the austenite on cooling, will be unnecessarily coarse. In general, the temperature should not be more than about 75 deg. F. above the theoretical temperature as shown in Fig. 2; this temperature, moreover, should be held only long enough to complete the transformation into austenite. The metal is then cooled at a rate which depends on the physical properties desired.

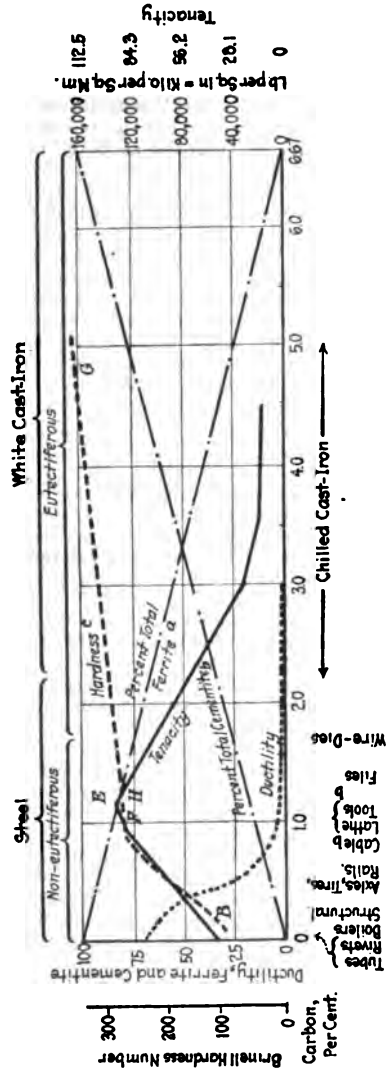


FIG. 5.

Annealing. Slow cooling from above the transition temperature produces pearlite either alone or with free ferrite or cementite according as the metal is of eutectoid, hypoeutectoid, or hypereutectoid composition. Annealing **results in a maximum of ductility and a minimum of tensile strength and elastic limit;** the values of these properties in annealed steel depend on the chemical composition. In carbon steels, the carbon content is the determining factor. The effect of this element on the hardness, tenacity and ductility is illustrated by Fig. 5 taken from "The Metallography of Steel and Cast Iron" by Henry M. Howe. Fig. 5 should be studied in connection with Fig. 4.

Hardening. When steel, after grain refinement as described above, is cooled rapidly through the transformation range, the transformation from austenite to pearlite is partly suppressed. It is impossible to prevent this transformation entirely in carbon steels; if more

than 1.5 per cent carbon is present, however, some of the austenite may be preserved unaltered. Alloying elements, also, such as manganese, nickel, etc., if present in sufficient quantity, entirely suppress the transformation. Under ordinary conditions, however, quenched steel contains no austenite. The types of microstructure resulting from cooling at various rates are given various names, such as **martensite, troostite, sorbite, sorbitic pearlite,** etc. The name **"osmondite"** has been used to designate a supposed structure between that of troostite and sorbite; various compound terms, also, such as **"troosto-sorbite"** are in use.

The writer believes that unnecessary confusion has resulted from the attempt to draw distinction between structures separated by no apparent dividing line. On an etched bar of high carbon steel, one end of which has been quenched in water, there is discernible a sharp line separating the structure designated "martensite" from that designated "troostite." From this line of demarcation which is visible to the unaided eye, there is a gradual change from "troostite" to laminated pearlite. Under the microscope, there are visible at this line of demarcation black lines in certain cleavage planes of the martensite and black nodular masses on the grain boundaries. These lines and masses merge gradually as the troostite field is entered until the whole structure consists of the dark colored constituent. Between this constituent and laminated pearlite there is no apparent dividing line.

Essentially, therefore, the constituents of hardened steel fall into two groups, the austenite-martensite group and the troostite-pearlite group. In the former group there is a gradual change in structure from austenite to martensite. While austenite has the usual structure of an annealed solid solution or pure metal, quenched carbon steel rarely exhibits this structure unaltered. In such material lines have appeared within each grain in three directions corresponding to the **cleavage planes** of the original austenite grains. These lines, which frequently have the appearance of needle shaped crystals, are considered by some authorities to be merely **twinning planes** in the austenite. A structure in which these lines are visible in greater or less number and prominence is known as martensite. Its properties are very different from those of the original austenite; while the unaltered austenite is comparatively soft and ductile, the appearance of the tri-directional markings coincides with great increase in hardness and the development of brittleness.

In the troostite-pearlite series, troostite (which may be considered to be emulsified ferrite and cementite as first formed from martensite) is next to martensite in hardness. The hardness gradually diminishes with coagulation of the pearlite until in laminated pearlite we have only the hardness of annealed steel. At one end of the series, therefore, we have troostite; at the other end, laminated pearlite.

The **hardest tool steel consists of** martensite at the point of transformation into troostite. It is readily produced by quenching medium or high carbon steel in water. Oil quenching produces a softer structure containing less martensite and more troostite. Still slower rates of cooling, such as air cooling, produce the more or less coagulated constituents usually designated "sorbite" or "sorbitic pearlite."

Tempering. Instead of varying the rate of cooling to produce these metastable structures, similar results may be produced by water or oil quenching followed by reheating or "tempering" at temperatures depending on the properties desired. The higher the reheating temperature the farther the transition proceeds from austenite toward pearlite. Although this reheating temperature is best determined by use of a pyrometer, it has been customary

to estimate it by means of a scale of "temper colors." These colors, with the approximate corresponding temperatures, are given in Table 3 below. It should be noted, however, that the temper colors depend not only on the temperature but also on the time of heating as shown in Table 4.

Table 3. Temper Colors and Temperatures

Color	Temperature, degrees C.
Light straw	220 to 230
Dark straw	240
Yellowish brown	255
Reddish brown	265
Purple	275
Violet	285
Cornflower blue	295
Pale blue	310 to 315
Sea green	330

Table 4. Temper Colors As Affected By Time

Temperature, degrees C.	Time of heating until color appears, minutes				
	Straw	Brown	Purple	Dark blue	Pale blue
200	6	49			
220	3	33	63		
250	1	10	39		
275	½	3	11	27	40

Tempering of hardened steel is usually necessary to relieve internal stresses as well as to produce alterations of structure. Such internal stresses are caused by the fact that the transformation products occupy greater volume than the austenite. If these stresses are not relieved they may lead to rupture of the metal either spontaneously or under the application of stresses in service.

Heat Treatment of Machinery Steel. The process of quenching and tempering is applied in the heat treatment not only of tool steel but also of many grades of carbon and alloy steels. As applied to steel other than tool steel it is often incorrectly designated "**double annealing.**" The term "tempering" rather than "annealing" seems applicable to these steels as well as to tool steels; the term "annealing" as applied to steel should be reserved to mean heating above the critical range followed by slow cooling. On applying this heat treatment, the grain refining temperature may be usually obtained by reference to Fig. 2, adding about 75 degrees F. to the theoretical temperature. The rate of accelerated cooling (water, oil or air) will depend on the size of the object; in general the cooling should be as rapid as possible with avoidance of cracking due to internal stresses. The **reheating temperature** will depend on the physical properties desired.

A. S. T. M. recommendations in regard to heat treatment of carbon steel have been condensed into the following paragraphs. Except for condensation and omission few alterations have been made in the language. To distinguish this material, quotation marks are used.

Rolled and Forged Objects; Methods of Heating. In case of large objects, the heating of the interior of which lags behind that of the outside, the final approach to the desired temperature should be slow. An exposure

of one hour at that temperature should be long enough for pieces 12 inches thick; thicker pieces need longer heating."

Grain Refining Temperature. "In general, the higher the carbon content, the lower should be the grain refining temperature; hence different temperatures are given for different ranges of carbon content. For each range of carbon content a range of temperature is given, the upper limit of which applies to larger objects and to the lower part of the corresponding range of carbon content. In Table 5, the temperatures are adapted for steel of the usual moderate manganese content. For steels with a manganese content higher than 0.75 per cent slightly lower temperatures suffice."

Table 5. Recommended Grain Refining Temperatures

Range of carbon content, per cent	Range of temperature, degrees C.
Less than 0.12	875 to 925
0.12 to 0.29	840 to 870
0.30 to 0.49	815 to 840
0.50 to 1.00	790 to 815

Cooling. "After the object has been held at the grain refining temperature long enough to make the temperature nearly uniform throughout and to complete the refining of the grain, it should be cooled in a way suited to its carbon content and to giving it the specific properties needed. To give the greatest softness and ductility, even at a certain sacrifice of strength and elastic limit, the metal should be cooled slowly either in the furnace or under cover of non-conducting material. To give great tensile strength and high elastic limit, even at a certain sacrifice of ductility, the cooling should be more rapid, the rate being governed by the thickness and carbon content of the object. To give an unusually high combination of ductility and tensile strength and elastic limit, the object should be quenched in oil or water and, if possible, removed from the quenching liquid before it has cooled below 160 deg. C. Tempering should then begin within a few hours after the quenching, and if possible before the piece has cooled below 100 deg. C. For very high elastic limit and tensile strength, the tempering should be at 500 to 650 deg. C.; in this case, the ductility will be low. For intermediate properties, best suited to the majority of cases, the tempering should be at 600 to 650 deg. C. For greatest ductility, with good strength and elastic limit, the tempering should be at 625 to 750 deg. C."

A. S. T. M. Recommendations For Heat Treatment of Carbon Steel Castings. "The castings should be heated slowly and uniformly to temperatures varying with the carbon content of the steel, approximately as follows:

Carbon, per cent	Temperature, degrees C.
Up to 0.16	925
0.16 to 0.34	875
0.35 to 0.54	850
0.55 to 0.79	830

Nothing in these recommendations shall operate against the annealing temperature being 50, and in special cases 100, deg. C. higher than those given in the table, when necessary to attain the desired result. The castings

should be cooled slowly and uniformly in the furnace when it is desired that the steel possess the maximum softness. They may be cooled at an accelerated rate when it is desired that the steel possess rather higher tensile strength and elastic limit than can be procured by very slow cooling."

Grain Growth of Low Carbon Steel. When metal that has been cold worked is heated to a sufficiently high temperature, new grains are formed from the fragments of the original grains. The lowest temperature of recrystallization, designated the **"germinative temperature,"** depends on the degree of prior plastic deformation. The greater the previous deformation the lower is the germinative temperature. Under certain conditions exaggerated grain growth may occur at this temperature. The conditions favoring exaggerated grain growth are: the presence of only one metallographic constituent, temperature gradient, strain gradient, and relatively high germinative temperature. Since the two latter conditions are most likely to occur in very slightly cold worked metal, exaggerated grain growth is often observed in such material after subjection to certain temperatures.

In steel having more than about 0.13 per cent carbon, pearlite is present in sufficient quantity to hinder grain growth of the ferrite. In steel having between 0.04 and 0.13 per cent carbon, however, formation of enormous grains may occur, if it is heated to 700 to 900 deg. C., and especially if the temperature is long maintained. Such grain growth in low carbon steel, with its accompanying brittleness, is known as **"Stead's brittleness."** Heating above 900 deg. C. breaks up the coarse grains and restores the ductility of the steel.

INFLUENCE OF CHEMICAL COMPOSITION ON PHYSICAL PROPERTIES

Sulphur in steel containing manganese occurs in dove gray non-metallic inclusions consisting largely of manganese sulphide. The effect of such inclusions on the physical properties of the metal depends on their number form and grouping. If manganese is insufficient in quantity to combine with the sulphur, the sulphur exists in combination with the iron as a film of iron sulphide on the grain boundaries. In this form it is very detrimental to the physical properties of the metal; it produces the effect known as **"red shortness."** For the above mentioned reasons, the percentage of sulphurs in high grade steel is usually limited to 0.02 to 0.05.

Phosphorus, unless in unusual quantity, occurs in steel in solid solution in the ferrite. It produces **"cold shortness"** or brittleness. This effect is greater in high than in low carbon steel. Specifications usually limit the percentage of phosphorus to 0.02 to 0.1 depending on the quality of steel desired.

Silicon in carbon steels may occur in any percentage up to 0.5 or 0.6. The higher percentage is frequently found in crucible steel. In this quantity, silicon is not considered harmful. Silicon promotes soundness in the metal either on account of its affinity for oxygen or because it increases the solubility of gases in the solid steel.

Silico-manganese steel is a medium carbon steel containing 1½ to 2 per cent silicon. It is used in tempered gears and springs.

Commercial high silicon steel usually contains from 2¾ to 3½ per cent silicon and has a low carbon percentage. When properly quenched and tempered, this steel has a permeability greater than that of pure iron. This property, with its high electrical resistance, makes silicon steel useful in transformers and other electrical apparatus.

Manganese. Since manganese as a deoxidising agent in the steel making process must be added in excess, practically all steel contains this element. If the finished steel contains less than about 0.2 per cent it usually displays some of the effects of incomplete deoxidation. Manganese increases the tensile strength and decreases the ductility; its effect on these properties increases with increase of the carbon content. If the percentage of manganese is raised above a limit of about 0.35 to 2 per cent according to the composition of the steel, the steel shows brittleness; in high carbon steels this brittleness manifests itself in a tendency of the steel to crack when quenched. Steels containing about 2 to 6 per cent manganese are so brittle that they may be pulverized by a hammer.

Manganese lowers the temperature and retards the rate of the austenite-pearlite transition. As the percentage rises above about 7 per cent, this effect becomes so marked that, when quickly cooled, part of the steel remains in the austenitic state. With further increase in the percentage of manganese, the proportion of austenite increases, with consequent increase in the ductility. In quickly cooled steels containing about 12 to 15 per cent manganese, the entire structure consists of austenite. When such steels are slowly cooled, more or less of the austenite is changed to martensite with consequent decrease in the ductility and increase in the hardness. The properties of a series of manganese steels are given in Table 6.

Table 6. Properties of Manganese Steels

C	Mn	Water quenched		Annealed	
		T. S.	Elong., 8 in.	T. S.	Elong., 8 in.
0.52	6.95	51,520	2	47,040	2
0.61	9.37	87,360	15	85,120	16
0.85	10.60	89,600	17	91,840	17
1.10	12.60	120,960	27	82,880	11
0.92	12.81	136,640	37	107,520	20
0.85	14.01	150,080	44	107,520	14
1.24	15.06	136,640	31	105,280	2
1.54	18.40	118,720	10	87,360	1
1.60	19.10	132,160	4	91,840	1

Austenitic manganese steel is so tough that it is practically unmachinable. Though not intensely hard to the file, it has the property of hardening greatly under slight deformation such as surface abrasion; this produces a hard surface with underlying tough metal. Manganese steel is used in machinery, such as rock crushing machinery, requiring a combination of hardness and toughness.

Nickel in steel occurs chiefly in solid solution in the ferrite; it is, therefore, not visible as a separate micro-constituent. Nickel raises the elastic limit and to a less extent the tensile strength, with only slight decrease in the ductility. In other words, for the same ductility, nickel steels have a higher tensile strength and elastic ratio than carbon steels. The effect of nickel on these physical properties, however, is less than half that of manganese. Nickel steel has a finer grain structure than carbon steeel; this accounts in part for its higher elastic limit and greater endurance. For the above reasons it is used in structural work such as bridges, in rails at curves, ordnance, engine forgings, shafting, etc. Although nickel steel in the annealed condition is superior to carbon steel, the full advantage of the use of nickel is obtained only by quenching and tempering.

Like manganese, nickel also lowers the temperature and retards the rate of the austenite-pearlite transition; its effect is about half that of manganese. Each 1 per cent of nickel up to 5 per cent lowers the transition temperature 10 or 20 deg. C. according as the transition is from pearlite to austenite or the reverse. In nickel as in manganese steels between certain percentage limits, there is a range of brittleness; this range includes steels containing from about 8 to 15 per cent nickel. With increase of nickel above 15 per cent, austenite appears in quickly cooled steels, and the proportion of this constituent increases until in 25 per cent nickel steel the structure is entirely austenitic. Austenitic nickel steels have considerable strength and ductility; they are used in apparatus or machinery requiring strength, toughness and resistance to corrosion.

With addition of nickel to steel, the coefficient of expansion decreases until in the alloy known as **"invar,"** containing about 36 per cent nickel, the coefficient is practically zero. With further increase in the percentage of nickel, the coefficient gradually increases until at 42 per cent the coefficient is about the same as that of glass or platinum; this alloy is therefore known as **"platinite."**

Chromium. This element, when present in steel, occurs partly in solid solution in the ferrite and partly in the cementite in the form of double carbides of iron and chromium. Though chromium strengthens the ferrite of steel, in the absence of carbon it does not produce any greater hardness than does silicon or aluminum. **Carbonless chromium steels** are not hardened by water quenching. The hardness of ordinary chromium steels, therefore, is due to the double carbide or **chromiferous cementite.** This double carbide, however, produces less brittleness in chromium steels than does an equal proportion of cementite in carbon steels. In chromium steels, proper **heat treatment** will produce a combination of great hardness and toughness.

Steel with about 0.5 per cent chromium and low carbon content is used in pieces that are to be case hardened; the absoprtion of carbon in this process forms a hard exterior of double carbides on a relatively soft interior. With about 0.35 to 1 per cent carbon, 0.5 per cent chromium steels are used in gears, springs, steel tires, saw blades, chisels, knives, razors, files, etc.

Steel with 1 to 1¾ per cent chromium and about 1 per cent carbon is used in ball races, cones, roller bearings, crushing machinery, safe steel, etc.

Steel with 2 per cent chromium is used in armor piercing projectiles, armor plate, cold rolls, drawing dies, etc.

Chromium steels are frequently used in the manufacture of three or five ply plate for plows, burglar proof safes, prison bars, etc. These plates or bars are made by rolling together alternate layers of chromium steel and wrought iron or low carbon steel, thus combining in one piece the hardness of chromium steel with the toughness of the softer metal.

Chromium, unlike manganese or nickel, slightly raises the temperature of the transition from pearlite to austenite. Like the former elements, however, chromium greatly retards the austenite-pearlite transition. When the chromium is increased above about 7 per cent, the structure even after slow cooling is martensitic. As the percentage is increased above about 12 to 15, white grains of the double carbides appear within the martensite. With more than about 20 per cent chromium, the structure consists entirely of double carbides.

Steel having about 12 to 15 per cent chromium is useful on account of its resistance to oxidation even at high temperatures. This steel, commercially known as **"stainless steel,"** is used in cutlery, valves of internal combustion engines, and other objects in which similar properties are required. When

properly treated, this steel has great strength and toughness as shown by the following example: Steel having 0.243 per cent carbon and 13.48 per cent chromium had the following physical properties: Tensile strength 107,500 lbs. per sq. in.; elastic limit 67,000; elongation in 2 inches, 25 per cent; reduction of area, 61.1 per cent.

Chromium-nickel steels have the valuable properties imparted by chromium and nickel separately. They have a combination of high elastic limit, toughness and wearing qualities. They **can be forged and machined more easily than chromium steels.** Care, however, is required in forging, and the temperature must not be allowed to fall below a bright yellow heat.

These steels are usually classified as low and high chromium nickel steels. **Low chromium-nickel steels** contain 0.5 to 0.8 per cent chromium and about 1½ per cent nickel; in this class is **"Mayari" steel,** which is made from a natural chromium-nickel ore. **High or "full" chromium-nickel steels** contain 1 to 1½ per cent chromium and about 3½ per cent nickel. The difference in properties between these two classes is not fully revealed by a comparison of the results of tension tests. Though either high or low chromium-nickel steel can be treated to produce nearly the same tension test results, the high chromium-nickel steel is much superior in shock resistance and endurance. In addition to these two classes, there are intermediate types.

The carbon content of these steels varies with the purposes for which they are to be used. With less than about 0.25 per cent carbon, they are used principally for case hardened pieces such as cam rollers, push rods, pivots, axles, crank shafts, gears, and any other parts requiring a combination of hardness and strength with toughness. High chromium-nickel steel, with 0.25 to 0.45 per cent carbon, is used for armor plate.

Vanadium Steel. With the exception of carbon, possibly no element has a greater effect on the physical properties of steel than has vanadium. When added in such quantity that 0.10 to 0.20 per cent remains in the finished metal, the tensile strength and elastic limit of the steel are greatly increased with little or no decrease in the ductility. The dynamic properties also are greatly improved.

A great part of this improvement in properties is probably due to the scavenging effect of vanadium. On account of its great affinity for oxygen, vanadium removes most of the residual oxygen after the usual deoxidizing agents have exerted their full effect. Since vanadium has an affinity for nitrogen, it is probable that it removes some of this element also, with consequent improvement in physical properties.

In addition to the improvement due to the scavenging effect of vanadium, there is probably also an improvement due to the effect of the vanadium remaining in the steel. Except in high speed steels, however, there seems to be no advantage, and there may even be a disadvantage, in the presence of more than about 0.3 per cent vanadium. Vanadium in steel occurs partly in the form of double carbides, which strengthen the steel, and partly in the form of a solid solution in the ferrite, which toughens the steel.

Chromium-vanadium Steels. Vanadium seems to exert its best effect in steel when in conjunction with chromium or nickel. In the former combination it seems to be most useful; most commercial vanadium steels contain chromium. Chromium-vanadium steels rival chromium-nickel steels in their usefulness in structural or machinery parts which require strength and fatigue resistance. With a carbon percentage of less than about 0.25,

chromium-vanadium steel is used chiefly in parts that are to be case hardened. With a carbon percentage of 0.25 to 0.40, this steel is used in quenched and tempered machinery parts, such as axles, crank shafts, connecting rods, etc. With a percentage of 0.45 to 0.65, it is used for springs.

Chromium-vanadium steel is much softer than chromium-nickel steel. It is more easily forged and machined. Like chromium-nickel steel, its full value is developed only by quenching and tempering. For effective treatment of this kind, however, it must be quenched from a higher temperature than is used in the treatment of chromium-nickel steel. The physical properties obtainable in the heat treatment of chromium-vanadium steel fall off very rapidly with increase in the size of the object treated. In large objects, the properties obtainable are inferior to those obtainable with chromium-nickel steel.

Molybdenum. This element occurs partly in solid solution in the ferrite, but chiefly in the form of double carbides in the cementite. In its effect on the physical properties it is similar in some respects to chromium and tungsten. On account of its lower atomic weight molybdenum can replace two or three times its weight of tungsten. Molybdenum is said to have a better effect than chromium on the ductility, toughness, and dynamic strength of steel. Excellent physical properties have been obtained in properly treated molybdenum steels. These steels have been used for rifle barrels, large guns, for machine parts such as propeller shafts, and to replace the tungsten of magnet and tool steels. The use of molybdenum in steel is rapidly increasing.

Chromium-molybdenum steels have recently been used in automobile parts requiring fatigue resistance. **Nickel-molybdenum** steels have also recently been used, but are said to be inferior to chromium-molybdenum steels.

Tungsten. Like molybdenum and chromium, this element occurs partly in solid solution in the ferrite of steel; it also occurs in the form of double carbides in the cementite, and probably in the form of iron-tungsten compounds. Though it retards the rate of transition from austenite to pearlite, its influence in this respect is less than that of chromium. In small percentages, tungsten increases the strength of steel with only slight increase in brittleness. For this reason, about 0.6 per cent tungsten is sometimes used in springs.

Magnets made of water quenched tungsten steel retain their magnetism better than carbon steels. Steel of the following percentage composition gives the best results: carbon, 0.5 to 0.7; tungsten, 4 to 7. Sometimes 0.5 per cent of chromium is added, and sometimes molybdenum is used in place of tungsten. These steels should be quenched from about 800 to 850 deg. C.

Self Hardening Tool Steels. The original air hardening or **"Mushet" steel** contained 4 to 12 per cent tungsten, 2 to 4 per cent manganese, and 1.5 to 2.5 per cent carbon. The self hardening properties of this steel were due more to the manganese than to the tungsten. Tungsten, even in high percentages and in the presence of carbon, confers very little self hardening properties on steel. Since the high percentage of manganese in this Mushet steel caused brittleness and tendency toward quenching cracks, the manganese was soon replaced by chromium, which confers self hardening properties with less brittleness. Becker gives the following range of composition of self hardening steels: carbon, 1.1 to 2.4 per cent; tungsten, 4.5

to 11.6 per cent; chromium, 0.07 to 3.4 per cent. The tungsten may be replaced by molybdenum.

High Speed Steel. Tools of self hardening steel, even if they contain considerable chromium and tungsten, have little if any better cutting speed than tools of carbon steel. If, however, such steels are heated to 1200 to 1350 deg. C. and cooled rapidly, they acquire the property of **"red hardness."** A tool made of this steel, when cutting so rapidly that the point is heated to dull redness, will not only retain its cutting edge but will be even harder than it was before use.

A study of the microstructure of the steel explains in part the remarkable properties produced by the high temperature heat treatment. At the high temperature, the double carbides of iron-tungsten compounds dissolve and form an austenitic structure. This structure, preserved in the steel by rapid cooling, is so stable that, when heated to 600 deg. C., the transformation proceeds only as far as martensite. The red hardness of high speed steel is, therefore, due to the stability of the martensitic structure even at dull redness.

The stability of the martensite at dull redness is due principally to the tungsten, while the hardness and self hardening properties are due principally to the chromium. *Taylor* and *White*, the discoverers of high speed steel, made many experiments to determine the best percentages of carbon, chromium and tungsten for these steels. They found that improvement could be made by decreasing the percentage of carbon and increasing the percentages of chromium and tungsten. The following percentage composition gave the best results: carbon, 0.674 to 0.682; chromium, 5.47 to 5.95; tungsten, 17.81 to 18.19.

Since the discovery of high speed steel many other elements have been tried. Many of the modern high speed steels are very complex; they may contain, in addition to chromium and tungsten, one or more of the following elements: uranium, zirconium, cobalt and vanadium. Of these recently tried elements, however, vandium is the only one that has proved of unquestioned advantage. Though this element is used in quantities as great as 3 per cent about 1 per cent appears to produce the best effect on the cutting properties.

Zirconium. In addition to its use in high speed steel, zirconium has been used in bullet proof thin steel plate. For this use, a **nickel-zirconium steel** was found superior to either chromium-nickel or molybdenum-nickel, steel. The steel had the following percentage composition: carbon, 0.42; manganese, 1.0; silicon, 1.5; nickel, 3.0; zirconium, 0.34.

Titanium. This element, on account of its strong affinity for oxygen, is a valuable **deoxidizing agent** in steel manufacture. By eliminating oxides to a greater extent than usual, titanium tends to prevent the formation of blow holes. The delayed solidification of the metal, due to its freedom from oxides and to the great heat of formation of titanium oxide, tends also to diminish the "pipe." Titanium has also a strong affinity for nitrogen, forming red crystals of titanium nitride.

Although titanium does not remain in the finished metal, the increased soundness and freedom from oxygen and nitrogen, produced by its use, greatly improves the physical properties.

Copper. It was formerly believed that copper, even in small percentages, produces red shortness in steel. Careful investigation, however, has shown that this element in percentages under about 1.5 does not produce red shortness, unless there is also present a rather high percentage of sulphur. Copper, in quantities up to about 1 per cent, produces a slight increase in tensile strength and elastic limit, with little if any decrease in ductility.

Copper has recently been used in steel plate to **diminish corrosion.** For this purpose it is used in percentages up to about 0.5. In the presence of this element, the rust film, instead of accelerating corrosion, is protective.

Case Carburization or Partial Cementation

Of the many investigations of this subject, the most valuable are those by *Giolitti* and his co-workers. These physico-chemical and metallographic investigations, especially of the action of gaseous carburising agents, have thrown much light on the entire subject of cementation of iron and steel.

Giolitti found that, in the absence of penetrating gases, carbon, even when finely divided and in close contact with iron, exerts a carburising action so slight as to be commercially negligible.

In most solid cements, the **carburising action is due largely to** the specific action of carbon monoxide or gaseous hydrocarbons generated from the cement.

The carburising action of carbon monoxide is due to a reaction expressed by the following equation: $2\ CO \rightleftarrows CO_2 + C$. This reaction, being reversible in the presence of a **catalyzer** such as iron, proceeds to an equilibrium, in which the relative concentrations of the two gases depend only on the temperature. If the concentration of carbon monoxide exceeds this definite relative value, the reaction as indicated by the above equation, sets free carbon. Since carbon monoxide penetrates red hot iron, the carbon set free by the above reaction is deposited in the interior of the metal, the concentration decreasing with depth. If the temperature of the metal is above the transition range, this carbon dissolves in the austenite.

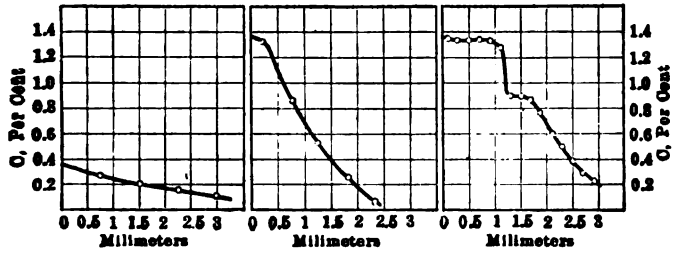


Fig. 6. Fig. 7. Fig. 8.

According to Giolitti, the concentration-depth curve obtained with carbon monoxide is of the form shown in Fig. 6. The curve shows the effect of passing carbon monoxide for 10 hours over iron at a temperature of 1100 deg. C. With increase of temperature, pressure or time of exposure, the depth of penetration increases; the maximum concentration, however, decreases with rise of temperature. When carbon monoxide alone is used, the maximum concentration never reaches the eutectoid value, and the decrease of concentration with depth is gradual. This form of curve was designated by Giolitti as Type I.

Though gaseous hydrocarbons, such as methane or ethylene, deposit carbon on the surface of the iron, their chief cementing action is due to their penetration of iron and liberation of carbon in the interior. Gaseous hydrocarbons give concentration-depth curves designated by Giolitti as Type II. Fig. 7 shows the effect of passing ethylene gas for five hours over iron at a tem-

perature of 1000 deg. C. The maximum carbon concentration is much greater, and the decrease of concentration with depth is much more rapid, than in cases of Type I. When slowly cooled, the case consists of an exterior hyper-eutectoid layer, an adjacent eutectoid layer, and an inner hypo-eutectoid layer.

When cases of Type II are slowly cooled from the carburizing temperature, the eutectoid layer is relatively broad, and the change in carbon content between this layer and either the exterior layer or the core is very abrupt. This is shown not only by the microscope but also by concentration-depth curves such as the one shown in Fig. 8. This curve was obtained from a piece of steel that had been subjected to the action of ethylene for four hours at 1050 deg. C. and allowed to cool very slowly. During the slow cooling, the cementite and ferrite separating out near the outer and inner boundaries of the carburized region attract cementite and ferrite respectively from the adjacent regions. This migration of the cementite and ferrite toward the exterion and interior respectively is known as **"liquation."**

Carburized cases of Type II, on account of the free cementite in the external layer, are excessively hard and brittle. On account of this brittleness and the sudden decrease of carbon content with depth, there may be scaling or **"exfoliation"** of the case. Exfoliation is especially likely to occur when there are distinct layers due to liquation. The cleavage in such material when hardened is usually at the boundary between the eutectoid and hypo-eutectoid layers, since, this is the boundary between hardened and unhardened metal.

Since these disadvantages result from the use of hydrocarbons alone, and since by use of carbon monoxide alone the resulting maximum carbon concentration is too small for practical purposes, commercial use is made of mixtures of carbon monoxide and hydrocarbons in the proper proportions to produce cases of any intermediate type. By this means the maximum carbon concentration may be so regulated that it does not exceed the eutectoid value.

Another method of obtaining cases of intermediate type consists in the use of carbon monoxide in the presence of excess carbon. The metal, surrounded by finely divided charcoal, is subjected to the action of carbon monoxide. If the temperature is sufficiently high, or the action long continued, this method may even produce hyper-eutectoid layers. By regulation of the temperature and the concentration of the carbon monoxide, however, it is possible to obtain cases of intermediate type. Practical regulation of the concentration of the carbon monoxide may be secured by the use of producer gas of properly adjusted composition.

Cases of Type II may be converted into cases of intermediate type by the use of carbon monoxide alone. The carbon monoxide thus exerts an equalizing action, reducing the carbon content in the external layer and flattening the carbon concentration gradient.

Most solid carburizing mixtures owe their carburizing action to the evolution of carbon monoxide or a mixture of carbon monoxide and hydrocarbons in the presence of excess carbon. The carburizing action of leather residues and other cements of animal origin is due, not to the formation of cyanides as formerly supposed, but to the evolution of gaseous hydrocarbons. One of the best known solid cements consists of a mixture of 60 parts charcoal and 40 parts barium carbonate. This mixture forms carbon monoxide by the action of the carbon on the carbon dioxide evolved from the carbonate. The mixture after use regains its value by absorption of carbon dioxide from the air.

Although the effect of cyanides in solid carbonaceous cements is slight, cyanides and ferro- and ferri-cyanides of the alkali metals have a very rapid carburising action. They are used to produce quickly a thin, hard case in which the carbon concentration decreases rapidly with depth. For this purpose, the metal is either immersed in a vessel of molten cyanide until the required carburization is obtained, or dipped in the cyanide and afterward heated to redness in the air. From five to fifteen minutes' immersion is required to give the desired effect.

Heat Treatment After Carburization. Since the object of case hardening is to produce a hard surface, the carburized pieces are usually quenched in water or oil. Frequently, however, they are allowed to cool slowly after carburization, and then reheated and quenched. Since, as illustrated by Fig. 8, the preliminary slow cooling may cause liquation and consequent exfoliation, it is better to quench the metal from the temperature of carburization. On account of the long heating during carburization, and consequent coarsening of the structure of the core, however, it is usually necessary to give additional heat treatment for grain refining. Since, on account of the higher carbon content of the case, the grain refining temperature of the case is usually lower than for the core, it is usually necessary to give two heat treatments to case carburized steel. The first or **"regenerative" heat treatment** consists in heating to a temperature slightly above the transition range of the core and quenching; this refines the core. The second or **hardening treatment** consists in heating to a temperature slightly above the transition temperature of the case and quenching; this refines and hardens the case. Although these two treatments are usually necessary to produce the best properties in core and case, one treatment may be sufficient if somewhat inferior properties in either case or core are unobjectionable.

CAST IRON

Foundry Pig Iron. Iron castings are made by remelting the blast furnace product known as "foundry pig iron." Foundry iron was formerly graded entirely by the appearance after fracture. The grades were numbered in order of decreasing coarseness of fracture from No. 1 to No. 6. In the series thus obtained, the color varies in the same order as the coarseness. In Nos. 1 to 4, the fracture is **gray**; in No. 5, usually called **"mottled" iron**, the fracture shows alternate patches of white and gray; No. 6, from the color of its fracture is known as **"white" iron.**

Though grading by fracture is still in use, increasing use is now being made of chemical analysis. In some specifications, while the system of grading by numbers is still retained, a range of chemical composition is prescribed for each grade. While there are great differences between the various specifications in the range of composition assigned to the different grades, the average ranges are approximately those given in Navy Department specifications for foundry pig iron, p. 148. In some specifications, such as those of the A. S. T. M., grading by numbers has been abandoned.

The differences in properties between the grades of foundry pig iron are due chiefly to variation in condition of the carbon. While the variation in total carbon content is relatively slight, the variation in the condition in which the carbon occurs is great. In No. 1 foundry iron, the carbon occurs almost entirely in the form of graphite. In white iron, it occurs entirely in the form of "combined carbon." In the various intermediate grades, the graphite and combined carbon occur in various proportions. The condition of the carbon is influenced largely by the chemical composition, and especially by the silicon content.

Microstructure of Gray Cast Iron. Since the various grades of foundry iron and gray cast iron are all transformation products of the austenite-cementite series, the microstructures of these irons should be studied in connection with those of the steel-white cast iron series. White cast iron, when slowly cooled, consists of the aggregates primaustenoid, eutectic, and sometimes primary cementite, see p. 85. Since the austenoid, both primary and eutectic, consists of pro-eutectoid cementite and pearlite, and since the pearlite consists of ferrite and cementite, the ultimate structure of white cast iron is made up entirely of these two constituents. In gray cast iron, however, more or less of the cementite has been decomposed into ferrite and graphite. The microstructure of low phosphorus gray cast iron, therefore, consists of ferrite, graphite, and usually cementite in various states of aggregation. In foundry irons, there is usually enough phosphorus to form an eutectic of iron phosphide, ferrite, and sometimes cementite. This eutectic is known as **"Steadite."**

Although the degree of graphitization of the cementite is influenced chiefly by the percentage of silicon, it is also greatly influenced by the duration of exposure to various temperatures between the solidification temperature and room temperature. The effect of temperature on graphitization is illustrated by the rapidity of decomposition of the primary cementite of hyper-eutectic cast iron. This primary cementite, which is formed at a temperature above that of final soldification of the iron, is often decomposed immediately after formation, the graphite thus formed floating on the surface of the molten iron as a scum known as **"kish."** In solidified iron, graphitization, though less rapid than in molten hyper-eutectic iron, may proceed rapidly if the iron is held near the temperature of final solidification. From this point, the rate of graphitization decreases rapidly as the temperature is lowered. The degree of graphitization, therefore, depends on the rate of cooling after solidification, and this in turn depends on the size of the casting.

If cast iron, having a sufficient percentage of silicon, is held at a temperature just below that of solidification, or is cooled at a slow enough rate, the eutectic cementite is decomposed, leaving a matrix consisting essentially of hyper-eutectoid steel. If the percentage of silicon is still higher, or the rate of cooling is still lower, there may be also a decomposition of the pro-eutectoid cementite, leaving a matrix of eutectoid steel. With still further increase in the percentage of silicon, or decrease in the rate of cooling, there may be partial or complete decomposition of the cementite of the pearlite, leaving a matrix of hypo-eutectoid steel or even of nearly pure ferrite.

The degree of graphitization depends also on the pouring temperature; the higher the pouring temperature, the less the decomposition of the cementite. For this effect of pouring temperature, no conclusive explanation has been given.

Properties of Cast Iron in Relation to Microstructure. Since gray cast iron consists of a matrix, varying from white cast iron to nearly pure ferrite, in which are embedded particles of graphite of various sizes and shapes, the physical properties may evidently vary over a wide range. These properties depend on the microstructure of the matrix and also on the amount and distribution of the graphite. Since the graphite particles break up the continuity of the matrix, they exert a weakening effect depending on their number, size and shape. **The softest and weakest cast iron** is that having a ferrite matrix and having the carbon all in the form of graphite. With increase in the amount of pearlite, the matrix increases in strength; the

accompanying decrease in the quantity of graphite also adds to the strength of the metal. **The strongest iron** is one having a pearlitic matrix and having the excess carbon all in the form of graphite. Further decrease in the quantity of graphite does not improve the physical properties of the metal, since it merely replaces graphite by brittle plates of cementite.

The relations between the percentages of total carbon, combined carbon, and graphite in the foundry irons and in the various types of iron castings are illustrated by Figs. 9, 10, and 11, taken from "*The Metallography of Steel and Cast Iron*," *by Henry M. Howe.* These diagrams show the relation between the gray irons and the steel-white cast iron series. Studied in connection with Figs. 4 and 5, these diagrams also show the approximate proportions of the micro-constituents and the approximate properties of the matrix of each type of foundry iron and gray iron casting.

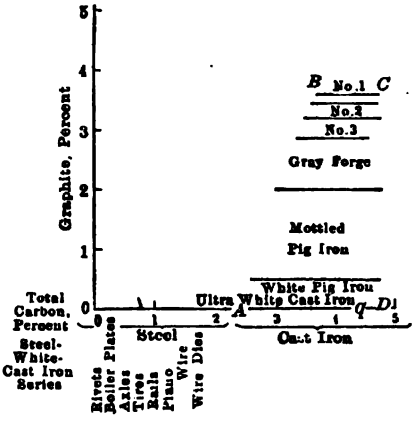


FIG. 9.

In these diagrams, the approximate ranges of composition of the various foundry irons and types of castings are represented by areas, whose boundary lines should be considered merely as approximations; in fact, the fields should overlap. Although the steels, since they have no graphite content, should be represented by positions on the lines designated *OA*, their names are written below the line *OA* in each of these figures in order to avoid confusion.

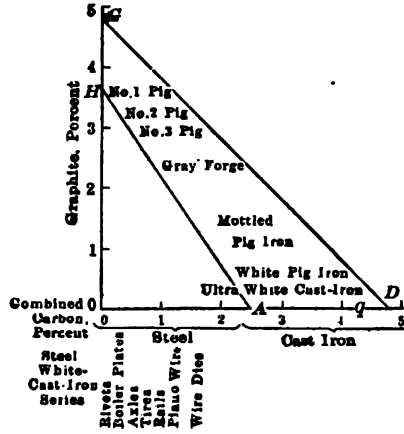


FIG. 10.

In Fig. 10, the abscissas and ordinates of the position designated by the name of each grade of pig iron, represent respectively the approximate percentages of cementitic carbon and graphite. Similarly in Fig. 11, each of the areas, 1 to 8, represents the approximate range of combined carbon and graphite in a type of cast iron; the matrix of each of these types of cast iron is similar in microstructure to the type of steel represented in the diagram by an equal abscissa.

Chemical Composition and Its Influence on Properties

Silicon. The ordinary range of this element in coke pig iron is from about 0.75 to 4 per cent. Pig irons having more than 4 per cent, however, are frequently used as a means of adjusting the percentage of silicon in mixtures for casting. Irons having from 3 to about 6 per cent silicon are sometimes called "**Scotch**" **irons**; those having about 6 to 14 per cent are known as "**silvery**" **irons** or "**softeners.**" If more than about 14 per cent is present, the alloy is called **ferro-silicon.** Although charcoal pig iron may have from 0.1 to 5 per cent silicon, iron having from 0.1 to 0.75 per cent is most useful. Since coke pig iron having less than about 0.75

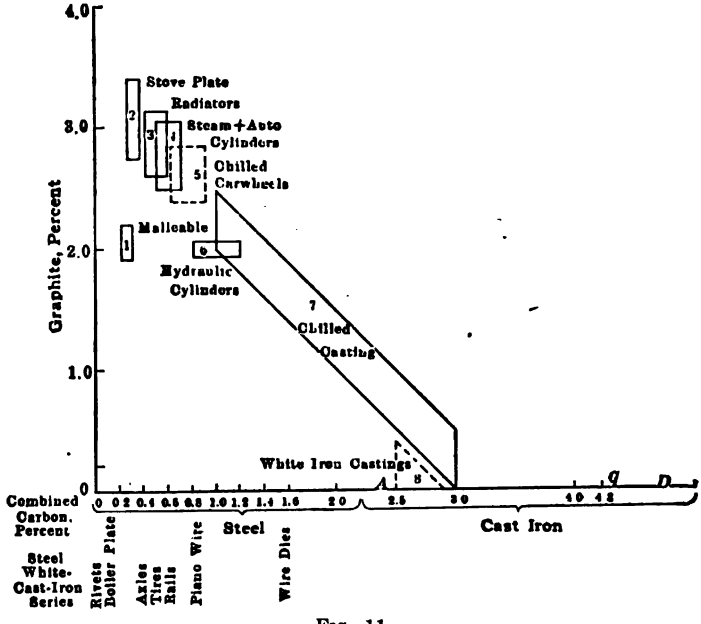


FIG. 11.

per cent silicon is of poor quality, chiefly on account of its high sulphur content, it cannot compete with charcoal pig iron of this composition.

Since silicon decreases the solubility of carbon in molten iron, the percentage of total carbon in pig iron decreases with increase in the silicon percentage. While in coke pig iron of low silicon content, the total carbon is nearly 4 per cent, in pig iron containing 4 per cent silicon the total carbon percentage is only about 3.2.

The most important influence of silicon, however, is on the condition of the carbon. Differences in silicon content are the chief cause of the differences in the proportions of combined and graphitic carbon in the various grades of pig iron as illustrated in Fig. 11. The usual silicon percentages in such foundry irons are approximately as listed in Navy Department Specifications, p. 148.

Sulphur and manganese, unlike silicon, increase the stability of the cementite and thus retard graphitization. A change of 0.01 per cent in the sulphur content has a noticeable effect on the quantity of cementite and therefore on the hardness of cast iron. Since sulphur in cast iron as in steel has an embrittling effect, its percentage is carefully limited by specifications. Manganese below about 1 per cent usually does not have much hardening effect; it may even have a softening effect due to its affinity for sulphur. When the percentage rises above 1, however, the hardening effect is considerable.

Since **phosphorus** increases the fluidity of cast iron, its presence is desired in castings of thin section or intricate form, if great strength is not essential. Phosphorus, occurring as it does in the form of the brittle eutectic, Steadite, increases the brittleness of cast iron. For this reason, castings in which strength is essential should contain less than 0.4 per cent. For ordinary castings, the average is about 0.7 per cent. In material such as stove plate, and in ornamental work, phosphorus may exceed 1 per cent.

Oxygen. *J. E. Johnson, Jr.*, as a result of a large number of experiments, formed the opinion that one of the most beneficial elements in cast iron is oxygen. He states that analysis has shown that the chief reason for the superiority of charcoal pig iron, especially cold blast charcoal iron, is the presence of oxygen. His theory is that the retention of oxygen in the iron is favored by the low hearth temperature of the charcoal blast furnace. By artificially oxygenating coke iron, Johnson was able to produce pig iron superior in strength to cold blast charcoal iron. The superiority of the oxygenated irons appears to be due to the fact that the graphite, instead of forming in long plates, forms in rounded particles. This spheroidising of the graphite would undoubtedly offset the detrimental effect of the oxygen on the matrix. Johnson's opinions in regard to the effect of oxygen are not shared by all prominent metallurgists.

Table 7

	Silicon, per cent	Sulphur, max. per cent	Phosphorus, max. per cent	Manganese, max. per cent	Total carbon, per cent
White iron castings	0.5 –0.9	0.1–0.25*	0.8	0.8	4.00 †
Chilled castings	0.6 –1.25	0.08	0.25	0.6	3.5 –3.75
Car wheels	0.6 –0.8	0.09	0.4	0.8	3.5 –3.75
Chilled rolls	0.6 –0.85	0.08	0.4	0.8	3.0 –3.25
Malleable castings, as cast	0.45–1.25	0.1	0.225	0.3	2.5 –3.5
Cylinders; air, gas, ammonia, hydraulic	1.0 –1.75	0.10	0.5	0.8	3.0 –3.5
Gun iron	1.0 –1.25	0.05	0.3	0.4	2.75–3.25
Bed plates	1.25–1.75	0.12	0.4–0.6*	0.8	3.25–3.75
Piston rings	1.5 –2.0	0.08	0.5	0.6	3.25–3.5
Automobile cylinders	1.75–2.0	0.08	0.4–0.5*	0.8	3.0 –3.25
Machinery castings, heavy	1.25–1.75	0.10	0.8	1.0	3.0 –3.25
Machinery castings, medium	1.75–2.0	0.08	0.8	0.8	3.25–3.5
Machinery castings, light	2.0 –2.4	0.06	0.7	0.6	3.25–3.75
Dynamo frames	2.0 –2.5	0.08	0.6	0.4	3.25–3.5
Grate bars	2.0 –2.25	0.06	0.2	0.6	3.25–3.5
Pipe fittings	2.0 –2.5	0.08	0.4	0.6	3.25–3.75
Stove plate	2.25–2.75	0.06	0.8–1.25*	0.6	3.5 –4.0
Ornamental castings	2.5 –3.0	0.06	1.0–1.25*	0.4	3.75–4.0

*Limits.
† Below.

Aluminum and nickel accelerate, while **vanadium and chromium** retard graphitization. In general the effects of these elements on the matrix of cast iron are the same as their effects on steel.

Castings for Various Purposes. Table 7, taken from "*Principles of Iron Founding*," *by Richard Moldenke*, gives the average compositions of castings for various purposes. The series, which is arranged approximately in order of silicon content, follows the same general order as the series shown in Fig. 11.

Chilled castings require careful adjustment of the quality and composition of the mixture. Charcoal pig iron, melted in the air furnace, is used in chilled castings of high quality such as chilled rolls. For car wheels, coke pig iron is now frequently used. The percentages of silicon necessary to give various depths of chill are given in the Table 8 taken from "*Principles of Iron Founding*," *by Moldenke*.

Table 8

Silicon in mixture, per cent	1.20	1.10	1.00	0.90	0.85	0.80	0.75	0.65	0.60	0.50	0.40
Silicon in casting, per cent	0.90	0.80	0.70	0.60	0.55	0.50	0.45	0.35	0.30	0.20	0.10
Depth of chill, in	¼	⅜	¾	1	1¼	1¾	1½	1¾	2	1½	3

"Gun iron," so named from the fact that this type of iron was formerly used in cast iron cannon, is gray iron of the highest quality. It is used in castings in which strength is essential. Gun iron is usually made in the air furnace from the best grades of pig iron and scrap.

For parts of electrical apparatus, such as dynamo pole pieces, in which high magnetic permeability is required, a soft grade of cast iron is necessary. For this reason, the combined carbon and manganese should be kept low. The percentage of graphite should also be reduced, if possible, by addition of steel to the mixture.

Growth of Cast Iron at Red Heat. When gray cast iron is heated above about 650 deg. C. there is a permanent increase in volume, the amount of increase depending on the temperature and length of exposure; the growth is especially noticeable after the iron has been repeatedly heated and cooled. An increase of about 17 per cent in length and 46 per cent in volume has been observed, with decrease in the specific gravity from about 7.1 to 5½. Although growth does not occur below 650 deg. C., above this temperature the rate of growth increases rapidly and reaches a maximum at about 735 deg. C.

It has been found that the growth is roughly proportional to the percentage of silicon, and hence to the percentage of graphite. That the growth is due chiefly to penetration of oxygen to the interior by way of the graphite plates is evident from the following facts: Microscopic examination shows that the graphite plates in the enlarged iron are surrounded by layers of oxides; the weight of the enlarged iron, as well as its dimensions, is considerably increased; growth does not occur if the iron is heated in a vacuum. Growth of cast iron after repeated heating, therefore, is due to the increased volume of the oxidized material.

Since growth by internal oxidation results in impairment of the physical properties and final failure of the material, the percentage of graphite in material for grate bars, furnace sections, annealing boxes, etc., should be as low as possible. When white cast iron is unsuitable on account of its brittleness, the best cast iron for such purposes is a gray iron whose composition is near the boundary between gray and mottled iron. Low per-

centages of sulphur and phosphorus also are desirable in such material. For this reason, the best pig iron for use in heat resistant castings is of the Bessemer grade.

Shrinkage of Cast Iron. Some writers make a distinction between the terms "**shrinkage**" and "**contraction.**" They apply the term "shrinkage" to the decrease in volume during solidification, and the term "contraction" to the decrease in volume after solidification. Since, however, both words have the same general meaning, and are in common use to designate decrease in volume either before or after solidification, a distinction between the terms seems inadvisable.

After solidification of cast iron, in addition to the natural decrease in volume with falling temperature there are from one to three volume changes due to transition from one kind of microstructure to another. There may be actual expansion of the iron at about 1100, 900, and 725 deg. C. The first of these expansions is due to the formation of the relatively bulky graphite; the second (in phosphoric irons) is due to the solidification of the phosphide eutectic; the third is due to the formation of pearlite. The net result of the cooling to room temperature, however, is a shrinkage, which varies with the composition of the material.

The shrinkage or contraction **in white cast iron** is about $\frac{1}{4}$ in. per foot. **In gray irons,** the shrinkage depends on percentage of silicon, size of casting, pouring temperature, etc. In other words, the shrinkage is decreased by every factor that increases the percentage of graphite. The average allowance for shrinkage in gray cast iron is $\frac{1}{8}$ in. per foot.

Malleable Cast Iron. This metal as cast is a white iron. It is afterward so altered by annealing that it greatly exceeds ordinary gray iron in strength and ductility. The improvement in properties is due to the fact that the malleablizing process either removes the carbon almost entirely from the white cast iron or converts it into scattered groups of fine particles known as "**temper carbon**" which are less harmful than the graphite flakes of gray cast iron.

There are **two processes of making malleable cast iron.** The original **Réaumur process,** still used in Europe, is chiefly one of **oxidation.** In this process, white iron, packed in iron oxide, is heated for about 110 hours at 850 to 950 deg. C. By this method, most of the carbon is removed. The resulting metal, since it gives a light colored fracture throughout its entire cross-section, is known as "**white heart**" **malleable cast iron.**

The **American malleablizing process** is essentially one of **graphitization,** though it also produces some surface oxidation. In this process, the white cast iron is packed in a mixture of one part new ore and two or three parts ore that from previous use has lost nearly all its oxidizing power. It is then heated for about 60 hours at about 675 to 730 deg. C. In the resulting product, the fracture is dark in color with the exception of a decarbonized exterior layer. For this reason, the product is called "**black heart**" **malleable cast iron.**

The difference in form between the graphite of gray and of malleable cast iron is due to the difference in the temperatures at which graphitization occurs in the two kinds of material. While the graphitization of gray cast iron occurs chiefly at a temperature not more than 30 to 40 degrees C. below the temperature of solidification, the graphitization of malleable cast iron occurs at a much lower temperature. At the lower temperature, the rigidity of the metal hinders the coalescence and growth of the graphite particles and thus prevents the formation of graphite flakes.

The composition of the iron for malleable castings must be so adjusted that no graphite forms during the casting process; yet there must be enough silicon present to facilitate graphitisation during the annealing process. Since sulphur diminishes the strength and increases the tendency to form shrinkage cracks, the percentage of this element must be kept low. Since phosphorus diminishes the shock resisting power, the percentage should not exceed about 0.25 per cent. The average composition of malleable cast iron is given in Table 7 above.

The strength of cast iron in relation to the various types of microstructure has already been discussed. Description of the tests and the usual numerical values representing the strength of the different kinds of cast iron are given in the specifications, p. 149. The **tensile strength** does not usually much exceed 20,000 lbs. per sq. in. unless considerable steel scrap has been used in the mixture. The tension test is of less value than the cross breaking or "transverse" test. For this test, square or round bars of various cross-sections and lengths are in use. The "arbitration bar" of the American Society for Testing Materials, however, is in general use. This is a cylindrical bar 1¼ in. in diameter, which is tested on supports 12 in. apart; see pp. 150 and 152. The **compressive strength** of cast iron is its most valuable property. It varies from 60,000 to 200,000 lbs. per sq. in.

Melting Processes for Cast Iron. Cast iron is usually melted in either the cupola or the "air furnace." The crucible, open hearth, or electric furnaces are sometimes used.

The **cupola** is a shaft furnace, usually at least 15 ft. high and 2½ to 7½ ft. in diameter, consisting of a vertical cylinder of boiler plate lined with fire brick. The top is open for the escape of the products of combustion, and near the top is a side opening to receive the alternate charges of fuel and metal. At the bottom is the hearth or crucible with tap holes for slag and metal. A little above the slag hole are the tuyeres for air blast.

In charging the cupola, coke is added first until the coke bed extends about 22 to 24 in. above the tuyeres. Alternate charges of metal and fuel are then added until the cupola is full. Each fuel charge except the first should be in sufficient quantity to make a layer about four inches deep. Each metal charge will ordinarily weigh about ten times as much as a fuel charge; if the metal charge contains considerable steel, however, the charge must be smaller, and may be only seven times as much as the coke charge. In adding the metal charge, any steel in the mixture should be added first, so that it will be in direct contact with the preceding fuel charge; a layer of pig iron is then added, and finally a layer of light cast iron scrap. Charged in this order, the three kinds of material will melt almost simultaneously.

After the charged cupola has remained for an hour or two with air blast nearly shut off, the full blast is turned on. The "first metal" should appear in the hearth in eight to ten minutes; if less time is required, the first fuel charge has been too thin. The melting may be allowed to continue without addition of further charge until the cupola is empty, or the charging may be continuous as in a blast furnace.

There are two types of **air furnace**. Both types are reverberatory furnaces. The first type is used in gray iron foundries for making the best grade of castings such as chilled and sand rolls, steam and gas engine cylinders, gun iron, etc. This furnace, being fairly high and wide in proportion to its length, can be charged with heavy scrap. The furnaces vary in capacity from 5 to 40 tons, 25 tons being the ordinary capacity. Natural draft is used.

The second type of air furnace is used in making malleable castings. Since no larger pieces than pig iron are charged, this furnace is made long and narrow with a low roof. Forced draft is used.

Since the metal in the air furnace is not in direct contact with the fuel, the sulphur absorbed is less in the air furnace than in the cupola. While the total carbon content can be readily reduced in the air furnace by the addition of steel scrap, such reduction cannot readily be made in the cupola on account of the absorption of carbon from the fuel. In the air furnace, a final adjustment of the composition of the metal can be made just before tapping.

Mixture Making for Cast Iron. In making mixtures for iron castings, allowance must be made for changes in composition during melting. These changes depend on the type of melting furnace, the quality of the fuel, and the care exercised in the melting process.

The **percentage of silicon** should be about 0.25 higher in the mixture than in the casting, if the cupola process is used. If the air furnace process is to be used, the allowance for loss should be 0.35 per cent.

The **losses of manganese** in the cupola and air furnace processes average 15 and 35 per cent respectively of the quantity of manganese in the mixture.

The **loss of carbon** in the air furnace is from 5 to 15 per cent of the quantity in the mixture. In the cupola, the change is variable. The absorption of carbon from the fuel is greater the lower the percentage of silicon, and the larger the percentage of steel scrap in the mixture.

In cupola practice, the **increase of sulphur** during melting is from 0.02 to 0.07 per cent. In the air furnace, the increase should not be more than about 0.01 per cent.

Since the percentage of sulphur in cast iron is increased by each remelting, the **proportion of scrap** in foundry mixtures **must be carefully limited.** If a large proportion of scrap is regularly used, there will be a progressive increase in the sulphur content of the foundry product.

FERRO-ALLOYS

With the exception of ferro-manganese, spiegeleisen, and ferro-silicons with lower silicon content, all ferro-alloys are made in the **electric furnace.** A list of the more important ferro-alloys, with the approximate percentage of each, is given in Table 9.

Table 9.—Ferro-alloy Compositions

	Special element, per cent	Carbon, per cent	Sulphur, per cent	Phosphorus, per cent	Manganese, per cent	Silicon, per cent	Chromium, per cent	Nickel, per cent
Ferro-manganese	.....	5.7	0.03	0. 1–1.0	80	0.05–1.0		
Spiegeleisen	.....	5.18	0.002	0.055–0.1	10–26	0.76		
Ferro-silicon, blast furnace	.....	1.2	0.07	0. 1–0.5	.....	10–20		
Ferro-silicon, electric furnace	.....	0.35	0.02	0.04	0.35	25–30		
	.....	0.30	0.015	0.03 –0.1	0.30	45–50		
	.....	0.30	0.015	0.03	0.25	70–80		
	.....	0.25	0.015	0.03	0.15	90–95		
Silico-spiegel	.....	1–15	0.01	0.15	15–20	8–15		
Silico-ferro-manganese	.....	.....	.....	.....	60–80	20–25		
Ferro-chronium	.....	0.6 –10.0	0.02–0.25	0.02 –0.07	0.07–0.20	0.2 –2.25	60–65	
Ferro-nickel	.....	0.5 –1.0	0.01–0.20	0.02 –0.30	.....	0.2 –0.3	.....	25–85
Ferro-chrome-nickel	.....	0.5	0.03	0.01	.....	0.6	52	18
Ferro-vanadium	20–36	0.4	.....	.....	0.6	0.9		
Ferro-molybdenum	50–85	0.35–4.0	0.02–0.05	0.007–0.20	.....	0.11–0.30		
Ferro-tungsten	73–85	0.3 –3.5	0.02–0.05	0.015–0.03	0.09–3.5	0.1 –0.4		
Ferro-titanium	51.3	2.8	0.047	0.02	0.08			
Ferro-boron	10–12							
	20–25							
	30–35	3	0.03	0.005				
Ferro-phosphorus	0.2–1.7	0.05–0.1	16–25	0.1 –6.0	0.4 –2.5			

SPECIFICATIONS FOR WROUGHT IRON

U. S. Navy Specifications (Adopted, 1915) for Wrought Iron for chain and miscellaneous purposes require material to be of best quality American refined iron, puddled from all-ore pig iron and free from any admixture of steel or scrap. Short pieces must not be used in piling. All material shall be free from injurious defects and have clear surfaces and workmanlike finish, bars to be straight and of true section. Grade A, which must be double refined and is specified for chain cable and similar articles, and Grade B, which is specified for general blacksmith work, are specified with physical and chemical requirements in accordance with Table 10.

Table 10

Grade	Minimum tensile strength per sq. in.	Minimum yield point	Minimum elongation[1]	Minimum contraction area	Maximum amount of	
					P	S
			Per cent	Per cent	Per cent	Per cent
A	48,000	½ ultimate strength	26	40	0.10	0.015
B	45,000	⅔ ultimate strength	25	40 rounds 35 flats	0.15	0.020

Grade	Bends	
	Cold	Quench
A	180 deg. around a diam. of one thickness.	Heat to 1700 deg. F. and bend to same requirements as cold bends.
B	Flats ⅝ in. and less around a diam. to two thicknesses, all other material to 180 deg. around a diam. of one thickness.	Temperature of the water in which the bar is to be quenched should be about 80 deg. F.

[1] The elongation will be measured in 8 in. with the following exceptions: Flats ¼ in. and less in thickness will be measured on a length equal to 25 times the thickness of the material tested.

On all other material less than ¾ in. diam. or thickness, the elongation will be measured on a length equal to 10 times the diam. or thickness of the material tested.

Tests. Material will be tested in sizes rolled where practicable, and for each ton or less of each size the following tests will be made: 1 tensile, 1 cold bend, 1 quench bend, 1 nick test, and 1 drift test. Each test specimen shall be taken from a different bar. When material can not be tested in sizes rolled, test pieces will be prepared to a sectional area as large as possible within the capacity of the testing machine for tensile tests, and machined to suitable size for bending and other physical tests. For material of sectional area above 4 sq. in., a reduction of 1 unit of percentage in elongation and contraction, and a reduction of 500 lbs. in tensile strength will be allowed for each additional 2 sq. in., and a proportionate amount of reduction for fractional parts thereof, provided the ultimate strength shall not fall more than 3000 lbs., nor the elongation more than 3 units of percentage, below the requirements of the grade of iron tested.

A bar nicked approximately 20 per cent of its thickness and bent back at this point through an angle of 180 deg., must show a long, clean, silky fiber, free from slag or dirt,

Chemical Composition and Its Influence on Properties

Silicon. The ordinary range of this element in coke pig iron is from about 0.75 to 4 per cent. Pig irons having more than 4 per cent, however, are frequently used as a means of adjusting the percentage of silicon in mixtures for casting. Irons having from 3 to about 6 per cent silicon are sometimes called "**Scotch**" **irons**; those having about 6 to 14 per cent are known as "**silvery**" **irons** or "**softeners.**" If more than about 14 per cent is present, the alloy is called **ferro-silicon.** Although charcoal pig iron may have from 0.1 to 5 per cent silicon, iron having from 0.1 to 0.75 per cent is most useful. Since coke pig iron having less than about 0.75

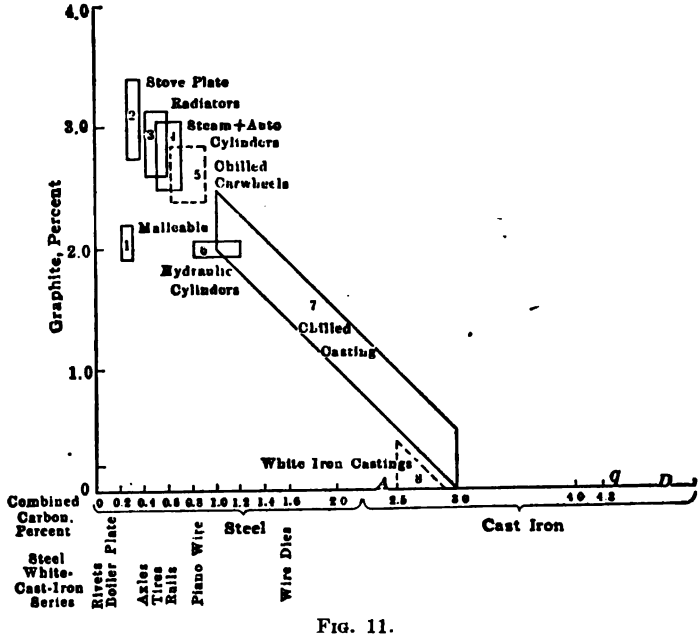


FIG. 11.

per cent silicon is of poor quality, chiefly on account of its high sulphur content, it cannot compete with charcoal pig iron of this composition.

Since silicon decreases the solubility of carbon in molten iron, the percentage of total carbon in pig iron decreases with increase in the silicon percentage. While in coke pig iron of low silicon content, the total carbon is nearly 4 per cent, in pig iron containing 4 per cent silicon the total carbon percentage is only about 3.2.

The most important influence of silicon, however, is on the condition of the carbon. Differences in silicon content are the chief cause of the differences in the proportions of combined and graphitic carbon in the various grades of pig iron as illustrated in Fig. 11. The usual silicon percentages in such foundry irons are approximately as listed in Navy Department Specifications, p. 148.

Sulphur and manganese, unlike silicon, increase the stability of the cementite and thus retard graphitization. A change of 0.01 per cent in the sulphur content has a noticeable effect on the quantity of cementite and therefore on the hardness of cast iron. Since sulphur in cast iron as in steel has an embrittling effect, its percentage is carefully limited by specifications. Manganese below about 1 per cent usually does not have much hardening effect; it may even have a softening effect due to its affinity for sulphur. When the percentage rises above 1, however, the hardening effect is considerable.

Since **phosphorus** increases the fluidity of cast iron, its presence is desired in castings of thin section or intricate form, if great strength is not essential. Phosphorus, occurring as it does in the form of the brittle eutectic, Steadite, increases the brittleness of cast iron. For this reason, castings in which strength is essential should contain less than 0.4 per cent. For ordinary castings, the average is about 0.7 per cent. In material such as stove plate, and in ornamental work, phosphorus may exceed 1 per cent.

Oxygen. *J. E. Johnson, Jr.*, as a result of a large number of experiments, formed the opinion that one of the most beneficial elements in cast iron is oxygen. He states that analysis has shown that the chief reason for the superiority of charcoal pig iron, especially cold blast charcoal iron, is the presence of oxygen. His theory is that the retention of oxygen in the iron is favored by the low hearth temperature of the charcoal blast furnace. By artificially oxygenating coke iron, Johnson was able to produce pig iron superior in strength to cold blast charcoal iron. The superiority of the oxygenated irons appears to be due to the fact that the graphite, instead of forming in long plates, forms in rounded particles. This spheroidizing of the graphite would undoubtedly offset the detrimental effect of the oxygen on the matrix. Johnson's opinions in regard to the effect of oxygen are not shared by all prominent metallurgists.

Table 7

	Silicon, per cent	Sulphur, max. per cent	Phosphorus, max. per cent	Manganese, max. per cent	Total carbon, per cent
White iron castings	0.5 –0.9	0.1–0.25*	0.8	0.8	4.00†
Chilled castings	0.6 –1.25	0.08	0.25	0.6	3.5 –3.75
Car wheels	0.6 –0.8	0.09	0.4	0.8	3.5 –3.75
Chilled rolls	0.6 –0.85	0.08	0.4	0.8	3.0 –3.25
Malleable castings, as cast	0.45–1.25	0.1	0.225	0.3	2.5 –3.5
Cylinders; air, gas, ammonia, hydraulic	1.0 –1.75	0.10	0.5	0.8	3.0 –3.5
Gun iron	1.0 –1.25	0.05	0.3	0.4	2.75–3.25
Bed plates	1.25–1.75	0.12	0.4–0.6*	0.8	3.25–3.75
Piston rings	1.5 –2.0	0.08	0.5	0.6	3.25–3.5
Automobile cylinders	1.75–2.0	0.08	0.4–0.5*	0.8	3.0 –3.25
Machinery castings, heavy	1.25–1.75	0.10	0.8	1.0	3.0 –3.25
Machinery castings, medium	1.75–2.0	0.08	0.8	0.8	3.25–3.5
Machinery castings, light	2.0 –2.4	0.06	0.7	0.6	3.25–3.75
Dynamo frames	2.0 –2.5	0.08	0.6	0.4	3.25–3.5
Grate bars	2.0 –2.25	0.06	0.2	0.6	3.25–3.5
Pipe fittings	2.0 –2.5	0.08	0.4	0.6	3.25–3.75
Stove plate	2.25–2.75	0.06	0.8–1.25*	0.6	3.5 –4.0
Ornamental castings	2.5 –3.0	0.06	1.0–1.25*	0.4	3.75–4.0

* Limits.
† Below.

cracks or flaws. This flange as measured from the outside of the tube shall be $\frac{3}{8}$ in. wide.

Hydrostatic Tests. Tubes under 5 in. in diam. shall stand an internal hydrostatic pressure of 1000 lbs. per sq. in., and tubes 5 in. in diam. or over an internal hydrostatic pressure of 800 lbs. per sq. in. Lapwelded tubes shall be struck near both ends, while under pressure, with a 2-lb. hand hammer or the equivalent.

Expanding. Tubes when inserted in the boiler shall stand expanding and beading without showing cracks or flaws, or opening at the weld.

Boiler Plates (1916)

Uses. *Class A* boiler plate is suitable for use where flanging or welding is not required. *Class B* boiler plate is suitable for use where flanging or welding is required, boiler manholes, condenser shells, baffles and supporting plates, feed water heater shells, baffles and plates, blower fans, evaporator shells, etc. *Class C* material is suitable for smokestacks, boiler casings, uptakes, etc., where great strength is not required.

Physical and chemical properties shall conform to Table 11.

Table 11

Class	Material	Tensile strength (lbs. per sq. in.)	Minimum elastic limit (lbs. per sq. in.)	Elongation (per cent in 8 in.)	Maximum percentage of P.	Maximum percentage of S.	Without showing cracks or flaw must cold bend about an inner diameter
A	Open-hearth steel.	65,000 to 75,000	½T. S.	22	0.035	0.035	Equal to thickness of plate and through 180 deg. for plates 1 in. in thickness and under and equal to 1½ times the thickness through 180 deg. for plates over 1 in. in thickness.
B	Open-hearth steel.	58,000 to 66,000	½T. S.	25	0.035	0.035	Flat back through 180 deg. for plates under 1 in. in thickness and equal to thickness of plate through 180 deg. for plates 1 in. and over in thickness.
C	Open-hearth or Bessemer steel.	To be of best commercial quality. Chemical and physical tests will not be made on this material unless the inspector has reason to doubt it is not of quality suitable for the purpose intended.					

When the finished plate is ¼ in. or less, the elongation shall be measured on an original length of 16 times the thickness of the plate tested.

Number of Tests. One longitudinal tensile test piece and one bending test piece (transverse) shall be cut from each plate as rolled at such points as may be designated by the inspector. Test pieces for circumferential plates of boiler drums shall be pulled in the direction of the circumference. The cold-bending test pieces may have their corners rounded to a curve, the radius of which is equal to ¼ the thickness of plate.

Surface Inspection. Boiler plates shall be flat and straight, free of all slag, foreign substances, brittleness, laminations, hard spots, brick or scale marks, scabs, snakes or other injurious defects.

Treatment of Bent and Flanged Sheets. Boiler-drum sheets, whenever bent, all flanged sheets, drum heads, headers, nozzles, and man and hand hole plates which are formed from boiler plate shall be formed hot. All flanged parts of boilers will be annealed in an approved manner after flanging. All boiler plates and stays will be well cleaned of mill scale by pickling or other approved means.

Boiler Rivets, Rods for (1915)

Uses. *Class A* material shall be used for all rivets where class A plate is used. *Class B* material may be used with class B boiler plate. *Class C* material shall be used for rivets where the strength of the boiler is not affected, such as:

Air ducts	Fireroom air screens
Ash dumps	Furnace doors
Ash pans	Ladders
Ash-pit doors	Oil tanks
Blower casings and fans	Smoke pipes and covers
Boiler casing, including that for water-tube boilers	Platforms and gratings
	Tallow and other tanks
Circulating plates for boilers	Uptakes and uptake doors
Coal and ash buckets	Feed and filter tanks

Chemical and physical requirements must be in accordance with Table 12.

Table 12

Class	Material	Min. tensile strength, lbs. per sq. in.	Min. elongation, per cent in 8 in.	Max. amount of		Bends[1]
				P.	S.	
A.....	Open-hearth nickel or carbon steel.	75,000	23	0.04	0.035	(1)
B.....	Open-hearth carbon steel.	58,000	28	0.04	0.035	(2)
C.....	Commercial steel.					

[1] Cold bend 180 deg. about an inner diam. equal to ½ the thickness of the test piece for diam. up to and including 1 in., and equal the thickness for diam. over 1 in.; quench bend 180 deg. about an inner diam. equal to the thickness of the test piece for diam. up to and including 1 in. and equal to 1½ times the thickness for diam. over 1 in.

[2] Cold bend flat back through 180 deg.; quench bend 180 deg. through an inner diam. equal to ½ the thickness of the test piece for diam. up to and including 1 in. and equal to the thickness for diam. over 1 in.

[3] Quench test pieces to be heated to a dark cherry red, as seen in daylight, and plunged into fresh clean water of 80 deg. to 90 deg. F.

Surface and Other Defects. The rods must be true to form, free from seams, hard spots, brittleness, injurious sand or scale marks, and injurious defects generally.

Upsetting Tests. From each heat of rounds as rolled there shall be cut 6 tests specimens about 1½ in. long, which shall stand hammering down cold, longitudinally, to ½ their original length without showing seams or other defects which would tend to produce imperfections in the finished product.

Boiler Rivets, Finished (1915)

Tests. Samples from each lot must stand the following tests without fracture, test (*a*) being applied to one lot and (*b*) to a second, etc.: (*a*) **Bend double cold** to a curve of which the inner diam. is equal to the diam. of the rivet; (*b*) **bend double hot** through an angle of 180 deg. flat back; (*c*) the **head to be flattened** when hot without cracking at the edges until its

diam. is 2½ times the diam. of the shank; (*d*) the **shanks of sample rivets to be nicked** on one side and bent cold to show the quality of the material.

Boiler Bracing (1918)

Chemical and Physical Requirements must be in accordance with Table 13.

Table 13

Class	Material	Min. tensile strength, lbs. per sq. in.	Min. elastic limit, lbs. per sq. in.	Elongation per cent in 8 in.	Elongation per cent in 2 in.
A.....	Shapes, open-hearth steel.....	75,000	40,000	23	26
A.....	Forgings, open-hearth steel.....	75,000	40,000	23	26
B.....	Shapes, open-hearth steel.....	60,000	32,000	26	30
B.....	Forgings, open-hearth steel.....	60,000	32,000	26	30

Class	Material	Maximum percentage of P.	Maximum percentage of S.	Bending tests	Opening and closing tests
A.....	Shapes, open-hearth steel.....	0.04	0.04		*
A.....	Forgings, open-hearth steel.....	0.04	0.04	*	
B.....	Shapes, open-hearth steel.....	0.04	0.04		*
B.....	Forgings, open-hearth steel.....	0.04	0.04		

* See following five paragraphs.

"A" Shapes Cold Bend. One test piece cut from each lot of class A shapes for bracing shall bend cold through an angle of 180 deg. to an inner diam. equal to twice the thickness of the piece tested without showing a fracture on outside of bent portion.

"A" Forgings Cold Bend. One bar ½ in. thick, cut from each lot of class A forgings for bracing, shall bend cold through an angle of 180 deg. to an inner diam. of 1 in. without showing a fracture on outside of bent portion.

"B" Shapes Cold Bend. One test piece cut from each lot of class B shapes for bracing shall bend cold through an angle of 180 deg. to an inner diam. equal to the thickness of the piece tested without showing a fracture on outside of bent portion.

"B" Forgings Cold Bend. One bar ½ in. thick, cut from each lot of class B forgings for bracing, shall bend cold through an angle of 180 deg. to an inner diam. of ½ in. without showing a fracture on outside of bent portion.

Opening and Closing Tests of Shapes. Angles, tee bars, and other shapes for boiler bracing shall be subjected to the following additional tests: A piece cut from 1 bar in 20 shall be opened out flat while cold without showing cracks or flaws; a piece cut from another bar in the same lot shall be closed down on itself until the two sides touch without showing cracks or flaws.

Annealing. All forged material for boiler bracing shall be annealed as a final process.

Surface Inspection. All boiler bracing shall be true to form, free from seams, hard spots, brittleness, injurious sand, or scale marks, or injurious defects generally.

Steel Shapes and Bars for Hull Construction (1917)

Finish. All shapes and bars shall be true to section, free from injurious defects, and shall have a workmanlike finish.

Physical and Chemical Requirements. All material shall be of uniform quality. The physical and chemical requirements of the various grades of material for shapes and bars shall be in accordance with Table 14.

Table 14

Grade	Material	Min. tensile strength, lbs. per sq. in.	Min. elongation in 8 in., Test specimens		Max. amount of		Cold bend
			Type 1, per cent	Types 2 and 3, per cent	P., per cent	S., per cent	
Soft, for flanging.	Open-hearth carbon steel.	50,000	35	30	0.05 acids; 0.04 basic	0.05	180 deg. flat on itself.
Medium.	Open-hearth carbon steel.	60,000	28	25	0.05 acid; 0.04 basic	0.05	For test specimens below ¾ in. in thickness, 180 deg. flat on itself. For test specimens ¾ in. or more in thickness the bends will be 180 deg. to a diam. of 1 thickness.
High tensile.	Open-hearth carbon, nickel, or silicon steel.	80,000	25	20	0.05 acid; 0.04 basic	0.05	180 deg. to a diam. of 1½ thicknesses.
Common.	Open-hearth or Bessemer steel.	55,000	27	22	No chemical analysis required.		180 deg. to a diam. of 1 thickness.

Steel Plates for Hulls (1916)

Physical and Chemical Requirements. The physical and chemical requirements and kind of material for plates shall be in accordance with Table 15.

Galvanized Plates. (*a*) *Physical and Chemical Requirements.* Plates to be galvanized shall meet the requirements for steel of the grade specified before galvanizing and shall conform to the permissible variations in weight and gage before galvanizing. (*b*) *Freedom from Surface Defects.* Galvanized plates must be thoroughly and evenly galvanized; of a bright appearance; free from pits, blisters, and other defects; and must be commercially flat. No rerolling of the plates after leaving the galvanizing bath will be permitted except for the purpose of straightening. The coating must not break off when scraped with a knife or if the plate is bent 90 deg.

Rods for Bolts, Nuts and Studs (1915)

Steel rods for bolts and studs of the grade ordered shall conform to the requirements of Table 16.

Table 15

Grade	Material	Minimum tensile strength, lbs. per sq. in.	Minimum elongation, per cent	Maximum amount of P., per cent	Maximum amount of S., per cent	Cold bend
Soft or flange steel.	Open-hearth carbon steel.	50,000	30	0.05 acid 0.04 basic	0.05	180 deg. flat on itself.
Medium steel..	Open-hearth carbon steel	60,000	25	0.05 acid 0.04 basic	0.05	For material under 30.6 lbs. per sq. ft., 180 deg. flat on itself for longitudinal, and 180 deg. to a diam. of 1 thickness for transverse. For material 30.6 lbs. per sq.ft. and over, 180 deg. to a diam. of 1 thickness for longitudinal and 2 thicknesses for transverse.
High tensile.. steel.	Open-hearth carbon, nickel, or silicon steel	80,000	20	0.05 acid 0.04 basic	0.05	180 deg. to a diam. of 1½ thicknesses for longitudinal, and 180 deg. to a diam. of 2½ thicknesses for transverse.
Common steel.	Open-hearth or Bessemer steel.	55,000	22	No chemical analysis required.		180 deg. to a diam. of 1 thickness.

Table 16. Physical and Chemical Requirements for Steel Rods

Class	Material	Lbs. per sq. in. Min. tensile strength	Lbs. per sq. in. Min. yield point	Min. elongation, per cent	Max. amount of P., per cent	Max. amount of S., per cent	Bends
A	Open-hearth nickel or carbon steel.	75,000	40,000	23	0.04	0.045	Cold bend 180 deg. about an inner diam. equal to ½ the thickness of the test specimen for diam. up to and including 1 in., and equal the thickness for diam. over 1 in.
B	Open-hearth carbon steel.	58,000	30,000	28	0.04	0.045	Cold bend flat back through 180 deg.
C	Common steel.						

Grade C shall not be tested, unless this is considered necessary by the Inspector to show that the material is equal to common steel of commercial quality

Material for nuts of steel or iron to be used with steel bolts or studs shall show a tensile strength of at least 48,000 lbs. per sq. in. and an elongation of at least 26 per cent in 8 in. A specimen from full-sized bar or one machined to ½ in. square or ⅝ in. in diam. shall bend back cold through an angle of 180 deg. without showing signs of fracture.

Upsetting tests will only be required for material under 1 in. in diam. Specimens shall be cut about 1¼ times the diam. of the round in length and shall be required to stand hammering down cold in a longitudinal direction to about ¼ of the original length of the specimen without showing seams or other defects in the manufactured product. The number of upsetting test specimens shall equal the number of tensile test specimens, but in no case shall it be less than 1 for each nominal diameter.

Manufactured Bolts, Nuts and Studs (1915)

Workmanship of Bolts. Bolts may be hot forged or upset cold; all bolts made by cold upsetting process must be annealed after the heading operation; bolts must be free from scale, abnormal fins, or other defects.

Bending tests as specified in Table 16 shall be made on each bending test specimen, selected as specified above. In case the bolt is too short to permit this to be made on the unthreaded portion of the shank, the bolt shall be flattened out hot to a thickness equal to ¼ of its diam., and this specimen when cold shall be subjected to the required test, using the new thickness of the specimen for purpose of this test, as the diam. on which the bending tests are based.

Tensile Tests. The tensile specimens selected as specified above shall be subjected to a tensile test with the nut in place, the stress to be applied on the bearing faces of the head and nut, unless the length of the bolt is too small to permit gripping by machine. The bolt must meet the tensile strength specified for the material called for by the order, and fracture must in all cases occur in the threaded portion of the bolt. When the order calls for bolts too short to apply the foregoing tensile test, tensile-test specimens may be selected from the rods from which short bolts are to be made in number as specified above. Bolts larger than 1½ in. in diam. shall be tested by turning therefrom 1½-in. studs. These studs shall be tested in a like manner as specified for testing bolts, by fitting a 1½-in. nut at each end. In determining the tensile strength of the bolt the mean thread area will be used as representing the effective area.

Workmanship of Nuts. Nuts shall be either hot forged or cold punched from a solid bar. They must be free from scale, fins, seams, or other injurious or unsightly defects, and must have clean and smoothly threaded holes of nominal size, square to the end faces of the nuts.

Physical Test of Nuts, Steel or Iron. One-third of the specimens selected shall be **drifted cold** until they break. The fracture must indicate homogenous structure of the material. If the material is wrought iron, the fibers must run at right angles to the axis of the hole.

One-third of the specimens selected shall stand **flattening out axially, cold,** to ⅓ the original thickness without showing cracks in the case of steel or cracks of any considerable size or number in the case of iron nuts.

The other third shall stand **flattening out axially, when heated to a cherry red** in daylight, to ⅓ the original thickness without showing cracks in the case of steel or cracks of any considerable size or number in the case of iron nuts.

Table 17

	Phosphorus, max.		Sulphur, max. per cent	Tensile strength, lbs. per sq. in.	Yield point, lbs. per sq. in. min.[1]	Elongation, min. per cent in 8 in.	Elongation, min. per cent in 2 in.	Reduction of area, per cent	Cold bends without fracture
	Basic, per cent	Acid, per cent							
Plates, shapes, and bars:									
a. Bridges	0.04	0.06	0.05	55,000 to 65,000	0.5 tens. str.	$\frac{1,500,000}{\text{tens. str.}}$	22	.....	180 deg. flat under ¾ in.; 180 deg. d = t, t = ¾ to 1¼ in.; 180 deg. d = 2t, t = 1¼ in. up.
b. Buildings	0.06	0.06	.....	55,000 to 65,000	0.5 tens. str.	$\frac{1,400,000}{\text{tens. str.}}$	22	.....	
Rivet steel:									
a. Bridges	0.04	0.04	0.045	46,000 to 56,000	0.5 tens. str.	$\frac{1,500,000}{\text{tens. str.}}$	.....	.....	180 deg. flat.
b. Buildings	0.06	0.06	0.045	46,000 to 56,000	0.5 tens. str.	$\frac{1,400,000}{\text{tens. str.}}$	.....	.....	180 deg. flat.
Steel castings (Class B):									
a. Hard	0.05	0.05	0.05	80,000	0.45 tens. str.	.....	15	20	No bend test.
b. Medium	0.05	0.05	0.05	70,000	0.45 tens. str.	.....	18	25	90 deg. d = 1 in.
c. Soft	0.05	0.05	0.05	60,000	0.45 tens. str.	.....	22	30	120 deg. d = 1 in.
Steel forgings (Class C annealed):									
a. Not over 12 in.	0.05	0.05	0.05	60,000	0.5 tens. str.	.....	$\frac{1,700,000}{\text{tens. str.}}$	$\frac{2,700,000}{\text{tens. str.}}$	
b. 12 to 20 in.	0.05	0.05	0.05	60,000	0.5 tens. str.	.....	$\frac{1,600,000}{\text{tens. str.}}$	$\frac{2,520,000}{\text{tens. str.}}$	
Reinforcing steel for concrete.									

a. Structural size grade.....	0.10 B 0.05 O. H.	Noted	55,000 to 70,000	33,000	1,250,000 tens. str.	
b. Intermediate grade.....	0.10 B 0.05 O. H.	Noted	70,000 to 85,000	40,000	1,125,000 tens. str.	
c. Hard grade....	0.10 B 0.05 O. H.	Noted	80,000 min.	50,000	1,000,000 ⅔ s. str.	
d. Cold twisted bars.....	0.10 B 0.05 O. H.	Noted	Recorded only	55,000	5 per cent.	

[1] As determined by drop of beam.
d = Diameter of pin about which the specimen is bent.
t = Thickness or diameter of specimen.
B = Bessemer steel.
O. H. = Open-hearth steel.

Tests of Studs. Studs shall be tested in the same manner as prescribed for bolts.

Uses. *Class A* steel bolts are used for shaft couplings, bolts for thrust-bearing side rods, main engine framing, cylinders or valve chest covers, valve bonnets, main bearing-cap bolts, and for securing parts of machinery. *Class B* steel bolts or studs are used in pipe flanges, valves beneath floor plates, connecting rod plugs, and for structural purposes. *Class C* steel bolts are used with plates, shapes, etc., where strength is not important.

Nonferrous bolts and nuts are used where exposed to action of salt water or where nonmagnetic material is required.

Structural Steel Work

Process. All steel, except reinforcement bars shall be made by the open-hearth process. Steel for reinforcement bars may be made by either the Bessemer or the open-hearth process.

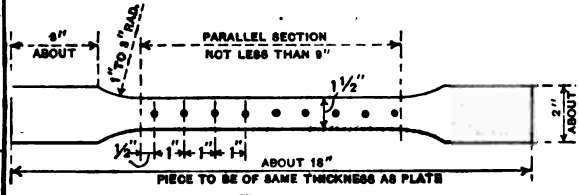


FIG. 12.

Chemical and physical properties shall conform to Table 17.

Specimens for tensile and bending tests for **plates, shapes,** and **bars** shall be cut from the finished product, and shall have both faces rolled and both edges milled to the form shown by Fig. 12, or have both edges parallel throughout; or, they may be turned to a diam. of ¾ in. for a length of at least 9 in., with enlarged ends. Test specimens or **rivet steel** shall be of the full size of the rod.

Specimens representing **steel castings** shall be made from coupons which are molded, cast, and annealed as integral parts of the castings and which are not cut from the castings until after the completion of the annealing process. Test specimens for tensile tests shall be of the form shown in Fig. 13.

Individual coupon tests and reports will be made for each heat unless otherwise elsewhere specified.

Floor Plates (1918)

Floor plates for use in engine and boiler rooms, in machinery spaces and similar locations shall be made of Bessemer or open-hearth steel.

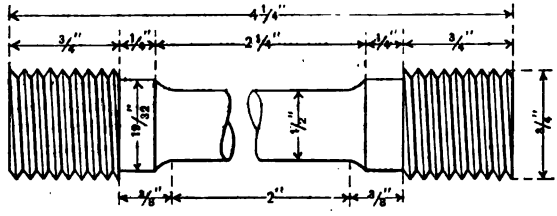


FIG. 13.

Nickel-steel Plates (1917)

Method of Manufacture. Nickel-steel plates shall be manufactured by the open-hearth process; all plates shall be oil or water tempered and annealed, and the whole of each plate shall be subjected to the same treatment at the same time.

Chemical Requirements. Plates shall contain not less than 3 per cent of nickel, not more than 0.05 per cent of phosphorus, nor more than 0.045 per cent of sulphur, and shall be of the best composition in all other respects.

Physical Requirements. All tests shall be made upon specimens taken from plates after final treatment.

Tensile Tests. One longitudinal specimen for tensile test will be taken from each plate. Each test specimen shall show a tensile strength of at least 80,000 lbs. per sq. in. and an elongation in 2 in. of at least 27 per cent. All tensile test specimens shall be taken from a location at the end of the plate nearest the bottom of the ingot from which the plate was rolled.

Cold Bending Tests. Two longitudinal pieces cut from different plates in each heat, shall be doubled cold around a mandrel, the diam. of which is not more than the thickness of the piece tested, without showing any cracks. The ends of the pieces shall be parallel after bending.

Flat Black and Galvanized, and Corrugated Galvanized Sheet Steel (1916)

Quality shall be as follows:

Black sheets shall be made of a high-grade steel of a quality that will stand double seaming without cracking; shall be commercially flat and free from scale, pits, laminations, buckles or any other injurious defects.

Galvanized sheets shall be thoroughly and evenly galvanized with zinc, in a bath which shall show not less than 98 per cent pure zinc; coating produced shall be of a bright appearance and shall not flake or peel off when scraped with a knife, or when the sheet is bent sharply at right angles. Sheets shall be free from blisters, ragged edges, buckles, laminations, and other injurious defects.

Steel flat galvanized sheets shall be made of the same quality of steel as flat black sheets; shall stand double seaming without cracking; and shall be commercially flat.

Steel corrugated galvanized sheets shall be of a good grade of steel, but shall not be subjected to the double-seaming test.

Cold-rolled or Cold-drawn Machinery Steel Rods and Bars (1917)

Physical and chemical requirements of cold-rolled or cold-drawn steel shall be in accordance with tables 18 and 19.

Steel, Welding, Wire and Rod (1917)

Uses. This material is suitable for electrical and autogenous welding of steel castings, forgings, etc.

Chemical composition shall conform to Table 20.

Table 18

	Ultimate tensile strength lbs. per sq. in.	Min. yield point, per cent	Type of test pieces	Minimum elongation, per cent
Class A:				
Steel rods and bars—				
Under ¼ in. in diam. or thickness	[1]	[1]	[1]	[1]
¼ in. to ⅜ in. inclusive, in diam. or thickness	80,000–110,000	75 ult.	3	12 in 2 in.
Over ⅜ in. to 1½ in. inclusive, in diam. or thickness	75,000–100,000	75 ult.	3	10 in 8 in.
			1	16 in 2 in.
Over 1½ in. in diam. or thickness	70,000–90,000	70 ult.	3	14 in 8 in.
			1	18 in 2 in.
Class B:				
Steel for automatic screw or bolt machines	70,000	50	[2]	6 in 8 in
				10 in 2 in.

[1] No physical tests required. [2] As for class A.

Table 19

	Maximum amount of		Cold bend
	Phosphorus per cent	Sulphur, per cent	
Class A:			
Steel rods and bars	0.06	0.06	180 deg. to 3 diam.
Class B:			
Steel for automatic screw or bolt machines	0.13	0.15	120 deg. to 3 diam.

Table 20

Grade	Carbon, per cent	Silicon, (max.) per cent	Manganese per cent	Phosphorus (max.) per cent	Sulphur (max.) per cent	Nickel, per cent	Copper (max.) per cent
Mild steel or iron	0 to 0.12	0.10	0 to 0.25	0.04	0.04		
Nickel steel	0.20 to 0.30	0.20	0.40 to 0.60	0.04	0.04	2 to 3.5	0.40

NOTE.—Mild steel or iron must not contain more than a trace of copper, nickel, or chromium.

Spring Steel, Rolled and Drawn (1917)

Process of Manufacture. Spring steel may be manufactured by either the open-hearth, crucible or electric-furnace process.

Classes. Rolled spring steel shall be furnished in either of the two following classes, as specified: **Class A,** carbon, 0.70 to 0.90 per cent; **class**

B, carbon, 0.90 to 1.10 per cent. Drawn spring steel shall be furnished in either of the two following classes, as specified; **Class C**, a carbon drawn acid steel; **class D**, an annealed, carbon steel, not tempered.

Chemical Requirements. The required percentage chemical contents of the above four classes shall be as follows: **Rolled spring steel class A**; carbon, 0.70 to 0.90; manganese, 0.25 to 0.50; silicon (max.), 0.25; sulphur (max.), 0.05; phosphorus (max.), 0.04. **Rolled spring steel class B**; carbon, 0.90 to 1.10; manganese, 0.25 to 0.50; silicon (max.), 0.25; sulphur (max.), 0.05; phosphorus (max.), 0.04. In both classes of rolled spring steel vanadium or other elements may be used, in which case only the phosphorus and sulphur requirements as given above need be adhered to. The bidder shall state, however, in his bid whether he proposes to furnish material meeting the composition specified or alloy steel; if the latter, what approximate composition, naming all elements.

The percentage chemical contents of **drawn spring steel class C** will be: Carbon 0.55 to 0.65; manganese, 1 to 1.25; silicon, 0.12 to 0.20; sulphur (max.), 0.05; phosphorus (max.), 0.04. For **drawn spring steel class D**: Carbon, 0.78 to 0.85; manganese, 0.40 to 0.55; silicon, 0.12 to 0.20; sulphur (max.), 0.05; phosphorus (max.), 0.04.

Tests. From each lot of 20 bars or fraction thereof, of the same size, made from the same open-hearth melt or furnace charge, one bar shall be selected at random and subjected to a nick-and-break test. The nick-and-break test will be made on the full-size specimen.

A specimen, when nicked and broken, shall present a fine uniform grain

Steel Slabs, Blooms, Billets, and Bars for Reforging (1918)

Process. Ingots from which billets are made shall be manufactured by the open-hearth, crucible, or electric-furnace process. Billets shall be rolled or forged from ingots having a cross-sectional area of at least 4 times the cross-sectional area of the finished billet up to 18 in.; over 18 in. the cross-sectional area of the ingot shall be not less, in any case, than that required for an 18-in. billet and that area should be increased as necessary in order that the ingot shall in each case have a cross-sectional area not less than 3 times that of the finished billet.

Discard. Sufficient discard shall be taken from each ingot to insure freedom from piping and undue segregation. Such discard shall be not less than 5 per cent of the total weight of the ingot taken from the bottom and 30 per cent from the top if top poured, and 5 per cent from the bottom and 20 per cent from the top if bottom poured or fluid compressed. When forgings are to be made from bored fluid-compressed ingots at least 3 per cent of the total weight of the ingot shall be discarded from the bottom and at least 10 per cent of the total weight from the top.

Surface and Other Defects. All billets shall be free from seams, pipes, flaws, cracks, blowholes, hard spots, sand, foreign substances, excessive slag, and all other defects affecting their value.

Chemical Requirements. Billets will be accepted on chemical analysis only and shall be within the requirements of the grade as specified in Table 21.

Treatment. No definite **mechanical properties** are required for material covered by these specifications. The manufacturer shall, if requested, furnish the inspector with a description of the heat treatment which is *guaranteed* to produce mechanical qualities as given in Table 22 for the respective grades.

Table 21

Grade	Carbon, per cent	Manganese, per cent	Sulphur, per cent max.	Phosphorus, per cent max.	Nickel, per cent min.
Alloy No. 1	0.50 max.	.....	0.035	0.03	
Alloy No. 2	0.50 max.	.....	0.035	0.03	
Alloy No. 3	0.45 max.	.....	0.04	0.04	
HG	0.30–0.45	0.40–0.80	0.045	0.04	3.0
An	0.25–0.45	0.40–0.80	0.045	0.04	3.0
Ac	0.40–0.60	0.40–0.80	0.045	0.04	
B–s (special)	0.40–0.60	0.40–0.80	0.045	0.04	
B	0.25–0.40	0.45–0.70	0.045	0.04	
C	.....	.....	0.07	0.07	
S	0.08–0.15	0.30–0.50	0.04	0.04	

Table 22

Minimum values

Grade	Tensile strength (lbs. per sq. in.)	Proportional limit or yield point (lbs. per sq. in.)	Longitudinal elongation (per cent in 2 in.)	Transverse elongation (per cent in 2 in.)	Cold bend
Alloy No. 1	170,000	140,000	10	8	None required.
Alloy No. 2	128,000	105,000	20	15	None required.
Alloy No. 3	105,000	80,000	20	18	180 deg. to inner diam. of 1 in.
HG	95,000	65,000	21	18	180 deg. to inner diam. of 1 in.
An	80,000	50,000	25	21	180 deg. to inner diam. of 1 in.
Ac	80,000	45,000	25	21	180 deg. to inner diam . of 1 in.
B—s (special)	75,000	40,000	22	19	180 deg. to inner diam. of 1 in.
B	60,000	30,000	30	25	180 deg. to inner diam. of ½ in.
C	50,000	.....	18	15	None required.
S	48,000	24,000	32	28	180 deg. flat.

Hot Rolled or Forged Alloy Steel (Chrome Vanadium)

Method of Manufacture. Special alloy steel bought under this specification must be manufactured either by the electric, crucible, or open-hearth process, depending on which process is specified in the requisition.

Slabs, Blooms, and Billets. All slabs, blooms, billets, or other forgings of special alloy steel shall be rolled or forged from ingots whose cross-section is at least twice that of the finished slab, bloom, or billet, and from ingots from which a discard of at least 5 per cent of the total weight has been taken from the bottom and 30 per cent from the top, if top poured; 5 per cent from the bottom and 20 per cent from the top if the ingot has been bottom poured or fluid compressed.

Chemical Composition. Special alloy steel to be bought under this specification must conform to the chemical composition given in Table 23.

Table 23

	Per cent	Per cent desired
Carbon	0.35–0.45	0.40
Sulphur	Not to exceed 0.04	.....
Manganese	0.50–0.80	0.65
Vanadium	0.12–0.25	0.18
Chromium	0.70–1.10	0.90
Phosphorus	Not to exceed 0.04	.....

Steel Forgings for Hulls, Engines, and Ordnance (1918)

Uses. Table 24 gives the uses to which the various grades of steel forgings are adapted.

Table 24

Material	Class	Purpose for which used
Engine forgings..	HG	Bolts and studs for all moving parts of main engines; shaft couplings, main-bearing caps, thrust-bearing side rods, main-engine framing, and moving parts of circulating pumps; connecting rods, caps, and bolts; eccentric rods; main circulating-pump engine working parts; piston rods; suspension links and link blocks; valve stems.
Engine forgings..	An	Coupling bolts; crossheads and slippers; crank, thrust, line, stern tube, tail, and propeller shafts; main-bearing cap bolts; outboard coupling; reverse arms and blocks; rotor shaft; thrust-bearing side rods; turning engine worm; working parts, reversing gear; working parts, pumps.
Engine forgings..	B	Bearer bars; turbine rotor shafts; engine columns and tie-rods; H. P. relief-valve stems; main steam valve stems; main-bearing cap bolts; piston-rod nuts; piston-valve followers; pipe flanges; rotor drum and wheel; sole-plate wedges; swivel pins for crosshead; working levers and gears.
Engine forgings..	C	Gland for cylinder liners; small parts for eccentrics; uptake and smoke-pipe forgings.

All steel forgings will be without welds and free from laminations.

All Class HG steel forgings 5 in. or more in diam. will have axial holes of approved diam.

The working parts of all auxiliary machinery, unless otherwise specified, will be made of class A nickel-steel forgings.

All steel pistons rods and valve stems of auxiliary engines will be oil tempered.

All steel joint pins of valve gear will be hardened and ground to true cylindrical surfaces.

All small pins of working parts will be well case-hardened and ground.

Drop forgings of parts subject to pressure or strains must meet the physical and chemical requirements of Class B forgings. Drop forgings of parts not subject to pressure will be required to have surface inspection only.

Process. Steel for forgings shall be made by the open-hearth, crucible, or electric process. They shall be forged or rolled from ingots, the original cross-sectional area of which is at least 4 times that of the finished forgings. Class C forgings may be made by the Bessemer process.

If bored ingots are used the wall of the ingot shall be reduced to at least ½ of its original thickness, or the reduction of area shall be at least 4 to 1. Palms, flanges, and other enlargements on shafting, need not be reduced in the ratio of 4 to 1, but shall be reduced in a ratio of not less than 1.7 to 1.

Discard. Sufficient discard shall be taken from each ingot to insure freedom from piping and undue segregation. Such discard shall be not less than 5 per cent of the total weight of the ingot taken from the bottom and 30 per cent from the top if top poured, and 5 per cent from the bottom and 20 per cent from the top if bottom poured or fluid compressed. When forgings are to be made from bored fluid compressed ingots at least 3 per cent of the total weight of the ingot shall be discarded from the bottom and at least 10 per cent of the total weight from the top.

Surface and Other Defects. All forgings shall be free from seams, pipes, flaws, cracks, scale, fins, porosity, hard spots, sand, foreign substances, excessive slag, and all other defects affecting their value.

Table 25

Class	Material	Maximum values			Minimum values						Cold bend without cracking
		Carbon, per cent	Sulphur, per cent	Phosphorus, per cent	Tensile strength, lbs. per sq. in.	Proportional limit or yield point, lbs. per sq. in.	Elongation in 2 in.		Reduction of area		
							Long., per cent	Trans., per cent	Long., per cent	Trans., per cent	
Alloy No. 1		0.50	0.035	0.03	170,000	140,000	10	8	35	20	None required.
Alloy No. 2		0.50	0.035	0.03	120,000	105,000	20	15	50	40	180 deg. to inner diam. of 1 in.
Alloy No. 3		0.45	0.04	0.04	105,000	80,000	20	18	50	40	180 deg. to inner diam. of 1 in.
HG	Nickel steel (nickel not under 3 per cent).	0.45	0.045	0.04	95,000	65,000	21	18	45	30	180 deg. to inner diam. of 1 in.
An	Nickel steel (nickel not under 3 per cent).	0.45	0.045	0.04	80,000	50,000	25	21	45	30	180 deg. to inner diam. of 1 in.
Ac	Carbon steel	0.60	0.045	0.04	80,000	45,000	25	21	45	30	180 deg. to inner diam. of 1 in.
B–s (special)	Carbon steel	0.50	0.045	0.04	75,000	40,000	22	19	40	30	180 deg. to inner diam. of 1 in.
B	Carbon steel	0.40	0.045	0.04	60,000	30,000	30	25	45	40	180 deg. to inner diam. of ½ in.
C	Carbon steel	....	0.07	0.07	50,000	.....	18	15			

For alloy steels No. 1 and No. 2 an electric furnace steel is desired.

Test specimens for cold bend tests shall be cut in the same direction as the tensile test specimens, *i.e.*, if longitudinal specimens are used in making the tensile tests, longitudinal specimens shall also be used in the cold bend tests, etc.

Class C forgings shall not be tested.

Chemical and Physical Properties. The respective classes of forgings shall have the properties given in Table 25.

Test Specimens shall fairly represent the average strength of the material and shall be taken at a point which has received the average amount of reduction. They should, in general, be located in that part of the forging which includes the uppermost part of the ingot as cast.

Types of Forgings. Covered by General Requirements. Parts of gun recoil system, including recoil and spring cylinders, piston rods; also nuts and bolts for same. Gun elevating and training gear shafts, worms, pinions, keys, and feathers, etc., for same. Parts of gun mounts, such as gun yokes, trunnion-bearing caps, floating supports, and other trunnion parts, trunnion bands, and slides. Parts of torpedo tubes and ordnance appurtenances, shafting, rammer, links, etc. Turret rollers, turret-turning pinions, turret racks, and tracks. Armor keys. Holding-down bolts for gun mounts and turret tracks. Rudder frames and rudder stocks. Anchor crane stocks.

Metallographic examination for purposes of record and information may be made on any forging, but such examination shall not cause rejection if otherwise satisfactory except in the case of propelling machinery shafting and couplings for vessels. All propelling machinery shafting and couplings for vessels shall be subjected to photomicrographic examination, and acceptance shall depend on the result of such examination.

Metallographic Requirements. Specimens when examined at a magnification of 100 shall show a homogeneous structure—*i.e.*, one in which the normal constituents are evenly distributed, free from decided segregation of any constituents, ingotism, or excessive impurities. Grains with any dimension greater than $\frac{1}{2}$ in. in diam. at 100 diam. ($\frac{1}{200}$ in.) will render the forging liable to rejection. In these specifications the term grain shall be taken to include the constituents which together made up the austenite cell from which the constituents were last evolved. If the outlines of this cell are not clear, the uninterrupted areas of the predominant metallographic constituent shall be used in determining the grain size.

Steel Castings (1918)

Uses. *Grade A* is intended for all important parts subject to crushing stresses or surface wear only, such as hawse pipes, chain pipes, turret roller paths, engine guides, slippers, etc.

Grade B is intended for parts subject to tensile or vibratory stresses, such as stems, sternposts, stern tubes, rudder frames, struts, engine bedplates, cylinders, gun-mount stands, carriages, slides, and other parts subject to the shock of recoil.

Grade C is intended for gun mounts, such as brackets, levers, wheels, etc., not subject to shock of recoil, and for commercial fittings where structural strength and separation of watertight compartments are not involved, such as pipe flanges (other than bulkhead and deck), cagemast fittings, stowage lugs and clips, hinges for doors and hatches where watertightness is not involved, etc.

Grade D is intended for the same general purpose as *Grade B*, but where greater strength is required with equal ductility.

Grade E is intended where design will permit its use for castings for similar purposes as *Grade F*.

Grade F is intended for castings for gun yokes, gun-mount stands, carriages, slides, deck lugs, etc., of large size.

Cast-steel H.P. pipe fittings will be Class *B*. Cast-steel L.P. pipe fittings may be Class *C*.

Castings forming a bearing surface will be of special mixture to secure a close-grained hard surface.

All flanges, collars, and offsets will have well-rounded fillets, and ample fillets will be provided in other places where necessary.

The metal of important castings subject to pressure must be homogeneous and the thickness of no part will be allowed to deviate materially from the dimensions shown on the approved drawings. Special steps will be taken to guard against porosity.

The Chemical and Physical Properties of steel castings shall be in accordance with Table 26.

Table 26

Grade	Chemical composition, maximum				Physical requirements				
	Tropenas		Open-hearth, electric furnace, or crucible		Minimum tensile strength, lbs. per sq. in.	Minimum yield point, lbs. per sq. in.	Minimum elongation, per cent, in 2 in.	Minimum reduction of area, per cent	Cold bend about a diam. of 1 in., deg.
	P. per cent	S. per cent	P. per cent	S. per cent					
F	0.05	0.05	0.05	0.05	85,000	53,000	22	35	120
E	0.05	0.05	0.05	0.05	80,000	44,000	22	30	120
A	0.05	0.05	0.05	0.05	80,000	45 per cent of tensile strength obtained.	17	20	90
D	0.05	0.05	0.05	0.05	70,000		22	30	120
B	0.06	0.08	0.06	0.06	60,000		22	30	120
C	0.06	0.08	0.06	0.07					

Classes *F* and *E* may contain nickel or other alloying metals.

Welding. Surface defects and cavities which are of more than minor importance shall not be so welded except by permission of the Inspector in charge of the district. In no case will any welding be allowed on steam piping or any other casting used in connection with steam piping or subjected to steam pressure, nor in the following ordnance castings: Gun yokes and slides in region of the trunnions, elevating gear lugs, and recoil cylinder and spring cylinder bearings for same.

Treatment. Rapid cooling of the castings by the rise of water, brine, oil, or air blast will be permitted only with the specific approval of the Inspector.

All castings shall be carefully annealed so that any fracture shall show to the eye a fine-grained structure. The grain size may be determined by the appearance of the fracture of the test coupons representing the casting in question.

Test Specimens. Coupons from which test specimens are to be taken shall, whenever practicable, be cast on the body of the casting.

Percussive Test. Large castings when required by the inspector shall be subjected to hammer tests, as follows: The castings shall be suspended and hammered all over with a hammer weighing not less than 7½ lbs. If, under such treatment, cracks, flaws, defects, or unsoundnesses are indicated which render the castings unfit for service, such castings shall be rejected.

Metallographic Examination. For the purposes of information and record only, and having no bearing upon the acceptance or rejection of the material represented thereby, from two castings weighing 200 lbs. or over

in each heat containing two or more such castings, test pieces shall be taken for metallographic examination; in case a heat should contain only one casting weighing 200 lbs. or over, one test piece shall be taken from that casting. These test pieces shall be sections of the broken tensile test bars.

Carbon Tool Steel

Uses. *Class* 1, drill rods, lathe and planer tools, and tools requiring keen cutting edges combined with great hardness, such as drills, taps, and reamers.

Class 2, milling cutters, mandrels, trimmer dies, threading dies, and general machine-shop tools requiring a keen cutting edge combined with hardness.

Class 3, pneumatic chisels, punches, shear blades, etc., and in general tools requiring hard surface with considerable tenacity.

Class 4, rivet sets, hammers, cupping tools, smith tools, hot drop-forge dies, etc., and in general tools which require great toughness combined with the necessary hardness.

Class 5, gauges and articles of all types which require a medium degree of hardness for the purpose of resisting wear and abrasion, combined with a minimum of deformation during the process of hardening.

Method of Manufacture. The tool steels must be made in either the electric or crucible furnace and must be of homogeneous composition. The bars or rods shall be forged or rolled accurately to the dimensions specified, and must be free from seams, checks, and other physical defects. They must be delivered annealed and, unless otherwise specified, in commercial lengths. Short pieces will not be accepted. Drill rods must be coated with a rust preventive.

Chemical Requirements of carbon tool steel must be in accordance with Table 27.

Table 27

	Class 1, per cent of limit		Class 2, per cent of limit		Class 3, per cent of limit		Class 4, per cent of limit		Class 5, per cent of limit	
	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
Carbon	1.35	1.20	1.10	1.00	0.90	0.80	0.80	0.70	1.05	0.80
Manganese	0.35	0.15	0.35	0.15	0.35	0.15	0.35	0.15	2.00	1.25
Phosphorus	0.015	0.00	0.015	0.00	0.02	0.00	0.02	0.02	0.00	0.00
Silicon	0.40	0.10	0.40	0.10	0.40	0.10	0.40	0.10	0.50	0.25
Sulphur	0.02	0.00	0.02	0.00	0.025	0.00	0.025	0.00	0.025	0.00
Vanadium	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	[1]	[1]
Other elements[2]										

[1] Optional. [2] As submitted.

Inspection. Access to the records of the manufacturers to identify heat numbers and other information of assistance to the Inspector shall be allowed. The Inspector will assign a lot number to all the material submitted for inspection at any one time and will forward to the Engineer Officer, with forms provided for the purpose, samples for chemical analysis on the basis of one sample for each 500 lbs. or fraction thereof of material submitted. The sample to be about 1½ ounces of finely divided drillings.

Samples for Selective Test. Each bidder shall furnish with his proposal sample bars of tool steel, stamped as specified, for the "selective test." The relation of the results obtained from the tests conducted as provided for under the heading "Selective tests" and the price of the material determine the selective factor. At the discretion of the bureau concerned, bidders who

have previously submitted samples identical with material offered may be classified according to the selective factor previously determined.

The dimensions of the sample bars shall be as follows:

Class 1. $1\frac{1}{16}$ inch diameter rod, 2 $\frac{1}{2}$ feet long.

Class 2. $1\frac{1}{16}$ inch diameter rod, $2\frac{1}{2}$ feet long.

Class 3. $\frac{3}{4}$ inch octagon rod, 5 feet long.

Class 4. 2 inch diameter rod, 2 feet long.

Class 5. 1 piece $5\frac{1}{2}$ inches in diameter, 9 inches long; 1 piece 2 inches in diameter, 12 inches long.

Treatment of Samples. Each bidder will state in his proposal, if he considers it necessary to do so, the treatment to which the material must be subjected in order to get, in his opinion, the best results.

Selective Tests. *Class* 1. Five $\frac{7}{16}$ in. diam. 4-tooth facing mills will be made from the sample rod and tested on a piece of $\frac{5}{8}$-in. ship's plate without lubricant. Each mill will be run until it is so dull that it breaks either in the teeth or in the shank. The depth of cut will be 0.08 in., the revolutions per minute of the mill will be 370, and the feed of material 20 in. per min. A record will be made of the length of time each mill operates.

Class 2. Five $\frac{7}{16}$ in. diam. 4-tooth facing mills will be made from the sample rod and tested on a piece of $\frac{5}{8}$-in. ship's plate without lubricant. Each mill will be run until it is so dull that it breaks either in the teeth or in the shank. The depth of cut will be 0.08 in., the revolutions per minute of the mill will be 370, and the feed of material 20 in. per min. A record will be made of the length of time each mill operates.

Class 3. Five $\frac{1}{2}$-in. pneumatic chisels will be made from the sample bar. Each chisel will be tested on a nickel-steel plate with a cut of $\frac{1}{16}$ in. deep. A record will be made of the distance each chisel cuts with a lubricant before it is ruined

Class 4. Two $\frac{1}{2}$-in. rivet sets will be made from the sample bar. A record will be made of the condition of the sets after a certain number of rivets have been driven.

Tungsten Tool Steel

Uses. *Class* 1, drill rods, lathe and planer tools milling-machine tools, and in general all tools for which high-speed steel is used.

Class 2, lathe and planer tools and general machine-shop tools which require a keen and durable cutting edge. This material is adapted for brass finishing and other finishing work.

Method of Manufacture, see Carbon Tool Steel.

Chemical Requirements of tungsten tool steel must be in accordance with Table 28.

Table 28

	Class 1, per cent of limit		Class 2, per cent of limit	
	Maximum	Minimum	Maximum	Minimum
Carbon	0.75	0.45	1.50	1.35
Chromium	5.00	2.50	0.00	0.00
Manganese	0.45	0.15	0.35	0.15
Phosphorus	0.02	0.00	0.02	0.00
Silicon	0.40	0.10	0.60	0.10
Sulphur	0.03	0.00	0.03	0.00
Tungsten	20.00	12.00	5.00	3.00
Vanadium	2.00	0.70		
Other elements, as submitted				

Table 29. Condensed Chemical and Physical Requirements of

Material	Class	Carbon	Manganese	Phosphorus		Sulphur	Silicon
Steel boiler tubes	..	0.08–0.18	0.30–0.50	0.04	.....	0.045	.....
Welded steel pipe	..	.....	.....	.....	.....	.....	.....
Flange steel	..	.....	0.30–0.60	0.05a	0.04b	0.05	.....
Fire box steel	..	0.12–0.25	0.30–0.50	0.04a	0.35b	0.04	.....
Boiler rivet steel	..	.....	0.30–0.50	0.04	.....	0.045	.....
Structural steel for buildings	..	.....	.....	.....	.....	.....	.....
Rivet steel for buildings	..	.....	.....	0.06	.....	0.045	.....
Structural steel for bridges	..	.....	.....	0.06a	0.04b	0.05	.....
Rivet steel for bridges	..	.....	.....	0.04a	0.04b	0.045	.....
Structural steel for ships	..	.....	.....	0.06a	0.04b	0.05	.....
Rivet steel for ships	..	.....	.....	0.06a	0.04b	0.045	.....
Structural nickel steel	..	0.45 max.	0.70 max.	0.05a	0.04b	.....	.....
Structural rivet nickel steel	..	0.30 max.	0.60 max.	0.04a	0.03b	.....	.....
Steel rails, Bessemer	..	0.37–0.55	0.80–1.10	0.10	.....	.....	0.20
Steel rails, open-hearth	..	0.50–0.75	0.60–0.90	0.04	.....	.....	0.20
Steel tires	..	0.50–0.85	0.75 max.	0.05	.....	0.05	0.15–0.35
Steel bars for springs	..	0.85–1.15	0.25–0.50	0.05a	0.04b	0.05	.....
Silico-manganese steel bars for springs	..	0.45–0.65	0.50–0.80	0.05a	0.045b	0.045	1.50–2.10
Chrome-vanadium steel bars for springs	..	0.45–0.65	0.50–0.90	0.05a	0.04b	0.05	.....
Blooms, billets and slabs for carbon steel forgings.	A	0.08–0.18	0.30–0.50	0.045	.....	0.05	.....
	B	0.15–0.25	0.30–0.50	0.045	.....	0.05	.....
	C	0.25–0.38	0.40–0.60	0.045	.....	0.05	.....
	D	0.38–0.52	0.40–0.60	0.045	.....	0.05	.....
	E	0.45–0.60	0.45–0.70	0.045	.....	0.05	.....
Carbon steel forgings for locomotives, unannealed.	..	.....	0.40–0.70	0.05	.....	0.05	.....
	...	.....	0.40–0.70	0.05	.....	0.05	.....
Carbon steel forgings for locomotives, annealed.	..	.....	0.40–0.70	0.05	.....	0.05	.....
	..	.....	0.40–0.70	0.05	.....	0.05	.....
	..	.....	0.40–0.70	0.05	.....	0.05	.....

the American Society for Testing Materials for Steel.

Nickel	Chromium	Vanadium	Diameter, Over	Diameter, To include	Tensile strength, thousand lbs. per sq. in.	Yield point, thousand lbs. per sq. in.	Elongation in 8 inches, per cent	Elongation in 2 inches, per cent	Reduction of area, per cent: Inverse ratio	Reduction of area, per cent: Minimum
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	50	30	18	....	.....	.....
.....	.....	.....	..	..	55-65	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	52-62	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	45-55	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	55-65	0.5 T. S.	$\frac{1,400,000}{\text{T. S.}}$	22	.....	.....
.....	.....	.....	..	..	46-56	0.5 T. S.	$\frac{1,400,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	55-65	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	46-56	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	58-68	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
.....	.....	.....	..	..	55-65	0.5 T. S.	$\frac{1,500,000}{\text{T. S.}}$	....	.....	.....
3.25 min.	.....	.....	..	..	85-110	50-55	$\frac{1,500,000}{\text{T. S.}}$	16-20	.....	25-35
3.25 min.	.....	.....	..	..	70-80	45	$\frac{1,500,000}{\text{T. S.}}$	....	.....	40
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	105-125	.....	.....	8-12	.....	12-16
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	0.80-1.10	0.15	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	..	.....	.....	.....	....	.....	.....
.....	.....	.....	..	8	75	0.5 T. S.	.....	18	$\frac{2,200,000}{\text{T. S.}}$	24
.....	.....	.....	8	12	75	0.5 T. S.	.....	17	$\frac{2,000,000}{\text{T. S.}}$	22
.....	.....	.....	..	8	80	0.5 T. S.	.....	20	$\frac{2,800,000}{\text{T. S.}}$	32
.....	.....	.....	8	12	80	0.5 T. S.	.....	19	$\frac{2,640,000}{\text{T. S.}}$	30
.....	.....	.....	12	20	80	0.5 T. S.	.....	18	$\frac{2,400,000}{\text{T. S.}}$	28

Table 29. Condensed Chemical and Physical Requirements of

Material	Class	Carbon	Manganese	Phosphorus		Sulphur	Silicon
Quenched and tempered carbon steel axles, shafts, etc., for locomotives and cars.	..	0.25–0.60	0.40–0.70	0.05	.....	0.05	.....
	..	0.35–0.60	0.40–0.70	0.05	.....	0.05	.....
	..	0.35–0.65	0.40–0.70	0.05	.....	0.05	.....
	..	0.35–0.70	0.40–0.70	0.05	.....	0.05	.....
Carbon and alloy steel forgings.	A	.....	0.30–0.55	0.05a	0.05b	0.05	.....
	B	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	B	.....	0.40–0.80	0.05a	0.05b	0.05	.....
Carbon and alloy steel forgings, annealed.	C	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	C	.....	0.40–0.80	0.05a	0.05b	0.05	.....
Carbon and alloy steel forgings, untreated.	D	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	D	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	D	.....	0.40–0.80	0.05a	0.05b	0.05	.....
Carbon and alloy steel forgings, annealed.	E	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	E	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	E	.....	0.40–0.80	0.05a	0.05b	0.05	.....
Carbon and alloy steel forgings, annealed.	F	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	F	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	F	.....	0.40–0.80	0.05a	0.05b	0.05	.....
Carbon and alloy steel forgings, quenched and tempered.	G	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	G	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	G	.....	0.40–0.80	0.05a	0.05b	0.05	.....
	G	.....	0.40–0.80	0.05a	0.05b	0.05	.....

the American Society for Testing Materials for Steel. (*Continued*)

Nickel	Chromium	Vanadium	Diameter Over	Diameter To Include	Tensile Strength, thousand lbs. per sq. inch	Yield point, thousand lbs. per sq. in.	Elongation in 8 inches per cent	Elongation in 2 inches per cent	Reduction of area, per cent: Inverse ratio	Reduction of area, per cent: Minimum
.....	.....	.....	..	4	90	55	$\frac{2,100,000}{T.S.}$	20.5	$\frac{4,000,000}{T.S.}$	39
.....	.....	.....	4	7	85	50	$\frac{2,100,000}{T.S.}$	20.5	$\frac{3,800,000}{T.S.}$	39
.....	.....	.....	7	10	85	50	$\frac{1,900,000}{T.S.}$	19.5	$\frac{3,600,000}{T.S.}$	37
.....	.....	.....	10	20	82.5	48	$\frac{1,800,000}{T.S.}$	19	$\frac{3,400,000}{T.S.}$	36
.....	.....	.....	..	all sizes	47-60	0.5 T.S.	$\frac{1,500,000}{T.S.}$	....	$\frac{2,500,000}{T.S.}$	
.....	.....	.....	..	12	60	0.5 T.S.	$\frac{1,550,000}{T.S.}$	22	$\frac{2,400,000}{T.S.}$	35
.....	.....	.....	12	20	60	0.5 T.S.	$\frac{1,480,000}{T.S.}$	21	$\frac{2,220,000}{T.S.}$	32
.....	.....	.....	..	12	60	0.5 T.S.	$\frac{1,700,000}{T.S.}$	25	$\frac{2,700,000}{T.S.}$	38
.....	.....	.....	12	20	60	0.5 T.S.	$\frac{1,600,000}{T.S.}$	24	$\frac{2,520,000}{T.S.}$	36
.....	.....	.....	..	8	75	0.5 T.S.	$\frac{1,600,000}{T.S.}$	18	$\frac{2,200,000}{T.S.}$	24
.....	.....	.....	8	12	75	0.5 T.S.	$\frac{1,500,000}{T.S.}$	17	$\frac{2,000,000}{T.S.}$	22
.....	.....	.....	12	20	75	0.5 T.S.	$\frac{1,400,000}{T.S.}$	16	$\frac{1,800,000}{T.S.}$	20
.....	.....	.....	..	8	75	0.5 T.S.	$\frac{1,800,000}{T.S.}$	20	$\frac{2,800,000}{T.S.}$	33
.....	.....	.....	8	12	75	0.5 T.S.	$\frac{1,725,000}{T.S.}$	19	$\frac{2,640,000}{T.S.}$	31
.....	.....	.....	12	20	75	0.5 T.S.	$\frac{1,650,000}{T.S.}$	18	$\frac{2,400,000}{T.S.}$	29
.....	.....	.....	..	8	80	0.5 T.S.	$\frac{1,800,000}{T.S.}$	20	$\frac{2,800,000}{T.S.}$	32
.....	.....	.....	8	12	80	0.5 T.S.	$\frac{1,725,000}{T.S.}$	19	$\frac{2,640,000}{T.S.}$	30
.....	.....	.....	12	20	80	0.5 T.S.	$\frac{1,650,000}{T.S.}$	18	$\frac{2,400,000}{T.S.}$	28
.....	.....	.....	..	4	90	55	$\frac{2,100,000}{T.S.}$	20.5	$\frac{4,000,000}{T.S.}$	39
.....	.....	.....	4	7	85	50	$\frac{2,000,000}{T.S.}$	20.5	$\frac{3,800,000}{T.S.}$	39
.....	.....	.....	7	10	85	50	$\frac{1,900,000}{T.S.}$	19.5	$\frac{3,600,000}{T.S.}$	37
.....	.....	.....	10	20	82.5	48	$\frac{1,800,000}{T.S.}$	19	$\frac{3,400,000}{T.S.}$	36

Table 29. Condensed Chemical and Physical Requirements of

Material	Class	Carbon	Manganese	Phosphorus		Sulphur	Silicon
Carbon and alloy steel forgings, annealed.	H	.....	0.40–0.80	0.04a	0.04b	0.05	.....
	H	.....	0.40–0.80	0.04a	0.04b	0.05	.....
Carbon and alloy steel forgings, quenched and tempered.	I	.....	0.40–0.80	0.04a	0.04b	0.05	.....
	I	.....	0.40–0.80	0.04a	0.04b	0.05	.....
	I	.....	0.40–0.80	0.04a	0.04b	0.05	.....
	I	.....	0.40–0.80	0.04a	0.04b	0.05	.....
Chrome-vanadium	K	0.28–0.42	0.40–0.70	0.05a	0.04b	0.05	.....
or chrome-nickel	..	.....	.....	.....	.....	.....	.....
or chromium-steel	..	0.28–0.42	0.40–0.70	0.05a	0.04b	0.05	.....
	..	.....	.....	.....	.....	.....	.....
	..	0.28–0.42	0.40–0.70	0.05a	0.04b	0.05	.....
Steel castings	A	0.30 max.	.....	0.06	.....	.....	.....
Steel castings, hard	B	.....	.....	0.05	.....	0.05	.....
Steel castings, medium	..	.....	.....	0.05	.....	0.05	.....
Steel castings, soft	..	.....	.....	0.05	.....	0.05	.....
Cold rolled steel axles	..	0.40 max.	0.40–0.80	0.05	.....	0.05	.....
Chrome-vanadium	L	0.40 max.	0.40–0.80	0.05	.....	0.05	.....
or chrome-nickel	..	0.40 max.	0.40–0.80	0.05	.....	0.05	.....
or chromium-steel	..	0.40 max.	0.40–0.80	0.05	.....	0.05	.....
Chrome-vanadium	M	0.35–0.50	0.50–0.90	0.04	.....	0.05	.....
or chrome-nickel steel	..	0.35–0.50	0.50–0.90	0.04	.....	0.05	.....

In this table, the letters "a," "b," and "T.S." respectively represent "acid steel," "basic steel," and "tensile strength."

Under "blooms, billets and slabs for carbon steel forgings," the classification is as follows:—

Class A, for welding and case hardening.
Class B, for case hardening when subsequently treated.
Class C, for special purposes.
Class D, axles, shafts, connecting rods and similar forgings.
Class E, for Class D forgings when they are to be heat treated.

Under "carbon and alloy steel forgings," the classification is as follows:

Class A, forgings which may be welded or case hardened.

the American Society for Testing Materials for Steel. (*Continued*)

Nickel	Chromium	Vanadium	Diameter, Over	Diameter, To include	Tensile strength, thousand lb per sq. in.	Yield point, thousand lbs per sq. in.	Elongation in 8 in., per cent	Elongation in 2 in., per cent	Reduction of area, per cent: Inverse ratio	Reduction of area, per cent: Minimum
3.00	.....	.....	..	12	80	50	2,000,000 / T. S.	22	3,600,000 / T. S.	40
3.00	.....	.....	12	20	80	50	1,900,000 / T. S.	21	3,400,000 / T. S.	38
3.00	.....	.....	..	4	100	70	2,200,000 / T. S.	20	4,500,000 / T. S.	41
3.00	.....	.....	4	7	100	65	2,100,000 / T. S.	20	4,300,000 / T. S.	41
3.00	.....	.....	7	10	90	60	2,000,000 / T. S.	20	4,100,000 / T. S.	41
3.00	.....	.....	10	20	85	55	1,900,000 / T. S.	20	3,900,000 / T. S.	41
.....	0.75–1.25	0.15 min.	..	2	95–115	70	.....	20	.....	50
.....	.....	.....	2	4	90–110	65	.....	20	.....	50
1.25 min.	0.70 min.	.....	4	7	90–110	65	.....	20	.....	50
.....	.....	.....	7	10	90–110	65	.....	20	.....	50
.....	0.60–0.90	.....	10	20	85–105	60	.....	20	.....	50
.....	.....	.....	..	..	.....	.....	.....	.....	.....	.....
.....	.....	.....	..	..	80	0.45 T. S.	.....	15	.....	20
.....	.....	.....	..	..	70	0.45 T. S.	.....	18	.....	25
.....	.....	.....	..	..	60	0.45 T. S.	.....	22	.....	30
.....	.....	.....	..	..	70	60	.....	18	.....	35
.....	0.75–1.25	0.15 min.	..	2	105–125	80	.....	18	.....	35
			2	4	100–120	75	.....	18	.....	35
1.25 min.	0.70 min.	.....	4	7	100–120	75	.....	18	.....	35
			7	10	100–120	75	.....	18	.....	45
.....	0.60–0.90	.....	10	20	95–115	70	.....	18	.....	45
.....	0.75–1.25	0.15 min.	..	2	125	105	.....	16	.....	50
			2	4	115	95	.....	16	.....	45
			4	7	110	85	.....	16	.....	45
			7	10	110	75	.....	18	.....	45
2.75 min.	0.70 min.	.....	10	20	110	70	.....	18	.....	45

Class B, mild steel forgings for structural purposes, for minor ship fittings, etc.

Class C, mild steel forgings for structural purposes, for ships, etc.

Classes D to I, various machinery forgings, choice depending on design and upon the stresses and services to be imposed.

Class K, L and M, various machinery forgings, including quenched and tempered axles, shafts and other forgings, choice depending on design, on the stresses and services to be imposed, and on the character of the machining to be done.

Under "steel castings," Class A includes ordinary castings for which no physical requirements are specified; Class B includes castings for which physical requirements are specified.

Samples for Selective Test, see carbon tool steel, except that dimensions of sample bars shall be as follows:

Class 1, ½ by 1 inch by 5 feet long.

Class 2, 1 1/16 inch diameter rod, 2½ feet long.

Treatment of Sample, see carbon tool steel.

Selective Tests. *Class* 1. The sample bar will be cut into five sections, one end of each section treated and ground to the No. 30 form of the Sellers system of lathe tools. Each tool will be tested on a nickel-steel forging of about 100,000 lbs. tensile strength, with a cut 3/16 in. deep, 0.044-in. feed, and a cutting speed of about 65 ft. per min. Each tool will be once reground and again tested. A record will be made of the length of time each tool cuts without a lubricant before it is ruined.

Class 2. Five 7/16 in. diam. 4-tooth facing mills will be made from the sample rod and tested on a piece of ⅝-in. ship's plate without lubricant. Each mill will be run until it is so dull that it breaks either in the teeth or in the shank. The depth of cut will be 0.08 in., the revolutions per minute of the mill will be 370, and the feed of material 20 in. per min. A record will be made of the length of time each mill operates.

U. S. NAVY SPECIFICATIONS FOR CAST IRON (ABSTRACTS)

Foundry Pig Iron (1917)

Uses and Grades. There shall be four grades of pig iron as follows:

(*a*) Grade 1 is suitable for general foundry purposes. It may be used for either heavy or light castings which are to be machined.

(*b*) Grade 2 is suitable for marine engine cylinders, turbine casings, and work of similar character.

(*c*) Grade 3 is suitable for hard, close-grained castings, which are to be machined, where great strength is required. It may also be used with grades 1 and 2 in varying proportions as the work requires.

(*d*) Grade 4 is suitable for use with grades 1, 2, and 3 where castings of great strength or high finish are desired.

Chemical properties are required as follows:

Grade	Carbon, (minimum) per cent	Silicon, per cent	Sulphur, (maximum) per cent	Phosphorus, per cent	Manganese, per cent	Remarks
No. 1.....	3.50	2.75–3.25	0.04	0.50–0.80	0.50–0.90	
No. 2.....	3.25	2.00–2.50	0.05	0.50–0.80	0.50–0.90	
No. 3.....	3.25	1.25–1.75	0.06	0.50–0.90	0.50–0.90	
No. 4.....	3.25	1.50–2.00	0.03	0.30 max.	0.75–1.25	Charcoal iron.

The sample shall be taken as follows: (*a*) One pig shall be taken for every four tons in the lot, chosen from different locations so as to represent as nearly as possible the average quality of the lot. A lot shall always consist of pig from the same heat, no mixing of iron from several heats being permissible.

(*b*) The pigs selected for test shall be broken so as to present a clean fracture over the entire cross-section of the pig. One-half of each pig shall have drillings taken from 3 points triangularly spaced on the clean part of the fracture. Equal proportion of drillings from each pig shall be thoroughly mixed and analysis made from this sample; no resampling will be permitted.

Method of Sampling. Not less than 6 ounces of the sample, taken and mixed as above, is used for analyses. In case the first analysis shows that

the material does not conform to the specifications a check analysis shall be made. The average of these analyses shall be considered final. Analyses shall be made according to the standard method of the American Foundrymen's Association, the gravimetric method being used for determination of sulphur. Other things being equal, preference will be given to machine cast over sand cast iron.

Sow Iron. Not more than 12 per cent of sow iron is acceptable.

Iron Castings (1916)

Physical properties of cast iron are to be in accordance with the following table:

Grades of iron castings	Tensile strength (pounds per square inch)—length of test piece not less than 2 inches	Transverse breaking load (for bar 1 inch square loaded at middle and resting on supports 1 foot apart)	Purposes for which intended
1	20,000 (min.)	2,200 (min.) 2,800 (max.)	Steam cylinder and valve-chest castings. Steam turbine casings, steam turbine parts. Gas-engine cylinder and valve-chest casings. Internal-combustion engine cylinders and valve-chest casings.
2	20,000 (min.)	2,500 (min.)	Cylinder liners and valve-chest liners. Steam, gas, and internal combustion engines. Cylinder and valve-chest liners, small gas engines, and internal-combustion cylinders when cast in one piece.
3	20,000 (min.)	2,200 (min.)	Other important parts, such as main and auxiliary engine parts, etc.
4	To be inspected to see if they are in all respects suitable for the purposes for which they are intended.		Minor parts, such as furnace fittings, etc.

Hardness Requirement. Great care must be taken to determine that the machinery specifications for *hardness* of cylinders, liners, and valve-chest liners are complied with, and a test piece from the casting should be machined in order to show the degree of hardness.

Quality of Material. The castings must be of uniform grain, smooth, free from blow-holes, porous places, shrinkage, and other cracks or defects, and must be well cleaned.

Number of Tests. Sound test pieces shall be taken in sufficient number to exhibit the character of the metal in the entire piece from all castings requiring physical test.

Finish. Scale must be removed from unfinished parts of the insides of all cylinders, cylinder covers, valve-chest covers and cylinder and valve-chest liners, and from ports and passages of cylinders and valve-chests, either by pickling or other approved process as may be required.

All engine castings must be finished to blue-print size.

Malleable Iron Castings (1917)

Workmanship and Finish. The castings shall be free from blemishes, scale and shrinkage cracks and shall substantially conform to the sizes and shapes of the patterns. A variation of $\frac{3}{32}$ in. per ft. is permitted.

Process. The castings shall be made from iron melted in either an air furnace, open-hearth furnace or electric furnace.

Chemical and Physical Requirements. Limiting percentage of sulphur is 0.10 (max.) and of phosphorous is 0.23 (max.); minimum tensile strength, 38,000 lbs. per sq. in.; elongation in 2 in., 5 per cent. The tensile test specimen is shown in Fig. 15.

The transverse test specimen, hereafter described, when tested with the cope side up on supports 12 in. apart, with pressure applied at the center, shall conform to the following requirements as to transverse properties:

Thickness of specimen, in.	½	900	1.25
Minimum load applied at center, lbs.	⅝	1400	1.00
Minimum deflection at center, lbs.	¾	2000	0.75

Transverse test specimens shall be 14 in. in length by 1 in. in width, and either ½ in., ⅝ in., or ¾ in., in thickness. The thickness of the specimen selected shall be in proportion to the thickness of the casting which it represents.

Two tension and two transverse test specimens shall be cast in each mold with risers of sufficient height at each end to secure sound bars. All specimens shall be cast without chills and with ends perfectly free in the mold.

Four molds shall be poured to represent each melt. When the entire melt is used for castings which are subject to these specifications two molds shall be poured within 5 minutes after tapping into the first ladle, and two molds from the last iron of the melt. When only part of the melt is required for such castings two molds shall be poured from the first ladle of iron used and two molds after the required iron has been tapped.

The test specimens from one mold from the first and one mold from the last of the melt shall be annealed in the hottest part of the annealing oven and the remaining specimens shall be annealed in the coldest part.

A. S. T. M. SPECIFICATIONS FOR CAST IRON (ABSTRACTS)

Malleable Iron Castings (1915)

Physical and Chemical Requirements. The castings shall be made from iron melted in either an air furnace, open hearth furnace or electric furnace.

Tension test specimens, Fig. 14, shall conform to the following minimum requirements as to tensile properties:

Tensile strength, lbs. per sq. in.	38,000
Elongation in 2 in., per cent	5

Transverse test specimens, tested with the cope side up on supports 12 in. apart, pressure being applied at the center, shall conform to the following requirements as to transverse properties:

Thickness of specimen, in.	Minimum load applied at center, lbs.	Minimum deflection at center, in.
½	900	1.25
⅝	1400	1.00
¾	2000	0.75

In addition to the tension and transverse tests, the inspector representing the purchaser may satisfy himself of the suitability of the iron used for the castings by breaking a reasonable number of castings before annealing to examine for excessive mottling or graphite spots. In the case of castings of special design or importance, he may also require test lugs of a size proportional to the thickness of the casting, but not exceeding ½ by ¾ in. in section. At least one of these lugs shall be left on the casting for final inspection.

FIG. 14.

Test Specimens. (*a*) Tension test specimens shall be of the form and dimensions shown in Fig. 15. Transverse test specimens shall be 14 in. in length by 1 in. in width and either ½, ⅝ or ¾ in. in thickness. The thickness of the specimen selected shall be in proportion to the thickness of the casting which it represents.

(*b*) Two tension and two transverse test specimens shall be cast in each mold with risers of sufficient height at each end to secure sound bars. All specimens shall be cast without chills, and with ends perfectly free in the mold.

(*c*) Four molds shall be poured to represent each melt. When the entire melt is used for castings which are subject to these specifications, two molds shall be poured within five minutes after tapping into the first ladle, and two molds from the last iron of the melt. When only part of the melt is required for such castings, two molds shall be poured from the first ladle of iron used and two molds after the required iron has been tapped.

(*d*) The molds shall be suitably stamped to identify the specimens.

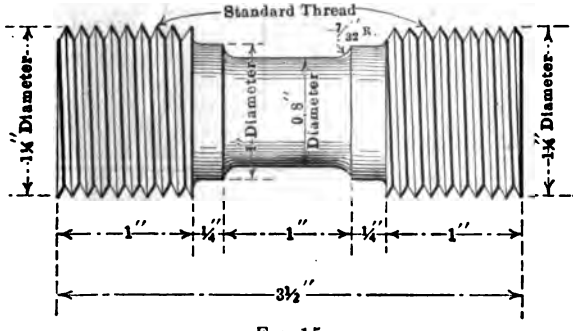


FIG. 15.

The test specimens from one mold from the first and one mold from the last of the melt shall be annealed in the hottest part of the annealing oven, and the remaining specimens shall be annealed in the coldest part.

Tests. One tension and one transverse test specimen from each of the four molds representing a melt shall be selected for test. The remaining specimens shall be reserved, and shall be tested in case of failure to conform to the requirements specified.

If more than one tension or transverse test specimen from each of the two molds annealed in the two points in the oven specified above fails to meet the requirements as to tensile or transverse properties, the castings from that melt will be rejected.

Workmanship. The castings shall substantially conform to the sizes and shapes of the patterns, and shall be made in a workmanlike manner. A variation of $\frac{3}{32}$ in. per ft. will be permitted.

The castings shall be free from blemishes, scale and shrinkage cracks.

Gray-iron Castings (1905)

Unless furnace iron is specified, all gray castings are understood to be made by the cupola process.

Chemical Properties. The sulphur contents to be as follows:

Light castings	Not over 0.08 per cent
Medium castings	Not over 0.10 per cent
Heavy casting	Not over 0.12 per cent

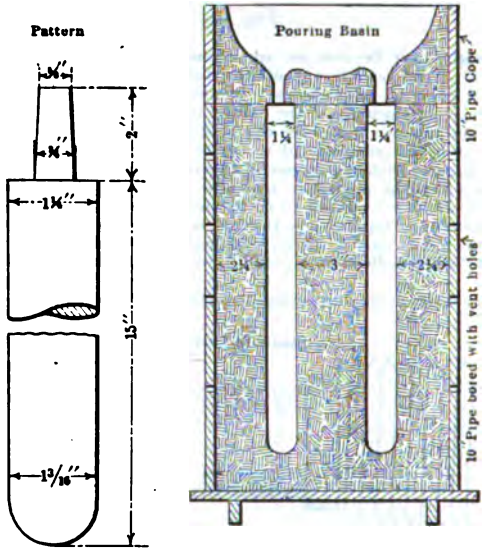


FIG. 16.

Classification. In dividing castings into light, medium and heavy classes, the following stands have been adopted:

Castings having any section less than ½-in. thick shall be known as *light castings*.

Castings in which no section is less than 2 in. thick shall be known as *heavy castings*.

Medium castings are those not included in the above classification.

Physical Properties. *Transverse Test:* The minimum breaking strength of the "Arbitration Bar" under transverse load shall be not under 2,500 lbs. for light; 2,900 lbs. for medium and 3,300 lbs. for heavy castings. In no case shall the deflection be under 0.10 inch. *Tensile Test:* Where specified, this shall not run less than 18,000 lbs. per sq. in. for light; 21,000 lbs. for medium; and 24,000 lbs. for heavy castings.

Arbitration Bar. The quality of the iron going into castings under specification shall be determined by means of the "Arbitration Bar." This is a bar 1¼ in. in diameter and 15 in. long. It shall be prepared as stated further on and tested transversely. The tensile test is not recommended, but in case it is called for, the bar as

shown in Fig. 15, and turned up from any of the broken pieces of the transverse test shall be used. The expense of the tensile test shall fall on the purchaser.

Number of Test Bars. Two sets of two bars shall be cast from each heat, one set from the first and the other set from the last iron going into the castings. Where the heat exceeds twenty tons, an additional set of two bars shall be cast for each twenty tons or fraction thereof above this amount. In case of a change of mixture during the heat, one set of two bars shall also be cast for every mixture other than the regular one. Each set of two bars is to go into a single mold. The bars shall not be rumbled or otherwise treated, being simply brushed off before testing.

Method of Testing. The transverse test shall be made on all the bars cast, with supports 12 in. apart, load applied at the middle, and the deflection at rupture noted. One bar of every two of each set made must fulfill the requirements to permit acceptance of the castings represented.

Mold for Test Bar. The mold for the bars is shown in Fig. 16. The bottom of the bar is $\frac{1}{16}$ in. smaller in diameter than the top, to allow for draft and for the strain of pouring. The pattern shall not be rapped before withdrawing. The flask is to be rammed up with green molding sand, a little damper than usual, well mixed and put through a No. 8 sieve, with a mixture of one to twelve bituminous facing. The mold shall be rammed evenly and fairly hard, thoroughly dried and not cast until it is cold. The test bar shall not be removed from the mold until cold enough to be handled.

Speed of Testing. The rate of application of the load shall be from 20 to 40 seconds for a deflection of 0.10 in.

Samples for Analysis. Borings from the broken pieces of the "Arbitration Bar" shall be used for the sulphur determinations. One determination for each mold made shall be required. In case of dispute, the standards of the American Foundrymen's Association shall be used for comparison.

Finish. Castings shall be true to pattern, free from cracks, flaws and excessive shrinkage. In other respects they shall conform to whatever points may be specially agreed upon.

PROTECTIVE COATINGS FOR IRON AND STEEL

Protective coatings are of three types: **Metallic coatings**; coatings formed by **oxidation or other chemical change of the surface** of the iron; **organic coatings** such as paints, varnishes, enamels etc.

Metallic Coatings. These may include any of the common metals and alloys that can be readily applied to steel. A list of these metals, together with iron, arranged in the order of their electric potentials is as follows: Aluminum, zinc, iron, cobalt, nickel, tin, lead, copper, silver. The relative solution tensions of these metals are such that any metal can displace a metal following it in the list from solutions of its salts. Consequently, when these two metals are in contact, the first metal protects the second from corrosion.

Of the metals in the above list, therefore, only two protect iron by their chemical action. Of these two, zinc is the more generally useful on account of its low cost and ease of application. Since zinc protects iron by its electrochemical action, a hole or flaw in the coating does not lead to corrosion of the iron unless the uncovered area is so large that the protective influence of the zinc does not extend to all parts.

The oldest and most generally useful method of applying the zinc coating is **hot dipping.** In this process, the steel after a preliminary cleaning of the surface is immersed in molten zinc.

Another process, called **"sherardizing,"** consists in subjecting the iron to an atmosphere of zinc vapor. The iron is packed in a revolving drum with zinc dust containing a small amount of zinc oxide; it is then heated at a temperature of 550 to 375 deg. C. for 3½ to 4 hours.

Zinc is also applied by **electroplating.** The plating liquid is either a zinc sulphate solution containing a little free acid, or a solution of the cyanide or oxide in a mixture of sodium cyanide and sodium hydroxide. Plating usually takes from ½ to 1 hour unless very heavy deposits are required.

In the **Schoop process,** zinc or other metal in the form of wire is fed into a spray gun where it is melted in an oxyacetylene flame and projected by a strong air current against the object to be plated.

In another process, called **"epicassit,"** the zinc in the form of filings is mixed with flux, made into a paste and painted on the steel; the steel is then heated until the zinc melts and forms a continuous coating.

Of these methods each has its advantages and disadvantages. Since zinc applied by the hot dipping method collects in depressions and re-entrant angles, this method is not as suitable as the sherardizing method for protecting accurately machined surfaces such as screw threads. Although good protection may be obtained by use of the sheradizing process, this process gives a less uniform product than does the hot dipping process; the lack of uniformity is due to the fact that the quality of the product is much influenced by variations in the purity of the zinc dust, in the uniformity of heating, and in other conditions. Neither the hot dipping nor the sherardizing methods are suitable for use on hardened steel or any other metal that would be unfavorably affected by the temperatures prevailing in these processes. For such material, the electroplating or the Schoop processes are suitable. Another advantage of the electroplating process is the ease with which the thickness of the coating can be controlled. The thickness on sharp projections, however, is greater than on depressions even on flat plates, the coating is somewhat heavier at the edges than near the center.

Aluminum, since it costs more and is less easily applied, it less useful than zinc as a protective coating. One method of application, called **"calorizing,"** is similar to sherardizing. The steel to be calorized is packed in a mixture of aluminum, aluminum oxide, and ammonium chloride; it is then heated to about 900 deg. C. for 2 of 3 hours. The absorbed aluminum vapor forms an alloy with the iron.

Though the calorizing process does not prevent corrosion at ordinary temperatures, it increases the resistance of steel to oxidation at high temperatures. It is claimed that this process protects steel indefinitely from oxidation at temperatures up to 1050 deg. C. and greatly increases the life of steel objects at still higher temperatures. The protection is due to a film of aluminum oxide on the surface of the calorized object.

Protective metals other than zinc and aluminum may be divided into two groups. In the one group are **metals of low melting point,** such as tin, lead, and their alloys; in the other group are the **metals of high melting point,** such as copper, nickel, cobalt, silver, and their alloys. Metals of the first group are usually applied by hot dipping; metals of the second group and some of the metals of the first group are applied by electroplating.

Tin as a metallic coating is widely used. Unless the coating is free from pin holes or abrasions, the iron in these regions will be corroded more rapidly than if no tin coating were present. Since, however, tin corrodes only slowly, the coating, when intact, affords efficient protection.

Since **lead** is much cheaper than tin, it is frequently used in coatings as and alloy with tin. Such an alloy, known as **terne,** contains about ¾ lead and ¼ tin. Recently lead alone applied by electroplating, has been much used.

Though **copper** is usually deposited by electroplating, it may also be

applied by the Schoop and epicassit processes. **Copper clad steel** is produced by casting the copper around a steel billet and rolling this down into rods or sheets. The coating thus formed is very efficient.

Nickel is usually applied by electroplating. **Nickel clad steel** is also in use; it is made by a process similar to that used in manufacture of copper clad steel.

Cobalt coatings are bluish white in color. They have the same uses as nickel coatings. Cobalt is applied by electroplating. Since solutions for cobalt plating can be made much more concentrated than those for nickel plating, it is possible to use a much greater current density in the deposition of cobalt than in the deposition of nickel. For this reason, cobalt can be deposited from its solutions from 4 to 15 times as fast as nickel. On account of its much greater hardness, cobalt coating is equal in protective power to nickel coating 4 times as thick. Although the cost of cobalt is greater than that of nickel, the higher cost is offset by the reduced quantity required and the greater speed of deposition.

Brass coatings are deposited by electroplating from a solution containing both copper and zinc. Aside from its color, which is considered desirable for small ornamental fixtures, brass has no advantages over either copper or zinc as a coating. **Brass and bronze** have also been applied by a process similar to the copper clad process.

Microstructure of metallic coatings. Metals applied by hot dipping usually form alloys with the iron. The coating thus formed, therefore, consists of one or more layers of alloy adjacent to the iron, also an exterior layer of the protective metal alone. Metals applied by sherardizing and similar processes also form alloys with the iron. The coating, in this case, contains one or more layers of iron alloy and an exterior layer of protective metal, which is thinner than on hot dipped objects. The iron alloys appear to be definite compounds. Since some of these compounds have electric potentials that are not intermediate between the potentials of their constituent elements, the protective qualities of the coating may vary considerably with the proportion of alloy in the coating. Coatings deposited by electroplating contain no layers of iron alloy. The coatings consist of crystals elongated in a direction perpendicular to the surface of the iron. Coatings deposited by the Schoop process also show no layer of iron alloy; they are chiefly mechanical conglomerates of the sprayed drops of metal.

Methods of Testing Metallic Coatings. The usual methods of testing zinc coatings are the Preece test, the basic lead acetate test, the hydrochloric acid-antimony chloride test, and the salt spray test.

In the **Preece test**, the carefully cleaned sample is dipped in a solution of copper sulphate, held in the solution for one minute, washed, and lightly rubbed to remove deposited copper. The solution is made by dissolving 36 parts commercial copper sulphate in 100 parts water, adding some cupric oxide to neutralise any free acid, then diluting with water until the specific gravity is 1.186 at 18 deg. C., at about which temperature the solution should be used.

As long as zinc remains on the surface, the copper deposited on it is loosely adherent; as soon as the zinc coating has been removed from any part of the surface, however, there appears at that point a deposit of bright, adherent copper. The number of dips necessary so reach this end point are an index of the effectiveness of the coating. The Preece method, therefore, is used to determine the effectiveness of a coating at its thinnest point. Since the Iron-zinc alloys are not readily removed by this method, the Preece test gives such variable results; especially in testing sherardized objects, that the A.S.T.M. Committee on Corrosion has recommended its abandonment.

The basic lead acetate test is used to determine the weight of zinc coating per unit of surface area. The solution is made by dissolving 400 gm. crystallised lead acetate in one liter of water, adding 4 gm. powdered litharge, shaking and filtering; the solution is then diluted until its specific gravity is 1.275 at 15.5 deg. C. In this solution, the weighed specimens are given successive immersions of about three minutes each, followed each time by brushing, until a bright iron surface is exposed. The specimens are then washed, dried, and reweighed. Since this method removes not only the pure zinc but also the zinc-iron alloys, it may be used to determine the total weight of coating on sherardised as well as on hot dipped objects.

The hydrochloric acid-antimony chloride test like the basic, lead acetate method, is used to determine the total weight of zinc coating per unit of surface area. The weighed specimens, whose total surface area should be not less than about four square inches, are dipped in a solution containing 100 c.c. concentrated hydrochloric acid to which has been added 5 c.c. of a solution of antimony trioxide in one liter of concentrated hydrochloric acid. The same solution may be used five times, if before each immersion 5 c.c. additional antimony chloride solution are added. The samples, after immersion for one minute, are brushed, rinsed in running water, and reweighed if the removal of the coating has been complete.

Immersion for one minute for hot dipped objects and four minutes for sherardised objects is usually sufficient. The antimony deposited on the iron protects it from solution, while not interfering with the solution of the zinc. The loss in weight, therefore, equals the weight of the zinc coating.

The **salt spray method**, which has been growing in favor recently, tests the endurance of a coating under accelerated service conditions. In this method, the specimens are exposed to a fog of fine particles of a saturated salt solution. The fog is produced by spraying the salt solution through an ordinary atomiser by means of an air blast; the specimens are so supported that the spray from the atomiser does not strike them directly. The removal of the coating is usually indicated by the appearance of iron rust on the regions where the coating was least effective. Pin holes may thus be detected in from three to ten hours. If the sample shows failure of the coating in less than 24 hours, it should be regarded as unsatisfactory. Endurance for two or three days indicates a coating that would endure under moderate service conditions, while endurance for four to six days indicates a coating that would endure under severe service conditions.

Flaws in coatings of lead, tin, copper etc. may be detected by various color reactions. One of the best reagents for this purpose is a one per cent solution of **sodium ferri-cyanide** in two per cent sulphuric acid. With this reagent, the appearance of a blue percipitate on any part of the coating indicates exposed iron.

Protection of iron by means of oxide of iron usually depends on the formation of the magnetic oxide, Fe_3O_4. This oxide, unlike the red oxide, Fe_2O_3, inhibits instead of accelerating corrosion. Repeated heating alternately in oxidising and reducing gases gives a comparatively heavy coating of this oxide. The etching and coloring processes give thinner coatings, usually of a lower order of resistance. The oxide coatings undoubtedly owe some of their rust resistance to the fact that they are usually kept oiled.

Bower-Barff Method. In this process, the steel, is first heated at about 350 deg. C. in air alone or in air and superheated steam. When a coating of ferric oxide has thus been formed, gaseous hydrocarbons are introduced to reduce the red oxide to the black oxide. A recent modification of this process consists in heating at a low red heat in a mixture of steam and bensine. The **Swann, Bontempi, Gesner,** and **Weigelen processes** are other modifications of the Bower-Barff process.

The black oxide coating may also be produced by heating the steel in contact with various organic mixtures, such as burnt bone and oil, burnt bone and charcoal, oil and sawdust, etc. Or the steel may be oiled first and then heated to about 350 deg. C.

In the **etching processes**, the steel after treatment with an etching solution is allowed to dry slowly in the air. The outer coating of red oxide thus formed is then scraped off, leaving a thin, adherent coating of magnetic oxide whose color depends on the etching solution employed. The following list from Bureau of Standards Circular No. 80 gives some typical solutions and the colors thus produced.

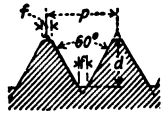
Color	Reagent	Parts by weight.
Black		
	First formula	
	Bismuth chloride	20
	Mercuric chloride	40
	Copper chloride	20
	Hydrochloric acid	120
	Alcohol	100
	Water	1000
	Second formula	
	Copper nitrate, 10 % solution	700
	Alcohol	300
	Third formula	
	Mercuric chloride	50
	Ammonium chloride	50
	Water	1000
Brown		
	First formula	
	Alcohol	45
	Iron chloride solution	45
	Mercuic chloride	45
	Sweet spirits of niter	45
	Copper sulphate	30
	Nitric acid	22
	Water	1000
Brown		
	Second formula	
	Nitric acid	70
	Alcohol	140
	Copper sulphate	200
	Iron filings	10
	Water	1000
Blue		
	Iron chloride	400
	Antimony chloride	400
	Gallic acid	200
	Water	1000
Bronze		
	Manganese nitrate, 10 per cent solution	700
	Alcohol	300

Similar protective coatings are produced by the action of various **other reagents**. Protective oxides are produced by immersion of the steel in boiling alkaline solutions such as **sodium hydroxide** solution containing oxidising agents such as **sodium picrate, sodium peroxide etc.** Similar oxides are produced by the action of fused salts such as **potassium nitrate, sodium nitrate,** or a mixture of the two, sometimes containing **manganese dioxide.** A protective oxide is also formed by dipping steel in a 10 per cent **potassium bichromate solution** and then heating it in a smoky flame. The coatings exhibiting **temper colors,** produced by heating steel to about 220 to 320 deg. C., give some protection against corrosion; of these coatings, the blue oxide is considered the most effective.

The **Parker or Coslett process** is of wide commercial application in this process, the iron is immersed in hot dilute phosphoric acid containing ferrous and ferric phosphates or other metallic salts. The coating, when dried, is grayish white in color, but becomes black when oiled. Its resistance to corrosion is about the same in degree as that of the light oxide coatings.

Table 1. U. S. (Sellers) Standard Screw Threads

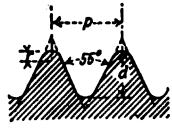
(For Pipe Threads see p. 1271)



p = pitch = 1 ÷ (No. of threads per in.)
d = depth = 0.6495p
f = flat = p ÷ 8

Diam. of screw, in.	Threads per in.	Diam. at root of thread, in.	Width of flat, in.	Area of section at root of thread, sq. in.	Diam. of screw, in.	Threads per in.	Diam. at root of thread, in.	Width of flat, in.	Area of section at root of thread, sq. in.
¼	20	0.1850	0.0063	0.027	2	4½	1.7113	0.0278	2.302
5/16	18	0.2403	0.0069	0.045	2¼	4½	1.9613	0.0278	3.023
⅜	16	0.2936	0.0078	0.068	2½	4	2.1752	0.0313	3.719
7/16	14	0.3447	0.0089	0.093	2¾	4	2.4252	0.0313	4.620
½	13	0.4001	0.0096	0.126	3	3½	2.6288	0.0357	5.428
9/16	12	0.4542	0.0104	0.162	3¼	3½	2.8788	0.0357	6.510
⅝	11	0.5069	0.0114	0.202	3½	3¼	3.1003	0.0385	7.548
¾	10	0.6201	0.0125	0.302	3¾	3	3.3170	0.0417	8.641
⅞	9	0.7307	0.0139	0.420	4	3	3.5670	0.0417	9.963
1	8	0.8376	0.0156	0.550	4¼	2⅞	3.7982	0.0435	11.329
1⅛	7	0.9394	0.0179	0.694	4½	2¾	4.0276	0.0455	12.753
1¼	7	1.0644	0.0179	0.893	4¾	2¾	4 2551	0.0476	14.226
1⅜	6	1.1585	0.0208	1.057	5	2½	4.4804	0.0500	15.763
1½	6	1.2835	0.0208	1.295	5¼	2½	4.7304	0.0500	17.572
1⅝	5½	1.3888	0.0227	1.515	5½	2⅜	4.9530	0.0526	19.267
1¾	5	1.4902	0.0250	1.746	5¾	2¾	5.2030	0.0526	21.262
1⅞	5	1.6152	0.0250	2.051	6	2¼	5.4226	0.0556	23.098

Table 2. Whitworth Standard Screw Threads



d = 0.6403p
r = radius = 0.1373p

Diam. of screw, in.	Threads per in.	Depth of thread, in.	Diam. at root of thread, in.	Diam. of screw, in.	Threads per in.	Depth of thread, in.	Diam. at root of thread, in.
¼	20	0.0320	0.1860	1⅝	5	0.1281	1.3689
5/16	18	0.0356	0.2414	1¾	5	0.1281	1.4939
⅜	16	0.0400	0.2950	2	4½	0.1423	1.7154
7/16	14	0.0457	0.3460	2¼	4	0.1601	1.9298
½	12	0.0534	0.3933	2½	4	0.1601	2.1798
9/16	12	0.0534	0.4558	2¾	3½	0.1830	2.3841
⅝	11	0.0582	0.5086	3	3½	0.1830	2.6341
11/16	11	0.0582	0.5711	3¼	3¼	0.1970	2.8560
¾	10	0.0640	0.6219	3½	3¼	0.1970	3.1060
13/16	10	0.0640	0.6844	3¾	3	0.2134	3.3231
⅞	9	0.0711	0.7327	4	3	0.2134	3.5731
1	8	0.0900	0.8399	4½	2⅞	0.2227	4.0546
1⅛	7	0.0915	0.9420	5	2¾	0.2328	4.5343
1¼	7	0.0915	1.0670	5½	2⅝	0.2439	5.0121
1⅜	6	0.1067	1.1616	6	2½	0.2561	5.4877
1½	6	0.1067	1.2866				

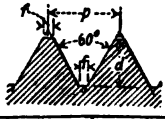
Table 3. British Association Screw Threads

Number	Diam. of screw, mm.	Approx. diam., in.	Pitch, mm.	Approx. pitch, in.	Diam. at root of thread, mm.	Number	Diam. of screw, mm.	Approx. diam., in.	Pitch, mm.	Approx. pitch, in.	Diam. at root of thread, mm.
0	6.0	0.236	1.00	0.0394	4.8	13	1.20	0.047	0.25	0.0098	0.90
1	5.3	0.209	0.90	0.0354	4.22	14	1.00	0.039	0.23	0.0091	0.72
2	4.7	0.185	0.81	0.0319	3.73	15	0.90	0.035	0.21	0.0083	0.65
3	4.1	0.161	0.73	0.0287	3.22	16	0.79	0.031	0.19	0.0075	0.56
4	3.6	0.142	0.66	0.0260	2.81	17	0.70	0.028	0.17	0.0067	0.50
5	3.2	0.126	0.59	0.0232	2.49	18	0.62	0.024	0.15	0.0059	0.44
6	2.8	0.110	0.53	0.0209	2.16	19	0.54	0.021	0.14	0.0055	0.37
7	2.5	0.098	0.48	0.0189	1.92	20	0.48	0.019	0.12	0.0047	0.34
8	2.2	0.087	0.43	0.0169	1.68	21	0.42	0.017	0.11	0.0043	0.29
9	1.9	0.075	0.39	0.0154	1.43	22	0.37	0.015	0.10	0.0039	0.25
10	1.7	0.067	0.35	0.0138	1.28	23	0.33	0.013	0.09	0.0035	0.22
11	1.5	0.059	0.31	0.0122	1.13	24	0.29	0.011	0.08	0.0031	0.19
12	1.3	0.051	0.28	0.0110	0.96	25	0.25	0.010	0.07	0.0028	0.17

A. S. M. E. Standard Screw Threads follow the U. S. Standard form except that the proportions are as follows:

$d = 0.7037\ p$; f (top) $= p + 8$; f (bottom) $= p + 16$

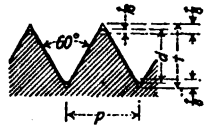
Table 4. French (Metric) Standard Screw Threads



p = pitch in mm.
$d = 0.6495p$
$f = p \div 8$

Diam. of screw, mm.	Pitch, mm.	Diam. at root of thread, mm.	Width of flat, mm.	Diam of screw, mm.	Pitch, mm.	Diam. at root of thread, mm.	Width of flat, mm.	Diam. of screw, mm.	Pitch, mm.	Diam. at root of thread, mm.	Width of flat, mm.
3	0.5	2.35	0.06		18	2.5	14.75	0.31	40	4.0	34.80	0.50
4	0.75	3.03	0.09	20	2.5	16.75	0.31	42	4.5	36.15	0.56
5	0.75	4.03	0.09	22	2.5	18.75	0.31	44	4.5	38.15	0.56
6	1.0	4.70	0.13	22	3.0	18.10	0.38	45	4.5	39.15	0.56
7	1.0	5.70	0.13	24	3.0	20.10	0.38	46	4.5	40.15	0.56
8	1.0	6.70	0.13	26	3.0	22.10	0.38	48	5.0	41.51	0.63
8	1.25	6.38	0.16	27	3.0	23.10	0.38	50	5.0	43.51	0.63
9	1.0	7.70	0.13	28	3.0	24.10	0.38	52	5.0	45.51	0.63
9	1.25	7.38	0.16	30	3.5	25.45	0.44	56	5.5	48.86	0.69
10	1.5	8.05	0.19	32	3.5	27.45	0.44	60	5.5	52.86	0.69
11	1.5	9.05	0.19	33	3.5	28.45	0.44	64	6.0	56.21	0.75
12	1.5	10.05	0.19	34	3.5	29.45	0.44	68	6.0	60.21	0.75
12	1.75	9.73	0.22	36	4.0	30.80	0.5	72	6.5	63.56	0.81
14	2.0	11.40	0.25	38	4.0	32.80	0.5	76	6.5	67.56	0.81
16	2.0	13.40	0.25	39	4.0	33.80	0.5	80	7.0	70.91	0.88

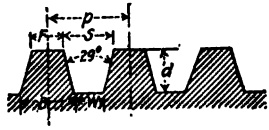
Table 5. International Standard Metric Screw Threads



$d = 0.7036p$
$t = 0.866p$

Diam. of screw, mm.	Pitch, mm.	Diam. of screw, mm.	Pitch, mm.	Diam. of screw, mm.	Pitch, mm.	Diam. of screw, mm.	Pitch, mm.	Diam. of screw, mm.	Pitch, mm.	Diam. of screw, mm.	Pitch, mm.
6	1.00	12	1.75	24	3.00	42	4.50	64	6.00	96	8.00
7	1.00	14	2.00	27	3.00	45	4.50	68	6.00	116	9.00
8	1.25	16	2.00	30	3.50	48	5.00	72	6.50	136	10.00
9	1.25	18	2.50	33	3.50	52	5.00	76	6.50	.....	.....
10	1.50	20	2.50	36	4.00	56	5.50	80	7.00	.....	.....
11	1.50	22	2.50	39	4.00	60	5.50	88	7.50	.....	.....

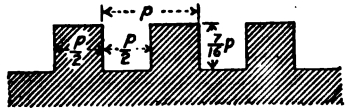
Table 6. Acme 29-deg. Screw Threads



$d = 0.5p + 0.01$ in.
$F = 0.3707p$
$W = 0.3707p - 0.0052$ in.

N	p	d	F	W	S	B
Number of threads per in.	Pitch of single thread, in.	Depth of thread, in.	Width of top of thread, in.	Width of space at bottom of thread, in.	Width of space at top of thread, in.	Thickness at root of thread, in.
1	1.0	0.5100	0.3707	0.3655	0.6293	0.6345
1½	0.750	0.3850	0.2780	0.2728	0.4720	0.4772
2	0.500	0.2600	0.1853	0.1801	0.3147	0.3199
3	0.3333	0.1767	0.1235	0.1183	0.2098	0.2150
4	0.250	0.1350	0 0927	0.0875	0.1573	0.1625
5	0.200	0.1100	0.0741	0.0689	0.1259	0.1311
6	0.1667	0.0933	0.0618	0.0566	0.1049	0.1101
7	0.1428	0.0814	0.0530	0.0478	0.0899	0 0951
8	0.125	0.0725	0.0463	0.0411	0.0787	0.0839
9	0.1111	0.0655	0.0413	0.0361	0.0699	0.0751
10	0.10	0.0600	0.0371	0.0319	0.0629	0.0681

Table 7. Sellers Standard Square Threads



Diam. of bolt, in.	Threads per in.	Diam. of root of thread, in.	Diam. of bolt, in.	Threads, per in.	Diam. at root of thread, in.	Diam. of bolt, in.	Threads, per in.	Diam. at root of thread, in.	Diam. of bolt. in.	Threads per in.	Diam. at root of thread, in.
¼	10	0.1625	⅜	5	0.575	1½	3	1.2084	3	1½	2.5
9/16	9	0.2153	1 3/16	4½	0.6181	1⅝	2¾	1.307	3¼	1¾	2.75
⅜	8	0.2658	⅞	4½	0.6806	1¾	2½	1.4	3½	1⅝	2.962
7/16	7	0.3125	1 5/16	4	0.7188	1⅞	2½	1.525	3¾	1½	3.168
½	6½	0.3656	1	4	0.7813	2	2¼	1.612	4	1⅝	3.418
9/16	6	0.4167	1⅛	3½	0.875	2¼	2¼	1.862	.....	.....	.....
⅝	5½	0.4666	1¼	3½	1.00	2½	2	2.0626	.....	.....	.....
11/16	5	0.512	1⅜	3	1.0834	2¾	2	2.3126	.....	.....	.....

Table 8. U. S. Standard Bolts and Nuts

Diam. of bolt, in.	Rough dimensions				Finished dimensions of heads and nuts				
	Heads and nuts				Hexagon heads and nuts				
	Width across flats, hex. and square, in.	Width across corners*: Hexagon, in.	Width across corners*: Square, in.	Thickness of head,† in.	Width across flats, in.	Width across corners, in.	Thickness, in.	Exact size of hole in nut, in.	Size of tap drill used, in.
¼	7/16	27/64	23/32	1/4	7/16	1/2	3/16	0.1850	0.191
5/16	17/32	1 5/16	2 7/32	19/64	17/32	39/64	1/4	0.2408	0.246
3/8	1 5/16	51/64	1 1/32	1 1/32	5/8	23/32	5/16	0.2938	19/64
7/16	21/32	29/32	1 7/64	25/64	23/32	53/64	3/8	0.3447	23/64
1/2	7/8	1 13/64	1 1/4	7/16	13/16	15/16	7/16	0.4001	15/32
9/16	31/32	1 7/16	1 3/8	31/64	29/32	1 3/64	1/2	0.4542	17/32
5/8	1 1/16	1 15/64	1 1/2	17/32	1	1 5/32	9/16	0.5069	33/64
3/4	1 1/4	1 49/64	1 25/32	5/8	1 3/16	1 21/64	11/16	0.6201	5/8
7/8	1 7/16	1 63/64	2 1/32	23/32	1 3/8	1 19/32	13/16	0.7307	47/64
1	1 5/8	1 7/8	2 7/64	13/16	1 9/16	1 13/16	15/16	0.8376	27/32
1 1/8	1 13/16	2 3/32	2 5/16	29/32	1 3/4	2 1/64	1 1/16	0.9394	61/64
1 1/4	2	2 5/16	2 23/64	1	1 15/16	2 5/64	1 3/16	1.0644	1 5/64
1 3/8	2 3/16	2 17/32	3 1/32	1 3/32	2 1/8	2 29/64	1 5/16	1.1585	1 11/64
1 1/2	2 3/8	2 3/4	3 3/64	1 5/16	2 5/16	2 43/64	1 7/16	1.2835	1 19/64
1 3/4	2 3/4	3 5/16	3 57/64	1 3/8	2 11/16	3 7/64	1 11/16	1.4902	1 33/64
2	3 1/16	3 39/64	4 25/64	1 9/16	3 1/16	3 17/32	1 15/16	1.7113	1 23/32
2 1/4	3 3/8	4 1/16	4 63/64	1 3/4	3 7/16	3 31/32	2 3/16	1.9613	1 31/32
2 1/2	3 7/8	4 21/64	5 31/64	1 15/16	3 13/16	4 13/32	2 7/16	2.1752	2 7/16
2 3/4	4 1/4	4 29/32	6 1/64	2 1/8	4 1/16	4 27/32	2 11/16	2.4252	2 7/16
3	4 5/8	5 11/32	6 17/32	2 5/16	4 9/16	5 3/32	2 15/16	2.6288	2 43/64

* Width across corners of hexagon heads or nuts = 1.155 × width across flats.
Width across corners of square heads or nuts = 1.414 × width across flats.
† Thickness of nut = diam. of bolt.

Table 9. Weight of 100 Bolts and Nuts, Lbs.

(For Bolt Material Specifications see p. 127)

Length under head, inches	Bolts with square heads and nuts									Bolts with hexagon heads and nuts				
	Diameter of bolt, inches									Diameter of bolt, inches				
	¼	5/16	⅜	7/16	½	⅝	¾	⅞	1	½	⅝	¾	⅞	1
1	4	7	11	15	22	37	56	84	122	19	33	52	76	110
1¼	4	7	11	16	23	39	59	88	128	20	34	54	80	116
1½	5	8	12	17	24	41	62	93	135	22	36	57	85	121
1¾	5	8	13	18	26	43	64	97	139	23	38	60	89	127
2	5	9	14	19	27	45	67	101	144	24	40	63	93	132
Lb. per inch additional	1.4	2.2	3.1	4.3	5.6	8.7	12.5	17.0	22.3	5.6	8.7	12.5	17.0	22.3

Table 11. Safe Loads for U. S. Standard Bolts

Nominal diam., in.	No. of threads per in.	Ultimate strength, lbs. per sq. in.						
		20,000	40,000	50,000	60,000	65,000	80,000	95,000
		Alloy Cu, 88%, Sn, 10%, Zn, 2%	Phosphor-bronze	Wrought iron and best rolled bronze	Class B bolt material	Class A bolt material	Class A Nos. 1 and 2 machinery forgings	High-grade machinery forgings
¼	20	57	115	143	172	186	229	272
5/16	18	99	198	247	297	322	396	470
⅜	16	150	301	376	451	488	601	714
7/16	14	207	415	519	623	675	830	986
½	13	282	564	704	845	915	1,125	1,340
9/16	12	365	730	912	1,095	1,186	1,460	1,730
⅝	11	456	913	1,140	1,370	1,480	1,820	2,170
¾	10	690	1,380	1,725	2,070	2,240	2,760	3,280
⅞	9	964	1,930	2,410	2,900	3,140	3,860	4,580
1	8	1,265	2,530	3,170	3,800	4,120	5,060	6,010
1⅛	7	1,595	3,190	3,990	4,790	5,180	6,380	7,570
1¼	7	2,070	4,140	5,180	6,210	6,730	8,280	9,830
1⅜	6	2,440	4,890	6,110	7,330	7,940	9,780	11,600
1½	6	3,020	6,040	7,540	9,060	9,800	12,050	14,300
1⅝	5½	3,530	7,060	8,820	10,600	11,500	14,100	16,750
1¾	5	4,060	8,120	10,150	12,200	13,200	16,200	19,250
1⅞	5	4,800	9,600	12,000	14,400	15,600	19,200	22,800
2	4½	5,360	10,750	13,400	16,100	17,400	21,500	25,500
2¼	4½	7,120	14,200	17,800	21,400	23,100	28,500	33,800
2½	4	8,750	17,500	21,900	26,300	28,400	35,000	41,500
2¾	4	11,000	22,000	27,500	33,000	35,700	44,000	52,200
3	4	13,400	26,800	33,500	40,200	43,600	53,600	63,600
3¼	4	16,100	32,200	40,200	48,400	52,400	64,400	76,400
3½	4	19,000	38,100	47,600	57,200	61,900	76,200	90,400
3¾	4	22,200	44,500	55,600	66,700	72,300	89,000	105,500
4	4	25,700	51,400	64,200	77,000	83,400	102,800	122,000
4¼	4	29,350	58,700	73,400	88,100	95,400	117,400	139,300
4½	4	33,300	66,600	83,200	100,000	108,000	133,000	158,000
4¾	4	37,400	75,000	93,700	112,000	122,000	150,000	178,000
5	4	41,900	83,800	105,000	126,000	136,000	167,500	199,000
5¼	4	46,600	93,200	116,500	140,000	151,000	186,000	221,000
5½	4	51,500	103,000	129,000	154,500	167,000	206,000	244,500
5¾	4	56,700	113,500	142,000	170,000	184,000	227,000	269,000
6	4	62,000	124,000	155,000	186,000	202,000	248,000	295,000

OXY-ACETYLENE WELDING

BY

STUART PLUMLEY

The Oxy-acetylene Process as applied to the welding of most metals and the cutting of iron and steel has become universal in recent years in connection with almost every industry in which metals are used.

Gases. **For welding purposes,** oxygen and acetylene are almost exclusively used. **For brazing,** lead burning, etc., oxygen and hydrogen or oxygen and illuminating gas or natural gas are used to a large extent. **For cutting,** oxygen and acetylene are used to a greater extent than any other combination of gases, although for extremely heavy cutting, oxygen and hydrogen have been found advantageous. For all general cutting, the oxy-acetylene torch is preferable on account of its economy in consumption of oxygen and the rapidity with which the cut may be started.

Oxygen is produced to an enormous extent in the United States by two principal processes: (1) the liquefaction of air and the subsequent separation of the oxygen from the nitrogen and (2) the electrolysis of water. In the liquid air process the atmospheric air is reduced to liquefaction through pressure, refrigeration and expansion; in the liquid state and under pressure of a few pounds nitrogen, which liquefies at a lower temperature than that of oxygen, will evaporate first, and pass out of the machine. The oxygen is collected in gas form in gasometers, and is afterwards compressed in oxygen cylinders to a pressure of approximately 1,800 lbs. per sq. in. These cylinders are in daily use throughout the United States and may be obtained already charged from central distributing points located in almost every city. In the electrolysis process electric current is utilized in an electrolytic cell to break up water into its component parts, oxygen and hydrogen.

Acetylene is usually generated in an acetylene generator. This generator contains calcium carbide in a hopper at the top of the machine and automatically feeds it into the water in the lower part of the apparatus. The reaction between the carbide and the water produces acetylene gas, which is piped directly to the point where it is used in the welding shop. Acetylene may thus be produced locally in any quantity required, and at a low operating cost.

Acetylene may also be procured, compressed and dissolved in cylinders. These cylinders are charged at central producing points and shipped to the user as required. The cylinder is a special cylinder completely filled with a porous filler of cement, charcoal, infusorial earth, etc., the filler being saturated with acetone. Acetone has the property of dissolving acetylene and thus making it safe while under pressure. Acetylene tanks are especially useful when the welding or cutting to be done necessitates the use of portable apparatus.

Pressure regulators or pressure reducers are used to reduce the high pressure of oxygen in the cylinders to suitable lower pressures at the torch, and the much lower pressures of acetylene flowing from the acetylene cylinders (or from the acetylene generator, if in use). **Pressure gages** indicate the pressures in the cylinders and at the torch. Regulators should always receive careful treatment. They are a sensitive diaphragm mechanism, and the service is severe. If properly used they will accurately regulate the gas pressure.

The Oxy-acetylene Burner, known as the "**torch,**" is connected to the source of supply by two lengths of hose. It is simple in construction. Fig. 1 shows how the oxygen and acetylene are brought together and intimately mixed in an interchangeable tip. This tip varies with the thickness of weld and size of flame required. It is essential that the two gases be brought together under independent pressures, and that at the point of mixture the gas jets be directed together so as to produce the most thorough intermingling.

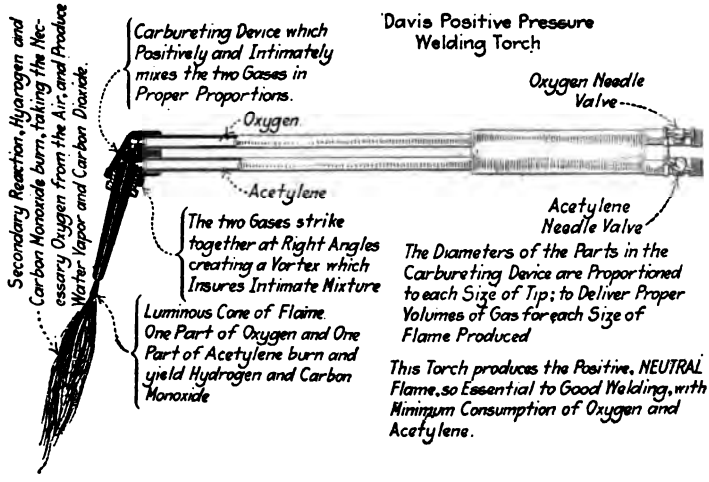

Davis Positive Pressure Welding Torch
Carbureting Device which Positively and Intimately mixes the two Gases in Proper Proportions.
Oxygen Needle Valve
Oxygen
Acetylene
Acetylene Needle Valve
Secondary Reaction, Hydrogen and Carbon Monoxide burn, taking the necessary Oxygen from the Air, and Produce Water Vapor and Carbon Dioxide.
The two Gases strike together at Right Angles creating a Vortex which Insures Intimate Mixture
Luminous Cone of Flame. One Part of Oxygen and One Part of Acetylene burn and yield Hydrogen and Carbon Monoxide
The Diameters of the Parts in the Carbureting Device are Proportioned to each Size of Tip; to Deliver Proper Volumes of Gas for each Size of Flame Produced
This Torch produces the Positive, NEUTRAL Flame, so Essential to Good Welding, with Minimum Consumption of Oxygen and Acetylene.

FIG. 1.—Welding Torch (See Table 1.)

Table 1. Welding Torches

(*Davis-Bournonville Co. Style "C"*)

(Approximate Acetylene and Oxygen Pressures with Styles 99 and 100 Tips)

Tip No.	Thickness of metal	Acetylene pressure	Oxygen pressure
00	Very Light	1 lb.	1 lb.
0		1 lb.	2 lbs.
1	1/32–1/16 in.	1 lb.	2 lbs.
2	1/16–3/32 in.	2 lbs.	4 lbs.
3	3/32–1/8 in.	3 lbs.	6 lbs.
4	1/8 –3/16 in.	4 lbs.	8 lbs.
5	1/4 –5/16 in.	5 lbs.	10 lbs.
6	5/16–3/8 in.	6 lbs.	12 lbs.
7	7/16–1/2 in.	6 lbs.	14 lbs.
8	1/2 –5/8 in.	6 lbs.	16 lbs.
9	5/8 –3/4 in.	6 lbs.	18 lbs.
10	3/4 – Up	6 lbs.	20 lbs.
11	Extra Heavy	8 lbs.	22 lbs.
12		8 lbs.	24 lbs.

For convenience, operators frequently adjust regulator to 2 lbs. above that shown in table for the size of tip being used, which will allow for adjusting for drop caused by lowering of pressure in tank, variations in gages, or other local causes.

Character of Flame. As shown in Fig. 1, the combustion of oxygen and acetylene produces a luminous cone of flame at the tip of the torch which has a temperature of approximately 6,300 deg. F. This cone of flame is large or small, depending upon the size of the tip in use. If a **heavy section of metal** is to be welded, a large tip with a correspondingly large cone is required. In **sheet-metal** welding the tips are small and the cone of flame approaches the size of a pencil point.

For **iron and steel, cast iron, cast steel,** etc., a neutral flame is required. The neutrality of the flame is a very essential feature in connection with the welding of these metals. By a slight change in the adjustment of the needle valves of the torch the flame may become neutral, oxidising or carbonising. The carbonising flame has an excess of acetylene, and the oxidising flame has an excess of oxygen. The flame is neutral when there is just enough oxygen and acetylene to produce combustion. If the welder does not use proper care in keeping the flame neutral he cannot expect to produce a good weld. Poor welding is often the result of carelessness in this respect.

Oxy-acetylene Welding consists of bringing the temperature of the two parts to be welded to the point of fusion by means of the torch flame. When in its molten condition, new metal is added to the fused mass by melting in it special welding stocks or rods made up of a suitable material for the work in process. When the two pieces to be joined have been fused or melted together in a mass without materially changing the characteristics of the metal, a weld has been accomplished.

It will be appreciated, however, that in certain metals such fusion, or in fact the application of any degree of heat above the critical temperature, will utterly change the characteristics of the metal, and a weld in such instances cannot be made by the simple fusion of the pieces. This is true of malleable iron which, while it may not be welded as above described without re-annealing, may be satisfactorily brazed with brass as a joining material under proper conditions.

Expansion and Contraction. All metals expand with heat and contract with cold. The amount of expansion caused in a steel plate by a considerable rise in temperature is relatively small, but it acts with great force. In making a weld between any two pieces of metal **consideration must be given to the effect of** expansion and contraction. This is particularly true when dealing with a casting, especially one having restricted or confined members.

Pre-heating. The stresses produced by expansion and contraction in the welded pieces may be so great as to break the weld or crack it when cold. To avoid such stresses the casting should always be pre-heated slowly and evenly by means of a flame which has a lower temperature and is not so concentrated as that produced by the oxy-acetylene torch.

The most suitable apparatus for pre-heating is a **fuel oil or kerosene burner,** which produces a large, long, soft flame. The casting to be heated is set upon a layer of brick upon the floor of the shop, if the floor is of concrete, and in a convenient position for the welder. A rough fire-brick wall is then built up around it. The brick wall is covered over with a piece of sheet iron to retain the heat. The casting is thus heated evenly all over and each member is expanded. After the welding has been accomplished the casting is re-heated to take care of any stresses which may have been set up, and then allowed to cool very slowly by being covered with asbestos, hot sand, lime or ashes, to exclude the air. Such treatment will reduce stresses in the adjacent metal which might otherwise have been set up.

If the fuel oil pre-heaters are not obtainable **charcoal** may be used, preferably **with a little compressed air.** If neither of these can be obtained the casting may be set up on the fire brick so that the air will reach the charcoal fire. If the casting is extremely large it may be preferable not to heat it in its entirety, but any portion of it may be heated by building up that portion as described above. Large castings, however, require considerable skill in this respect so that preheating may be applied to the right members.

Size of Flame is governed by the amount of gas passing through the ports, and this flow of gas is controlled by the size of the interchangeable tip. The tip should be chosen for the thickness of metal. Sheet metal will require a small flame and a small tip, while heavy plates require larger flames and larger tips. The size of the flame must, in short, bear a certain relation to the size of the work in order to obtain best results. Larger plates absorb and dissipate the heat rapidly, and if insufficient gas is burned to raise the joint to the fusing temperature welding cannot be accomplished. Absorption of heat is reduced by preheating, and rapid dissipation of heat is prevented by protecting the parts with some heat insulator like sheet asbestos, fire brick, etc.

Preparing the Joint. If the parts to be welded are thin, little or no special preparation of the joint may be required, but on heavy parts it is necessary to chip away the edges to an angle of not less than 90 deg. This is necessary in order that fusion of the joint may begin at its lower surface. When the torch is applied to the bottom of the *V* the lower edges are fused, and the sides of the parts are slowly raised to a fusing temperature.

When fusion has been accomplished throughout the immediate part of the section being welded, adding material is supplied by means of a suitable welding rod. It will, however, be appreciated that unless the joint is chipped away to permit the welder to begin at the bottom of the part, he cannot expect to create fusion clear through the section.

Adding Material (Welding Rods). Adding material is usually made up in the form of wire of various diameters and approximately in 36-in. lengths, or in cast form of various diameters and approximately in 20-in. lengths. These rods, which are generally called welding rods, must be of a suitable alloy for the work in hand.

Most manufacturers, through a long series of experiments, have developed that certain alloys combined with certain fluxes produce the best results under given conditions, and it is essential that the welder keep this in mind in selecting a suitable welding rod for the job to be accomplished.

While these rods are usually made up upon a special analysis or formula proven best by experimental work, it is also necessary that the rods be made up by a special method and under definite conditions in order to produce satisfactory results. Two rods of like analysis will not always produce the same result when used for welding, unless the method of manufacture has been the same.

Manipulation of Torch see Fig. 2. Select a suitable size of tip for the work to be accomplished. This is generally shown in the manufacturer's pamphlets. Connect the torch to the source of supply of oxygen and acetylene, also as described in the manufacturer's directions. Open the valves, adjust the pressures and light the torch as indicated for the particular torch in use.

It will be noted that when the pressure of each gas is properly adjusted the flame will possess a clearly defined luminous cone. If a little more acetylene is turned into the torch a greenish tinge of flame will appear at the tip

of the luminous cone. If a little more oxygen is turned on the cone will become shorter and more transparent and less luminous. To be neutral the flame should be just between these points. There should be no greenish tinge, but the flame should be clearly defined and luminous. (See Fig. 1 for outline appearance of flame.)

The work having already been prepared or preheated as has been described heretofore, hold the torch in the right hand and the adding material in the left. Play the flame slowly along the edges to be united, beginning always at the edges farthest away from the operator. Play the torch momentarily at this point and quickly fuse the sides of the V until a puddle is formed. Apply the adding material in actual contact with the puddle and oscillate the torch

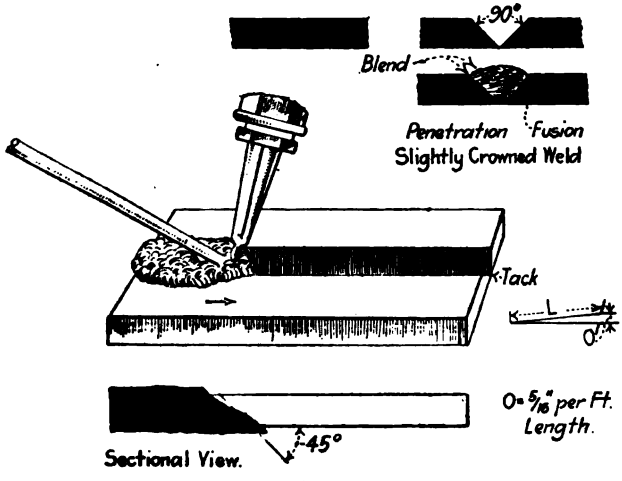


FIG. 2.—Manipulating the Torch.
(Courtesy Davis-Bournonville Welding Institute.)

slowly backward and forward in a semi-circle, keeping the flame and the tip pointing to the completed work. Oscillate the welding rod in similar manner to the torch, but in the opposite direction, which will keep the puddle free from inclusions of gas. Continue this motion and keep the puddle flowing constantly until the weld is complete or until the operator has reached the point where he can permit the work to cool down. Do not permit the hot molten puddle to flow over the cold unmolten surface. If this is done, fusion between the puddle and the cooler surface will not occur, and a lap will be produced. Apply the torch in such manner that the parts being welded are always in a state of fusion before melting the welding rod into the puddle.

Fluxes. The effect of a flux is to lessen oxidation and reduce the oxide film that prevents perfect union. The choice of flux is important, as success may hinge on the flux used. Home-made fluxes generally lack the careful proportioning of ingredients and the thorough grinding and mixing necessary for best results.

Direction of Welding. When welding thin steel it is customary to tack at the left and weld from the right. But when thick sections are welded the welding should proceed from the left, as shown in Fig. 2. The advancing puddle and side walls are thus kept directly under the flame, and lapping and cold shuts are more easily avoided.

Welding

Welding Cast Iron. Cast iron should always be preheated as described on p. 165. The edges of the crack or break should be carefully prepared before welding, by chipping away the metal so that there will be ample room to permit the welder to start fusion at the bottom of the groove. The casting should be placed in position so that its parts are in proper alignment and will remain so during welding. It is preferable to have the crack or break to be welded on the upper side of the casting.

When this has been accomplished the casting may be bricked up, using a rough firebrick wall, and preheated as above described. When the casting has reached a temperature a little less than dull red, the brick or cover may be removed to permit the welder to work.

In welding cast iron a special cast-iron welding rod should always be used together with a proper flux. It is essential that the **welding rod be high in silicon and of good grade of cast iron,** and the flux properly prepared to keep the metal flowing smoothly.

As the sides of the V-break are fused and the added material united with the molten mass the welder uses the rod to scrape away any dirt or slag that may be encountered and to break up inclusions of gas. When the weld is completed in this manner the casting should be covered up and the preheating burners re-lighted for a short period of time to bring the heat again sufficiently high to eliminate any strain or stresses which may have been set up. The casting should then be completely covered up and allowed to cool very slowly.

Welding Malleable Iron. Welding malleable iron **is not advisable except** where high strength of the joint is not required. The effect of the process by which white cast iron is converted into so-called malleable iron is destroyed by fusing the metal, as it is changed again to brittle cast iron. Broken malleable parts may be strongly united by brazing with bronze welding rods and flux. Care should be observed that too high a heat is not used in brazing. A full red heat produced by a neutral flame will yield best results. The iron and bronze should freely alloy in order to give a strong joint.

Welding Steel. In welding steel, the welding rod used consists in most instances of a good grade of domestic iron wire, liquor drawn and annealed. In some instances it is preferable to use a low-carbon steel as a welding rod, but this depends upon the immediate case and the local conditions.

The plates should be prepared by chipping the edges to an angle of 90 deg., and fusion should be produced at the bottom of the V and along its sides, before any welding wire is added. If the welder is not careful in this respect a series of cold-shuts and laps will be produced. Care must be taken not to burn the welding rod, and the puddle produced must be continually maintained as far as is practicable. The speed of rotation of the flame as it is manipulated by the welder, and its advance along the seam, must be suited to the work in hand.

Welding Steel Castings. Steel castings vary widely in carbon content, and no fixed rule can be given for welding. When the carbon content is

below twenty points (0.20), welding can be accomplished with ease, but castings with higher percentage of carbon are likely to give trouble. A little copper added to the weld will assist in difficult cases, but its use in general is to be avoided, as it hardens the weld.

Welding Cast Aluminum. The aluminum parts to be welded should be thoroughly cleaned to remove all grease, oxide and foreign material. The edges should be beveled the same as cast iron and steel if the thickness exceeds ¼ inch. **Preheating is highly important** in the case of aluminum, on account of its rapid heat absorption and dissipation. To secure perfect welds it is necessary to preheat comparatively small parts all over to a temperature of about 600 deg. F. Adding material is supplied in alloy sticks which should be free from sand or other foreign matter. A flattened iron rod is required to puddle the melted aluminum, break up the oxide and make it fill the joint. Care should be taken not to heat the end of the rod to a high temperature, as the resulting iron oxide will mix with the melted aluminum and cause a defective weld.

Welding Sheet Aluminum. Sheet aluminum is welded with aluminum flux and drawn aluminum wire adding material. Care must be taken to choose a size of tip suited to the work. The operator must work rapidly, as thin sheets heat to the fusing point quickly, and the welding must be accomplished without delay in order to obtain a smooth, clean job.

Welding Copper. Copper welding requires a larger tip and flame than steel of the same gage, because of the greater heat radiation, and the preheating of comparatively large pieces is necessary. It is desirable to use a flux and a strictly neutral flame to prevent oxidation and consequent weakening of the joint. The joint has the tensile strength of cast copper, but it may be increased by judicious hammering.

Welding Brass and Bronze. Care should be exercised in welding brass and bronze, not to overheat the metal and volatilize the zinc or tin. The white fumes resulting from overheating with the torch are poisonous, and the operator should avoid breathing them. A flux should be used, and preheating will be necessary on large pieces. Use special cast-bronze welding rod for adding material.

Welding Cast Iron and Steel. The steel must first be heated to the welding temperature, as its fusing point is higher than that of cast iron. The torch flame is then directed against the cast iron, and, as fusion begins, adding material is supplied from regular cast-iron welding rods or sticks. The parts to be welded should be prepared the same as described for other metals. A cast-iron welding flux will facilitate the work, and is recommended.

Welding Copper and Steel. Heat the steel to the welding point and apply the copper, letting it absorb heat from the steel until it begins to melt. The flame should be withdrawn as soon as fusion starts.

Welding High-speed Steel and Machine Steel. High-speed steel may be used economically for lathe and planer tools by welding cutting bits to machine steel shanks. The high-speed steel piece should be fitted to a notch in the end of the shank, which will afford support beneath. The sides of the bit to be welded should be heavily coated or "tinned" with soft iron welding rod, and when this is accomplished the welding to the shank is done in the usual way. The welding of high-speed drills to machine-steel shanks by the oxy-acetylene process can be done successfully, but is not advised unless special facilities are provided. If necessary for special work to provide

a few long-shank high-speed drills, scarf the shanks and drills, and unite by brazing, using bronze welding rod and flux.

Welding Dissimilar Metals in General. Metals having widely different melting temperatures may be successfully united by coating or "tinning" the metal having the higher melting point, with the metal having the lower melting point. Then the two are fused together. The flux to be used depends on the metals to be welded.

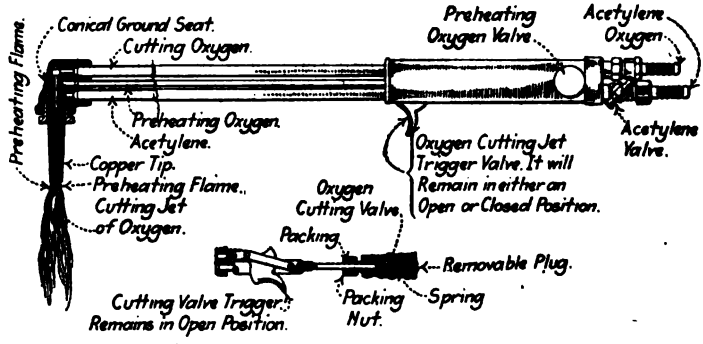


FIG. 3.—Cutting Torch.

Table 2. Cutting Torches

(*Davis-Bournonville Co. Style "C."*)

(Approximate Acetylene and Oxygen Pressures with Style 12 Tips)

Tip No.	Thickness of metal	Acetylene pressure	Oxygen pressure
1	1/8 in.	3 lbs.	10 lbs.
1	3/16 in.	3 lbs.	15 lbs.
1	1/4 in.	3 lbs.	20 lbs.
1	5/16 in.	3 lbs.	20 lbs.
2	1/4 in.	3 lbs.	10 lbs.
2	1/2 in.	3 lbs.	20 lbs.
2	3/4 in.	3 lbs.	30 lbs.
2	1 in.	3 lbs.	35 lbs.
3	1 in.	4 lbs.	30 lbs.
3	1 1/2 in.	4 lbs.	40 lbs.
3	2 in.	4 lbs.	50 lbs.
3	3 in.	4 lbs.	60 lbs.
4	3 in.	5 lbs.	60 lbs.
4	4 in.	5 lbs.	70 lbs.
4	5 in.	5 lbs.	85 lbs.
4	6 in.	5 lbs.	100 lbs.
5	6 in.	6 lbs.	90 lbs.
5	7 in.	6 lbs.	100 lbs.
5	8 in.	6 lbs.	125 lbs.
5	10 in.	8 lbs.	150 lbs.

Cutting

Cutting Iron and Steel. A special type of oxy-acetylene torch is made for cutting wrought iron and steel. The action of cutting iron and steel is based on the principle that wrought iron and steel will be oxidised or burned when raised to a red heat and exposed to a jet of oxygen gas.

The apparatus for cutting differs in the type of torch and the oxygen working pressure, the pressure being much higher for cutting than for welding. The construction of a cutting torch is shown in Fig. 3.

Machines for Cutting. A number of ingenious machines have been developed for holding and guiding the cutting torch. They make it possible to cut die blanks and all sorts of forms accurately from thick plates.

Speed of Cutting. The speed of cutting depends upon the size of the torch, gas consumption, and thickness of steel. It is not unusual to cut ¾-in. steel at the rate of 15 in. per min.

Depth of Cutting. The depth or thickness that can be cut is remarkable. Armor plate up to 16 to 18 in. thick can be cut quicker and cheaper than by any other method. Cutting to depths of 6 to 8 in. is well within the skill of ordinary operators. With mechanically operated torches thicknesses up to 24 in. may be readily cut.

THE THERMIT WELDING PROCESS

BY

JOHN H. DEPPELER

REFERENCE: R. L. Browne, paper presented before N. Y. R. R. Club, April 19, 1918.

This process was first introduced commercially in 1902 and is fundamentally adapted to the **welding of heavy cast iron and steel** sections. Sections under 1 in. in thickness and sheet steel do not readily lend themselves to the Thermit process, therefore this process does not conflict with oxy-acetylene, electric or other autogenous welding processes which are specially adapted to welding such parts. In **marine repairs**, the field of application of the Thermit Process embraces the welding of broken stern frames, rudder frames, crank shafts, tail shafts, anchors, davits and similar heavy sections of wrought iron and steel.

This process is specially valuable for large marine repairs owing to the fact that dry dockage is extremely costly. Such repairs as stern post welds are usually executed by contract and seldom keep the vessel in dry dock more than 48 hours. The U. S. Navy was one of the first to appreciate the advantage of the process and has used it extensively at its various yards. A number of Navy repair ships are completely equipped with Thermit welding outfits.

Composition. Thermit is a mixture of finely divided **aluminum and iron oxide** in such proportions that upon reaction the Al combines with all the O of the iron oxide according to the formula $2Al + Fe_2O_3 = 2Fe + Al_2O_3$. An approximate temperature of at least 2200 deg. F. **is required to start the reaction.** This temperature is obtained by means of a special barium compound ignition powder.

The reaction, which is started in one spot, supplies its own heat and spreads gradually throughout the entire mass. The aluminum oxide (slag) floats on top of the iron which is precipitated to the bottom of the containing vessel in the form of a highly superheated low carbon liquid steel. **The temperature of the reaction** is not measureable by any pyrometer but has been calculated theoretically by *Professor Joseph W. Richards* of Lehigh University as 4881 deg. F. The reaction of quantities of Thermit ranging from a few pounds to 1000 lbs. is completed in from 25 to 50 sec.

Thermit produces half its weight in liquid steel but in practice the amount of steel is increased by mixing from 15 per cent to 25 per cent of small mild steel punchings with the Thermit mixture: less than 500 lbs. of Thermit about 15 per cent of punchings; from 600 to 1,000 lbs. of Tnermit, 25 per cent of punchings.

Kinds of Thermit. At the present time there are three varieties of Thermit produced for welding purposes: (1) Railroad Thermit; (2) Plain Thermit; (3) Cast Iron Thermit.

Plain thermit is simply a mixture of aluminum and iron oxide and is used in making pipe welds where the Thermit is merely used as a heating agent to bring the pipe ends up to a welding temperature.

Railroad thermit is Plain Thermit with the addition of $\frac{5}{8}$ per cent nickel, 1 per cent manganese and 15 per cent mild steel punchings. This grade of Thermit is used in connection with all steel and wrought iron welds.

Cast iron thermit is Plain Thermit with an addition of 3 per cent ferrosilicon and 20 per cent mild steel punchings and is used for welding cast iron parts.

Physical properties of Railroad Thermit shown by recent tests made on a sample taken from one of the gates of a weld and forged are: Tensile strength 72,000; elastic limit, 58,400; elongation in 2 in., 17.5 per cent; reduction in area, 64 per cent.

Large welds are made by casting Railroad Thermit steel around and between the parts to be welded together. The parts to be united are first arranged with a space between them, the extent of which depends upon the size of the sections to be welded, see table 1; a section 6 in. × 6 in. would call for 1½ in. opening.

TABLE 1. Welding Portions For Welding Rectangular Sections

Width of Section	Depth of Section	Width of Thermit Steel Collar	Thickness of Thermit Steel Collar at Center	Quantity of Thermit* Required for Weld
Inches	Inches	Inches	Inches	Pounds
3	2	4	1	40
3	2½	4	1	40
3	3	4	1	45
3	3½	4	1	50
3	4	4	1	55
4	4	4	1	65
4	4½	4	1	65
4	5	4	1	70
4	5½	5	1¼	75
4	6	5	1¼	75
4½	4½	5	1¼	70
4½	5	5	1¼	75
4½	5½	5	1¼	75
4½	6	5	1¼	80
5	5	5	1¼	75
5	5½	5	1¼	80
5	6	6	1½	85
5	7	6	1½	90
5½	5½	6	1½	85
5½	6	6	1½	90
5½	7	6	1½	110
6	6	6	1½	100
6	6½	6	1½	120
6	7	7	1⅝	130
6½	6½	7	1⅝	130
6½	7	7	1⅝	150
6½	8	7	1⅝	160
7	7	7	1⅝	155

*Railroad Thermit for Wrought Iron and Steel.
Cast Iron Thermit for Cast Iron.

Where the piece to be welded is in two parts it is a simple matter to provide this space but in the case of a crack, it is necessary to cut out sufficient steel to provide this space. The object of this space is to properly preheat the ends to be welded and confine enough superheated Thermit steel around and between these ends to thoroughly melt them up and amalgamate with them.

After the space is provided, a wax pattern is formed around the ends to be united of the exact shape of the reinforcement of Thermit steel which is to be cast at this point to make the weld. The object of this reinforcing collar is principally to provide sufficient heat to assure perfect fusion of the interior of the weld and the complete welding of the outer fibers.

A sand mold is next rammed around the wax pattern and inside a sheet iron box, provision being made in the mold at the lowest point of the wax pattern for a pouring gate and a heating gate and at the highest point for a riser. The gates and riser openings in the mold are made with the aid of wooden patterns which are withdrawn after the mold is completed, Fig. 1.

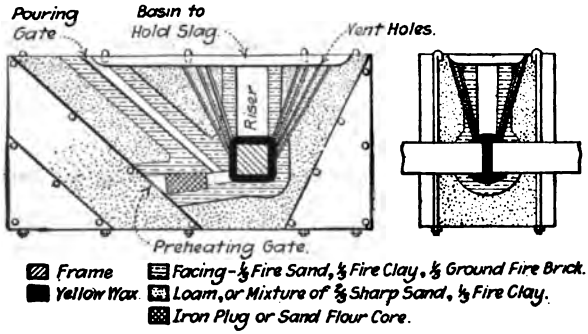
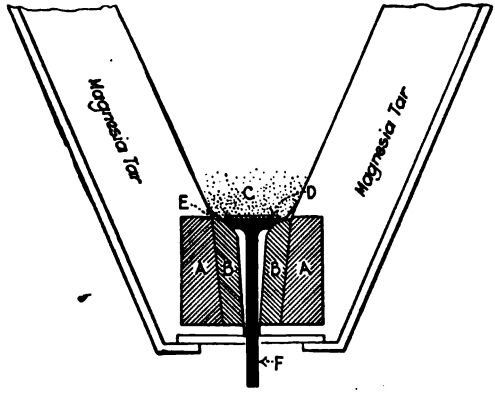

Pouring Gate
Basin to Hold Slag
Vent Holes.
Riser
Preheating Gate.
Frame
Yellow Wax
Facing-⅓ Fire Sand, ⅓ Fire Clay, ⅓ Ground Fire Brick.
Loam, or Mixture of ⅔ Sharp Sand, ⅓ Fire Clay.
Iron Plug or Sand Flour Core.

FIG. 1.—Mold Construction, Thermit Welding.



AA—Magnesia Stone.
BB—Magnesia Thimble
C—Refractory Sand.
D—Metal Disc.
E—Asbestos Washer.

FIG. 2.—Crucible Section, Showing Plug.

The function of the vent holes is primarily to vent the interior of the mold and prevent the occlusion of gases. The basin shown on the top of the mold is for the purpose of catching the slag.

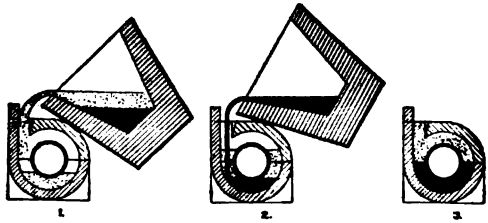
The flame of a compressed air gasoline or kerosene heater directed into the heating gate melts the wax pattern and leaves a space (later occupied by the

Thermit steel) between and around the ends to be welded. After the wax is melted out heat is continued until the parts to be welded have been brought to a red heat and the mold is thoroughly dried out.

The charge of Thermit is placed in a **conical shape sheet steel crucible lined with magnesite** and supported over the pouring gate of the mold When the sections to be welded are red hot and the heat has been allowed to soak in so that the pieces will retain this heat, the burner is removed, the heating gate plugged up, and the Thermit charge in the crucible ignited. When the reaction is completed the Thermit steel is tapped from the bottom of the crucible into the mold in bulk where it flows around and between the sections to be welded together uniting them into one solid mass forming a cast weld. Fig. 2 shows the cross section of the crucible plugged ready to receive the Thermit.

The mold is allowed to stay in place sufficiently long to insure **slow cooling.** When the mold is stripped, excess metal in the pouring gate and riser is removed and, if necessary, the Thermit steel collar surrounding the weld can also be removed. (If not necessary, it is left on to add strength to the repair). The **advantages of the process** for large welding operations are:

(1) Portability of outfit enabling welds to be made anywhere that compressed air is available; (2) welds made without removing broken section from their positions; (3) steel to make the weld is produced in bulk and poured around the sections in one operation so that internal shrinkage strains are eliminated. The only shrinkage to allow for is shrinkage tending to draw the two parts together and allowance for this can be easily and accurately made; (4) results are not dependent on the skill of the operator, as in the case of other welding processes.



THERMIT PIPE WELDING OPERATION

1—Slag flowing into mold and coating outside of pipe and inside of mold

2—Slag in mold and steel following, displacing slag in bottom part

3—Both slag and steel in mold but steel separated from pipe

FIG. 3.

Crank Shaft Repairs. Heavy crank shaft repairs present a very interesting application of the Thermit Process. There is no limit to the size or weight of shaft that can be welded but the shafts most frequently encountered vary from 6 in. diameter up to 14 in. diameter, and sometimes weigh as much as 10 tons. In making a crank shaft weld the parts of the shaft are supported in the proper position by "V" blocks mounted on a machined bed plate. The fracture is cut out and a space provided in the usual manner. The parts of the shaft are separated a certain amount to allow for the contraction which will subsequently take place, and held rigidly in line by means of clamps

during the operation of ramming the mold. After the mold is rammed, these clamps are removed so that both ends of the shaft will be free to move during solidification and contraction of the weld, thus avoiding all possibility of the setting up of shrinkage strains. In fact, the shaft can not only move lengthwise in the "V" blocks but the "V" blocks are so arranged as not to restrict even side motion of the shaft. It is obviously impossible to guarantee perfect alignment of the welded shaft because it is most essential that no shrinkage strains be introduced, and slight inaccuracies in alignment are of lesser importance. It is furthermore a fact that the strain that caused the fracture of the shaft originally, undoubtedly was sufficient to distort the shaft at other points and thus necessitates machining subsequent to welding. It is always understood, therefore, in connection with crank shaft repairs that, after welding, a slight finishing cut will have to be taken from the weld to the shorter end of the shaft. This necessitates the relining of the bearings at this end to compensate for the slight reduction in shaft diameter. Usually it is not necessary to machine the pin journals as the inaccuracy is very slight. In the event of an end main journal being broken off, it is usually preferable to discard the piece and weld on a new billet slightly larger in diameter, thus necessitating no change in bearing sizes.

Pipe Welds. Plain Thermit is used for butt welding pipe lengths together. A cast iron mold is provided and the pipe ends are accurately faced and held tightly together by clamps. The Thermit is reacted in a flat bottom crucible similar to a small foundry ladle, but lined with magnesite, and on completion of the reaction the contents are poured into the mold, Fig. 3. Slag first enters and coats the inside of the mold and the outside of the pipes with a protective coating. This prevents the steel (which follows the slag) from touching either the mold or pipes. In this case the Thermit steel does not comprise the weld but acts merely as a heating agent, advantage being taken of the extremely high temperature of the chemical reaction to bring the pipe ends to a welding heat, at which time they are drawn together by means of the clamps and a butt weld obtained. These welds are largely used for ammonia refrigerating pipe. **Two men can make from 40 to 50 medium size pipe welds per day.**

In iron and steel foundry practice, Thermit is used to increase the temperature of iron and steel. For this purpose it is put up in cans that may be introduced into the foundry ladle. It is also used in casting risers to keep the metal liquid for a longer time than usual. This allows a better opportunity for gases to escape, and thus improves the quality of castings.

HEAT

BY

G. A. GOODENOUGH

From the "Mechanical Engineers Handbook" by courtesy of the Editor-in-Chief thereof, Lionel S. Marks.

REFERENCES: Preston, "Theory of Heat," Macmillan. Zeuner, "Technical Thermodynamics," Van Nostrand. Goodenough, "Principles of Thermodynamics," Holt. Marks and Davis, "Steam Tables and Diagrams," Longmans Green. Bryan, "Thermodynamics," Teubner. Peabody, "Thermodynamics of the Steam Engine," Wiley. Locke, "Engineering Thermodynamics," McGraw-Hill. Bulletins 30, 40, 66, 75, Engineering Experiment Station, Univ. of Illinois.

THERMAL PROPERTIES OF BODIES

The Measurement of Temperature

Thermometers. The scale of the constant-volume **hydrogen thermometer** is taken as the standard temperature scale. Within the limits of ordinary use the scale of the mercury thermometer agrees closely with the standard scale, but above 500 deg. fahr. the divergence between the two scales may be appreciable.

The ordinary **mercury thermometer** may be used to about 600 deg. fahr.; this limit may be extended to 1000 deg. fahr. if the capillary tube above the mercury is filled with nitrogen or carbon dioxide under high pressure. The lower temperature limit for the mercury thermometer is −39 deg. fahr. For lower temperatures, alcohol, pentane or petroleum ether may be used as the thermometric substance. Very high temperatures are measured by various forms of **pyrometers.** See pp. 316–318.

Thermometer Scales. Let F and C denote the readings on the Fahrenheit and Centigrade (or Celsius) scales, respectively, for the same temperature; then

$$C = \tfrac{5}{9}(F - 32), \qquad F = \tfrac{9}{5}C + 32$$

Fixed Temperatures. The fixed temperatures in Table 1, adopted by the U. S. Bureau of Standards, are useful in the calibration of pyrometers.

Table 1. Fixed Temperatures, U. S. Bureau of Standards

	Deg. fahr.	Deg. cent.
Liquid tin solidifies at	449	232
Liquid lead solidifies at	621	327
Liquid zinc solidifies at	787	419.4
Liquid sulphur boils at	832.5	444.7
Liquid antimony solidifies at	1167	630.5
Liquid aluminum (97.7 per cent pure) solidifies at	1216	658
Solid gold melts at	1947	1064
Liquid copper solidifies at	1983	1084
Solid nickel melts at	2615	1435
Solid palladium melts at	2815	1546
Solid platinum melts at	3187	1753

High Temperatures and Color. High temperatures may be judged approximately by color, though the estimate of an experienced observer is likely to be 100 deg. fahr. from the true value. The following table associating color and temperature of iron or steel is due to White and Taylor.

	Deg. fahr.		Deg. fahr.
Dark blood red, black red...	990	Orange, free scaling heat.....	1650
Dark red, blood red, low red.	1050	Light orange.....	1725
Dark cherry red.....	1175	Yellow.....	1825
Medium cherry red.....	1250	Light yellow.....	1975
Cherry, full red.....	1375	White.....	2200
Light cherry, light red.....	1550		

Expansion of Bodies by Heat

Coefficients of Expansion. The **coefficient of linear expansion** of a solid is defined as the increment of length in a unit of length for a rise in temperature of 1 deg. Likewise, the **coefficient of cubical expansion** of a solid, liquid, or gas is the increment of volume of a unit volume for a

Table 2. Coefficients of Linear Expansion

(For pure metals, see p. 41)

COEFFICIENTS OF LINEAR EXPANSION
(Mean values of 10,000a' between 32 and 212 deg. fahr.)

METALS					
Aluminum bronze.....	0.094	Steel:		Graphite.....	0.044
Brass, cast.....	0.104	Bessemer, rolled hard.	0.056	Gutta percha	1.100
Brass, wire.....	0.107	Bessemer, rolled soft.	0.063	Limestone.....	0.014
Bronze.....	0.100	Nickel (10% Ni).....	0.073	Marble.....	0.065
Constantan (60 Cu, 40 Ni).....	0.095	Type metal.....	0.108	Masonry.....0.025 to	0.050
German silver.....	0.102	OTHER MATERIALS		Porcelain.....	0.017
Iron:		Brick.....	0.031	Rubber.....	0.428
Cast.....	0.059	Caoutchouc.....	0.372	Vulcanite.....	0.400
Soft forged.....	0.063	Carbon—coke	0.030	Wood (‖ to fiber):	
Wire.....	0.080	Cement, neat.....	0.060	Ash.....	0.053
Magnalium (86 Al, 13 Mg).....	0.133	Concrete.....	0.080	Chestnut and maple	0.036
Phosphor bronze.....	0.094	Ebonite.....	0.468	Oak.....	0.027
Solder.....	0.134	Glass:		Pine.....	0.030
Speculum metal.....	0.107	Thermometer.....	0.045	Across the fiber:	
		Hard.....	0.033	Chestnut and pine..	0.019
		Plate and crown....	0.050	Maple.....	0.027
		Granite.....	0.048	Oak.....	0.030

Table 3. Coefficients of Cubical Expansion

(Mean values of 1000a''' at ordinary room temperatures)

LIQUIDS					
Acetic acid.....	0.80	Hydrochloric acid.....	0.27	Sulphuric acid, 50% solution.....	0.45
Alcohol (ethyl).....	0.61	Hydrochloric acid, 50% solution.....	0.52	Turpentine.....	0.56
Alcohol (methyl).....	0.80	Mercury.....	0.10	Water.....	0.10
Benzene.....	0.77	Olive oil.....	0.41	SOLIDS	
Benzol.....	0.70	Petroleum.....	0.55	Fluorspar.....	0.035
Calcium chloride ($CaCl_2$); 5 to 50% solution.	0.28	Phenol (C_6H_6O).....	0.50	Ice (4 to 30 deg. fahr.)	0.62
Chloroform.....	0.77	Rape-seed oil.....	0.50	Paraffin wax.....	0.61
Ether.....	1.20	Sodium chloride:		Rock salt.....	0.67
Glycerine.....	0.28	1.6% solution.....	0.60	Sulphur.....	0.40
		26% solution.....	0.24	Wood (beech).....	0.016
		Sulphuric acid.....	0.27	Wood (pine).....	0.028

rise of temperature of 1 deg. Denoting these coefficients by a' and a''', respectively,

$$a' = \frac{1}{l}\frac{dl}{dt} \qquad a''' = \frac{1}{V}\frac{dV}{dt}$$

in which l denotes length, V volume, and t temperature. For homogeneous solids $a''' = 3a'$, and the **coefficient of superficial expansion** $a'' = 2a'$.

The coefficients of expansion are in general dependent upon the temperature, but for ordinary ranges of temperature constant mean values may be taken. If lengths, areas, and volumes at 32 deg. fahr. (0 deg. cent.) be taken as standard, then these magnitudes at other temperatures t_1 and t_2 are related as follows:

$$\frac{l_1}{l_2} = \frac{1+a't_1}{1+a't_2} \qquad \frac{A_1}{A_2} = \frac{1+a''t_1}{1+a''t_2} \qquad \frac{V_1}{V_2} = \frac{1+a'''t_1}{1+a'''t_2}.$$

Since for solids and liquids the expansion is small, the preceding formulæ for these bodies become approximately

$$l_2 - l_1 = a'l_1(t_2 - t_1). \quad A_2 - A_1 = a''A_1(t_2 - t_1). \quad V_2 - V_1 = a'''V_1(t_2 - t_1)$$

For certain metals the variation of the coefficient of expansion with temperature has been investigated by Holborn and Day and by Dittenberger. Denoting by l_0 the length at 32 deg. fahr., and by l the length at temperature t, the following relation is obtained.

$$l = l_0\left[1 + a\left(\frac{t-32}{1000}\right) + b\left(\frac{t-32}{1000}\right)^2\right]$$

The following table gives values of the constants.

Metal	1000 a	1000 b	Temperature range, deg. fahr.
Aluminum	13.380	2.182	32–1130
Cast iron	5.441	1.747	32–1160
Ingot iron	6.375	1.636	32–1380
Malleable iron	6.503	1.622	32– 930
Ingot steel	6.212	1.623	32–1380
Copper	9.278	1.244	32–1160
Nickel	7.652	1.023	32–1830

The tensile or compressive **stress set up** in a prismatic bar by a **temperature change** of t deg. is $P = a'EAt$, in which E denotes the modulus of elasticity and A the area of the cross-section.

The **linear shrinkage of castings** is approximately as follows:

Bar iron, rolled	1:55	Cast iron	1:96	Steel, puddled	1:72
Bell metal	1:65	Gun metal	1:134	Steel, wrought	1:64
Bismuth	1:265	Iron, fine grained	1:72	Tin	1:128
Brass	1:65	Lead	1:92	Zinc, cast	1:62
Bronze	1:63	Steel castings	1:50	8 Cu + 1Sn (by wt.)	1:134

Table 4. Volume, Density and Specific Heat of Water at Saturation Pressure

(From Marks and Davis's Steam Tables)

Temp., deg. fahr.	Pressure, lb. per sq. in.	Specific volume, cu. ft. per lb.	Density, lb. per cu. ft.	Specific heat	Temp., deg. fahr.	Pressure, lb. per sq. in.	Specific volume, cu. ft. per lb.	Density, lb. per cu. ft.	Specific heat
20	0.06	0.01603	62.37	1.0168	240	24.97	0.01692	59.11	1.012
30	0.08	0.01602	62.42	1.0098	250	29.82	0.01700	58.83	1.015
40	0.12	0.01602	62.43	1.0045	260	35.42	0.01708	58.55	1.018
50	0.18	0.01602	62.42	1.0012	270	41.85	0.01716	58.26	1.021
60	0.26	0.01603	62.37	0.9990	280	49.18	0.01725	57.96	1.023
70	0.36	0.01605	62.30	0.9977	290	57.55	0.01735	57.65	1.026
80	0.51	0.01607	62.22	0.9970	300	67.00	0.01744	57.33	1.029
90	0.70	0.01610	62.11	0.9967	310	77.67	0.01754	57.00	1.032
100	0.95	0.01613	62.00	0.9967	320	89.63	0.01765	56.66	1.035
110	1.27	0.01616	61.86	0.9970	330	103.0	0.01776	56.30	1.038
120	1.69	0.01620	61.71	0.9974	340	118.0	0.01788	55.94	1.041
130	2.22	0.01625	61.55	0.9979	350	135.0	0.01800	55.57	1.045
140	2.89	0.01629	61.38	0.9986	360	153.0	0.01812	55.18	1.048
150	3.71	0.01634	61.20	0.9994	370	173.0	0.01825	54.78	1.052
160	4.74	0.01639	61.00	1.0002	380	196.0	0.01839	54.36	1.056
170	5.99	0.01645	60.80	1.0010	390	220.0	0.01854	53.94	1.060
180	7.51	0.01651	60.58	1.0019	400	247.0	0.0187	53.5	1.064
190	9.34	0.01657	60.36	1.0029	410	276.0	0.0189	53.0	1.068
200	11.52	0.01663	60.12	1.0039	420	308.0	0.0190	52.6	1.072
210	14.13	0.01670	59.88	1.0050	430	343.0	0.0192	52.2	1.077
220	17.19	0.01677	59.63	1.007	440	381.0	0.0194	51.7	1.082
230	20.77	0.01684	59.37	1.009					

Specific Heat

Units of Heat. The mean **British thermal unit** (B.t.u.) is defined as the $\frac{1}{180}$ part of the heat required to raise the temperature of 1 lb. of water from 32 deg. to 212 deg. fahr. This is substantially equal to the heat required to raise 1 lb. of water from 63 deg. to 64 deg. fahr.

The **mean calorie** is $\frac{1}{100}$ of the heat required to raise the temperature of 1 g. of water from 0 deg. to 100 deg. cent. It is practically the same as the 17½-deg. calorie, that is the heat required to raise 1 g. of water from 17 deg. to 18 deg. cent. The 15-deg. calorie is also used extensively. Because of the variation of the heat capacity of water, this is slightly larger than the mean or 17½-deg. calorie. The present tendency is toward the mean calorie (and mean B.t.u.) as the standard heat unit.

In countries which have adopted the metric system, engineers employ the **kilogram calorie** (or "large calorie") as the unit in heat measurements. 1 kilogram calorie = 1000 g. calories = 3.968 B.t.u. (1 B.t.u. = 0.252 kilogram calorie). This is the "wärmeeinheit" (WE) mentioned in German texts.

Specific Heat. The specific heat of a substance is the ratio of the heat required to raise the temperature of unit weight of the substance 1 deg. to the heat required to raise the temperature of unit weight of water 1 deg. at some specified temperature. The specific heat is thus numerically equal to the quantity of heat required to raise the temperature of a unit weight of the substance by 1 deg.

Denoting by c the specific heat, the heat required to raise the temperature of M lb. of a substance from t_1 to t_2 is $Q = Mc(t_2 - t_1)$, provided c is a constant.

In general, c varies with the temperature, though for moderate temperature ranges a constant mean value may be taken. If, however, c is taken as variable, then $Q = M\int_{t_1}^{t_2} c\,dt$. The *mean* specific heat from 0 deg. to t deg. is given by $c_m = \frac{1}{t}\int_0^t c\,dt$. If $c = a_1 + a_2t + a_3t^2 + \ldots$,

$$c_m = a_1 + \tfrac{1}{2}a_2t + \tfrac{1}{3}a_3t^2 + \ldots$$

Specific Heat of Water. The specific heat of water between 32 deg. and 212 deg. fahr. has been investigated by Barnes, Ludin, Dieterici, and others (see Marks and Davis's "Steam Tables," p. 88). For this range of temperatures, two distinct values of the specific heat may be noted, according as (1) the pressure on the water is the saturation pressure corresponding to the temperature, or (2) constant atmospheric pressure. In Table 4 the values of the specific heat are according to the first system. For a full discussion of this topic see Marks and Davis's "Steam Tables," p. 90.

Table 5. Mean Specific Heats of Various Solids and Liquids Between 32 and 212 Deg. Fahr.

(For gases, see p. 201; for pure metals, p. 41)

SOLIDS		SOLIDS		SOLIDS	
Alloys:		Corundum	0.198	Talc	0.209
Bismuth-tin	0.040–0.045	Dolomite	0.222	Tufa	0.33
Bell metal	0.086	Ebonite	0.33	Vulcanite	0.331
Brass, yellow	0.0883	Glass:		Wood:	
Brass, red	0.090	Normal	0.199	Fir	0.65
Bronze	0.104	Crown	0.16	Oak	0.57
Constantan	0.098	Flint	0.12	Pine	0.67
D'Arcet's metal	0.050	Graphite	0.201	LIQUIDS	
German silver	0.095	Ice	0.504	Acetic acid	0.51
Lipowitz's metal	0.040	India rubber (Para)	0.27–0.48	Alcohol (absolute)	0.70
Nickel steel	0.109	Oxides:		Benzol	0.43
Rose's metal	0.050	Alumina (Al_2O_3)	0.183	Chloroform	0.23
Solders (Pb and Sn)	0.040–0.045	Cu_2O	0.111	Ether	0.503
Asbestos	0.20	Lead oxide (PbO)	0.055	Gasoline	0.70
Ashes	0.20	Lod stone	0.156	Glycerine	0.576
Borax	0.229	Magnesia	0.222	Hydrochloric acid	0.60
Brick	0.22	Magnetite (Fe_3O_4)	0.168	Ke osene	0.50
Carbon-coke	0.203	Silica	0.191	Naphthalene	0.31
Chalk	0.215	Soda	0.231	Machine oil	0.40
Charcoal	0.20	Zinc oxide (ZnO)	0.125	Mercury	0.033
Cinders	0.18	Paraffin wax	0.69	Olive oil	0.35
Coal	0.314	Salt, rock	0.21	Paraffin oil	0.52
Cork	0.485	Sand	0.195	Petroleum	0.498
		Serpentine	0.25	Sulphuric acid	0.336
		Sulphur	0.180	Sea water	0.94

Table 6. Mean Specific Heat of Iron (c_m) Between 32 and t Deg. Fahr. (Oberhoffer)

t	600	800	1000	1200	1400
c_m	0.127	0.133	0.139	0.148	0.167
t	1600	1800	2000	2250	2500
c_m	0.170	0.169	0.168	0.167	0.167

Table 7. Specific Heat of Salt (NaCl) Solutions

Parts of NaCl by weight in 100 parts of the solution	Temperature, deg. fahr.										
	0	15	32	50	60	70	80	90	100	120	140
0	.....	.....	1.006	1.002	1.000	0.999	0.998	0.999	1.000	1.001	1.003
2	.....	.....	0.966	0.969	0.971	0.973	0.976	0.978	0.980	0.983	0.986
4	.....	.....	0.944	0.947	0.949	0.951	0.954	0.956	0.958	0.961	0.964
6	.....	.....	0.923	0.927	0.929	0.930	0.932	0.934	0.936	0.939	0.942
8	.....	.....	0.904	0.907	0.909	0.910	0.912	0.914	0.916	0.919	0.922
10	.....	.....	0.885	0.889	0.891	0.892	0.894	0.896	0.899	0.901	0.903
12	.....	.....	0.869	0.872	0.874	0.875	0.877	0.879	0.881	0.883	0.885
14	.....	0.851	0.854	0.857	0.859	0.860	0.861	0.863	0.865	0.867	0.870
16	.....	0.838	0.840	0.843	0.844	0.845	0.847	0.849	0.851	0.853	0.855
18	.....	0.825	0.827	0.830	0.831	0.832	0.833	0.835	0.837	0.839	0.841
20	.....	0.812	0.814	0.816	0.817	0.818	0.820	0.822	0.823	0.824	0.827
22	0.795	0.798	0.801	0.804	0.805	0.806	0.807	0.809	0.810	0.812	0.814
24	.....	0.787	0.789	0.791	0.792	0.793	0.794	0.796	0.798	0.799	0.801
26	.....	0.776	0.778	0.780	0.781	0.781	0.782	0.784	0.785	0.787	0.788

The specific heat of salt solutions increases slowly with rising temperature. Table 7, due to Gröber, gives the specific heat of solutions of NaCl for various temperatures and concentrations. The values are probably more accurate than those given by Siebel in Table 13.

For **specific heat of gases** see table 28, p. 201.

Temperature of Mixtures

The temperature of a mixture of weight $M_1 + M_2$ lb. consisting of M_1 lb. of a substance at temperature t_1 and with specific heat c_1 and M_2 lb. of a substance at temperature t_2 with a specific heat c_2, is given by the equation

$$t_m = (M_1c_1t_1 + M_2c_2t_2)/(M_1c_1 + M_2c_2).$$ In general, $t_m = \Sigma Mct/\Sigma Mc$.

If it be required to raise the temperature of M_1 lb. of a substance having the specific heat c_1 from the temperature t_1 to t_m, the weight M_2 required of a second substance at temperature t_2 and having a specific heat c_2 is

$$M_2 = M_1c_1(t_m - t_1)/c_2(t_2 - t_m)$$

If M_1 lb. of a gas (corresponding to V_1 cu. ft.) at t deg. be mixed with M_2 lb. of the same gas (corresponding to V_2 cu. ft.) the pressure being constant, then the temperature of the mixture is $t_m = (M_1t_1 + M_2t_2)/(M_1 + M_2) = [(V_1 + V_2)/[(V_1/T_1) + (V_2/T_2)]] - 459.6$, in which T_1 and T_2 are the absolute temperatures.

Changes of State Effected by Heat

Table 8. Freezing Mixtures

(The lowest temperature that may be produced by a freezing mixture is the freezing point of the solution)

Mixtures	Composition, parts by weight	Reduction in temperature, deg. fahr. from	to
Sodium phosphate..	9	34.3	21.2
Sal ammoniac.....	6		
Diluted nitric acid.	4		
Sodium sulphate...	6	50.0	−13.0
Ammonium nitrate.	5		
Diluted nitric acid.	4		
Sal ammoniac.....	5	50.0	10.4
Saltpeter.....	5		
Water.....	16		
Sodium carbonate.	1	50.0	6.8
Ammonium nitrate.	1		
Water.....	1		
Sodium sulphate...	3	50.0	−2.2
Diluted nitric acid.	2		
Sodium sulphate...	6	50.0	−9.4
Sal ammoniac.....	4		
Saltpeter.....	2		
Nitric acid.....	4		
Ammonium nitrate.	1	50.0	3.2
Water.....	1		
Sodium phosphate..	9	59.0	15.8
Diluted nitric acid.	4		
Potassium hydrate.	4	32.0	−38.3
Snow.....	3		
Sulphuric acid.....	1	−2.2	−40.0
Nitric acid.....	1		
Snow.....	2		
Sodium chloride	1	32.0	− 1.6
Snow.....	3		
Saltpeter.....	1	46.4	−11.2
Sal ammoniac....	1		
Water.....	1		
Diluted nitric acid.	1	57.2	−31.0
Snow.....	1		
Sodium sulphate..	8	50.0	− 0.4
Hydrochloric acid.	5		
Sodium sulphate..	8	50.0	5.0
Saltpeter.....	5		
Sal ammoniac....	5		
Water.....	16		
Sodium sulphate..	5	50.0	3.2
Diluted nitric acid.....	4		
Sodium chloride..	1	32.0	−0.4
Snow.....	1		
Diluted sulphuric acid.	1	23.0	−41.8
Snow.....	1		
Calcium chloride.	3	32.0	−27.4
Snow.....	2		
Calcium chloride..	2	32.0	−43.6
Snow.....	1		

Table 9. Melting Points of Various Solids, Deg. Fahr.

(For pure metals, see p. 41)

Alloys:	
Bismuth solder..	200–262
Brass and bronze (about).....	1650
80 Cu + 20 Zn....	1845
80 Cu + 50 Zn....	1615
20 Cu + 80 Zn....	1300
Delta metal.....	1742
20 Sn + 80 Pb....	550
50 Sn + 50 Pb....	400
80 Sn + 20 Pb....	388
Fusible alloys:	
33 Bi + 33 Pb + 33 Sn.	250
18 Bi + 36 Pb + 46 Sn	305
10 Bi + 40 Pb + 50 Sn	324
Tin solder.....	275–350
Blast-furnace slag.	2370–2600
Borax.....	1040
Cast iron, gray.....	2200
Cast iron, white.....	2070
Chlorides:	
Calcium.....	1330
Potassium.....	1350
Sodium.....	1422
Enamel colors.....	1760.0
India rubber.....	257.0
Paraffin.....	129.2
Phosphorus.....	111.2
Porcelain.....	2820.0
Potassium.....	143.6
Sodium.....	204.8
Spermaceti.....	120.2
Stearine.....	122.0
Steel.....	2370–2550
Wrought iron.....	2460–2640

Table 10. Melting Points of Refractories

C. W. Kanolt, U. S. Bureau of Standards

Refractory	No. of samples tested	Melting point, deg. fahr.*	Refractory	No. of samples tested	Melting point, deg. fahr.*
Fire-clay brick	41	2830-3140	Pure alumina	1	3720
Bauxite brick	8	2850-3245	Chromite brick	1	3930
Silica brick	3	3090-3100	Magnesia brick	1	3930
Kaolin	3	3150-3160	Chromite	1	3960
Bauxite	1	3260	Pure lime	1	4660
Bauxite clay	1	3310	Pure magnesia	1	5070

* Not reliable to better than 10 deg. fahr.

The melting point of cristobalite, the stable form of silica is given as 1710 deg. ± 10 deg. cent. by J. B. Ferguson and H. E. Merwin in the Journal of Science, Vol. 46, August, 1918.

Table 11. Freezing Points of Liquids at Atmospheric Pressure

(Deg. fahr.)

Alcohol (absolute)	−148.0	Calcium chloride (sat. sol.)	− 40	Rape-seed oil	25.7
Ammonia	−108.4	Ether	−180	Turpentine	14.0
Aniline	21.2	Glycerine	− 4	Sulphuric acid	−105
Benzol	41.0	Naphthalene	176	Salt (NaCl) sol., sat.	− 0.4
Carbon bisulphide	−171.0	Linseed oil	− 4	Seawater	27.5
Carbon dioxide	−110.2			Toluene	−148
Chloroform	− 83.0				

Mixtures of glycerine and water (Bolley)			Mixtures of alcohol and water (F. Beilstein)			
Per cent by weight of glycerine	Specific gravity	Freezing point, deg. fahr.	Per cent by weight of alcohol	Freezing point, deg. fahr.	Per cent by weight of alcohol	Freezing point, deg. fahr.
10	1.0245	30.2	2.58	30.2	21.7	10.4
20	1.0498	27.5	5.22	28.4	23.8	6.8
30	1.0771	20.8	7.36	26.6	26.0	3.2
40	1.1045	1.0	9.58	24.8	28.0	− 0.4
45	1.1183	−15.2	11.50	23.0	30.0	− 4.0
50	1.1320	−25.6	13.27	21.2	33.5	−11.2
60	1.1582	Below −31.0	16.53	17.6	37.3	−18.4
			19.09	14.0	41.2	−25.6

Table 12. Freezing Point, Density, and Specific Heat of Calcium Chloride Solutions

(U. S. Bureau of Standards)

Per cent of $CaCl_2$ by weight	Specific Gravity at 39 deg. fahr.	Freezing point, deg. fahr.	Specific heat at 32 deg. fahr.	Per cent of $CaCl_2$ by weight	Specific Gravity at 39 deg. fahr.	Freezing point, deg. fahr.	Specific heat at 32 deg. fahr.
14.88	1.12	15.8	0.799	23.03	1.20	−11.2	0.714
16.97	1.14	8.6	0.775	24.89	1.22	−20.2	0.690
19.07	1.16	3.2	0.753	26.77	1.24	−29.2	0.679
21.13	1.18	− 4.0	0.733	28.55	1.26	−40.0	0.668

Table 13. Freezing Point, Density, and Specific Heat of Sodium Chloride Solutions

(From Siebel's "Compend of Mechanical Refrigeration")

Per cent of NaCl by weight	Specific Gravity at 39 deg. fahr.	Freezing point, deg. fahr.	Specific heat at 32 deg. fahr.	Per cent of NaCl by weight	Specific Gravity at 39 deg. fahr.	Freezing point, deg. fahr.	Specific heat at 32 deg. fahr.
1.0	1.007	30.5	0.992	8	1.061	21.2	0.919
2.0	1.015	29.3	0.984	9	1.068	19.9	0.905
2.5	1.019	28.6	0.980	10	1.076	18.7	0.892
3.0	1.023	27.8	0.976	12	1.091	16.0	0.874
3.5	1.026	27.1	0.972	15	1.115	12.2	0.855
4.0	1.030	26.6	0.968	20	1.155	6.1	0.829
5.0	1.037	25.2	0.960	24	1.187	1.2	0.795
6.0	1.045	23.9	0.946	25	1.196	0.5	0.783
7.0	1.053	22.5	0.932	26	1.204	−1.1	0.771

Table 14. Boiling Points (Deg. Fahr.) at Atmospheric Pressure

(See also p. 208)

Zinc	1680	Glycerine	554	Turpentine	320.0
Sulphur	823	Phosphorus	554	Toluene	230.0
Mercury	675	Naphthalene	424	Sodium chloride (sat. sol.)	226.4
Linseed oil	538	Aniline	363	Helium	−450.0
Paraffin	572	Calcium chloride (sat. sol.)	356		

Heat of Fusion. The heat of fusion of a solid is the heat required in B.t.u. to convert 1 lb. of the substance from the solid to the liquid state, without change of temperature.

Table 15. Heat of Fusion, B.t.u. per Lb.

Aluminum	138.0	Sodium	57.1	Lipowitz's metal	12.3
Bismuth	22.8	Tin	25.0	Rose's metal	12.3
Blast-furnace slag	90.0	Zinc	50.6	Wood's metal	14.0
Cadmium	24.6	Alloys:		Zinc and bismuth	20–23
Copper	77.4	30.5 Pb + 69.5 Sn.	21.0	Benzol (C_6H_6)	55.0
Iron, gray cast	40.0	36.9 Pb + 61.3 Sn.	28.0	Calcium chloride sol. ($CaCl_2 + 6H_2O$)	72.5
Iron, white	60.0	63.7 Pb + 36.3 Sn.	21.0	Glycerine	76.5
Lead	10.6	77.8 Pb + 22.2 Sn.	17.0	Ice	144.0
Mercury	5.08	78.4 Sn + 21.6 Zn.	42.3	Naphthalene ($C_{10}H_8$)	64.1
Nickel	8.35	93.56 Sn + 6.44 Zn.	31.8	Phosphorus	9.0
Palladium	65.0	97.32 Sn + 2.68 Zn.	27.2	Phenol (C_6H_6O)	45.0
Platinum	49.0	Brittania metal (9 Sn + 1 Pb)	13.7	Sulphur	16.9
Potassium	28.09	D'Arcet's metal	10.4		
Silver	36.2				

Table 16. Latent Heat of Vaporization at Atmospheric Pressure, B.t.u. per Lb.

Alcohol	385	Methyl chloride	175	Hydrogen	222.
Aniline	198	Mercury	122	Nitrogen	81 5
Benzol	169	Sulphur	650	Oxygen	92
Chlorine	112	Carbon bisulphide	152.5	Water	970.4
Chloroform	110	Turpentine	126		
Ether	162	Acetone	233		

Table 17. Solution of Gases in Water

t (deg. fahr.) =	32	68	212	t (deg. fahr.) =	32	68	212
Air	0.032	0.020	0.012	Hydrogen	0.023	0.020	0.018
Acetylene	1.89	1.12	.....	Hydrogen sulphide	5.0	2.8	0.87
Ammonia	1250	700	.....	Hydrochloric acid	560	480	.....
Carbon dioxide	1.87	0.96	0.26	Nitrogen	0.026	0.017	0.0105
Carbon monoxide	0.039	0.025	.....	Oxygen	0.053	0.034	0.0185
Chlorine	5.0	2.5	0.00	Sulphuric acid	87	43	.....

Table 18. Solubility of Ammonia in Water (H. Mollier)

One pound of water at the given pressures and temperatures absorbs the following weights of ammonia:

Pressure, lb. per sq. in.	Temperatures, deg. fahr.													
	32	60	80	100	120	140	160	180	200	220	240	260	280	300
2	0.29	.154	.086	0.040	.002	.....	.....	.....	.....	.....	.....	.....	.....	.....
5	0.47	.296	.200	0.128	.071	0.028	.....	.....	.....	.....	.....	.....	.....	.....
10	0.70	.462	.327	0.243	.164	0.102	.056	.016	.....	.....	.....	.....	.....	.....
15	0.90	.607	.426	0.334	.223	0.168	.106	.057	.020	.....	.....	.....	.....	.....
20	1.15	.726	.539	0.423	.302	0.215	.151	.093	.048	.010	.....	.....	.....	.....
30	1.71	.920	.698	0.539	.408	0.310	.228	.158	.100	.050	.014	.....	.....	.....
40	2.31	.....	.862	0.663	.511	0.398	.302	.215	.162	.090	.046	.007	.....	.....
50	.....	.....	.992	0.778	.604	0.465	.358	.271	.195	.128	.079	.035	.001	.....
60	.....	.....	.....	0.890	.686	0.528	.410	.314	.235	.162	.126	.058	.021	.....
70	.....	.....	.....	1.003	.776	0.606	.470	.363	.275	.197	.135	.080	.040	.003
80	.....	.....	.....	.....	.894	0.660	.512	.402	.309	.227	.162	.106	.062	.021
100	.....	.....	.....	.....	.....	0.794	.620	.486	.381	.288	.213	.150	.099	.053
120	.....	.....	.....	.....	.....	0.926	.727	.563	.427	.323	.262	.188	.132	.081
140	.....	.....	.....	.....	.....	1.063	.822	.636	.509	.394	.304	.224	.163	.111

Heat of Vaporization (Table 16). The latent heat of vaporization is the quantity of heat in B.t.u. required to convert 1 lb. of a liquid into vapor at the same temperature under a constant external pressure. The latent heat depends upon the temperature at which the process takes place.

For the **latent heats of steam** at different pressures, see p. 209; and for the **latent heats of refrigerating media**, see pp. 218, 221 and 222.

One cubic foot of water at atmospheric pressure and at the temperature t will dissolve the volumes of gas (in cu. ft. at 32 deg. fahr. and at atmospheric pressure) given in Table 17.

TRANSMISSION OF HEAT

Preliminary Statements. There are three methods by which heat may be propagated or conveyed from one place to another.

1. By **Conduction.** In this method heat passes from one body to another or from one part of a body to another by contact. It is assumed that the warmer molecules impart heat to the colder ones.

2. By **Convection.** Heat is carried from place to place by the medium with which it is associated. Thus, in a hot-water heating system, heat is carried from the furnace through the building by the water in the system; in a boiler, heat is carried through the mass of water by the currents produced by circulation.

3. By **Radiation.** A source of heat, as the sun or a fire, gives off energy in the form of radiant heat. This energy is assumed to be propagated in all directions as a wave motion in the ether, and radiation falling upon a body is to some extent absorbed by it. According to the modern **theory of exchanges**, all bodies whose molecules are in vibration are sources of radiation. If two bodies, one hotter than the other, are placed within an enclosure, there is a continual interchange of energy between them. The hotter body A radiates more energy than it absorbs, the colder body B absorbs more than it radiates. The result is an equalization of temperature, but even after the equilibrium of temperature the process of radiation continues, each body radiating and absorbing energy.

Transmission of Heat by Convection and Conduction

Phenomena of Heat Transmission. In the cases of heat transmission that usually occur in practice—in boilers, condensers, the cooling of engine cylinders, etc.—heat is transmitted from one fluid to another through a wall separating the two. The character of the process is shown in Fig. 1. At the surface of the plate there is a film of the hot fluid of indefinite thickness. This film offers a considerable resistance to the transmission, as shown by the temperature difference $t_1 - t'_1$ through it. A corresponding film on the other side of the plate offers resistance measured by the temperature drop $t'_2 - t_2$. Let Q denote the total heat in B.t.u. transmitted in z hours through a plate area of A sq. ft. Then, for film f_1, $Q = k_1Az\ (t_1 - t'_1)$; for the plate, $Q = (K/b)\ Az\ (t'_1 - t'_2)$; for the film f_2, $Q = k_2Az\ (t'_2 - t_2)$.

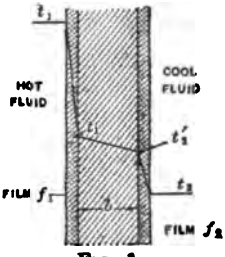


FIG. 1.

The coefficients k_1 and k_2 are the **conductances** of the films f_1 and f_2, respectively, K is the thermal conductivity of the plate and, analogously, (K/b) is the conductance of the plate. The reciprocals of these, viz., $1/k_1$, $1/k_2$, b/K, are the **resistances** of the two films and plate, respectively. The total resistance R to the flow is the sum of the separate resistances, $R = (1/k_1) + (1/k_2) + (b/K)$, and the conductance of the two films and plate is $k_0 = 1/R$. Hence, $Q = k_0Az\ (t_1 - t_2)$.

If the wall is composed of several layers of different materials having thicknesses b', b'', b''', etc., and thermal conductivities K', K'', K''', etc., the total resistance is $R = (1/k_1) + (1/k_2) + (b'/K') + (b''/K'') + (b'''/K''') + \ldots$, and $k_0 = 1/R$.

The **resistance of the fluid film** depends upon the kind of fluid and the character of the motion of the fluid along the wall. The resistance of a film of hot gas is very high. It has been estimated that in certain cases 98 per cent of the available temperature drop is required to force the heat through the gas film. If the hot fluid is saturated steam, condensation insures a film of water in contact with the plate and the resistance is relatively small compared with that of a gas film. The film resistance is considerably influenced by the motion of the fluids along the plates. If the hot gas, as in a boiler tube, is given a high velocity the resulting sweeping action partially destroys the film and decreases the resistance. The same is true on the water side of the wall. It is well established that the **rate of heat transmission** is **dependent on the velocity of the fluid** along the plate.

A layer of soot or scale on a plate has the effect of making a composite wall. The resistance is increased by b'/K', where b' is the thickness and K' the thermal conductivity of the layer. If k_0 is the total conductance for the clean plate, the ratio of the heat transmitted to that transmitted through the clean plate is

$$Q'/Q = \frac{1}{k_0} \Big/ \left(\frac{1}{k_0} + \frac{b'}{K'}\right) = 1 \Big/ \left(1 + \frac{b'k_0}{K'}\right)$$

The experiments of Clement and Garland (*Bulletin* No. 40, Engineering Experiment Station, University of Illinois) afford valuable data on the **transmission of heat from steam to water.** A stream of water was made to flow at various velocities through a steel tube of 0.985 in. inside diam. and with walls 0.134 in. thick. The tube was surrounded by steam, and the tem-

perature of the outer wall of the tube was measured by thermo-couples. The temperature of the inner wall was calculated from the known conductivity of the metal composing the tube. The results of one series of tests are shown in Table 19.

Table 19. Transmission of Heat from Steam to Water through Steel Tubes

No. of test	Temperatures, deg. fahr.				Velocity of water, ft. per sec.	B.t.u. transmitted per min. per sq. ft.	Conductances, B.t.u. per sq. ft. per sec.		
	of steam	of outside wall	of inside wall	of water, mean			Steam film	Water film	Both films and wall
1	330.2	220.0	165	69.0	17.13	3995	0.606	0.694	0.255
2	330.0	229.4	177	70.8	14.05	3756	0.621	0.591	0.246
3	330.0	233.1	185	74.6	10.39	3455	0.594	0.524	0.225
4	330.2	241.8	197	78.07	8.06	3260	0.614	0.457	0.215
5	330.2	257.4	220	89.5	4.25	2672	0.611	0.340	0.185
6	330.2	267.1	234	110.2	2.31	2370	0.625	0.318	0.179

It appears from these results that the velocity of flow has a marked effect on the conductance of the water film and upon the rate of transmission from the steam to the water.

Conductance of Fluids. As stated in a preceding paragraph, the rate of transmission of heat between a fluid and a metal surface depends upon the conductance of the fluid film. If h denotes the difference of temperature between the fluid and the plate wall, and k the conductance, then $Q = ksAh$. The value of k depends upon the nature of the fluid and also upon the velocity of the fluid along the surface. The following values may be taken for the conditions stated (k = B.t.u. per sq. ft. per hour per deg. fahr. difference of temperature):

1. For boiling water ..... k = 800 to 1200
2. For condensing steam. ..... k = 2000

The experiments of Clement and Garland show a mean value of 0.61 × 3600 = 2200, approx.

3. Water, not boiling:

If the water is at rest, ... k = 100.

If the water is in motion, k will depend upon the velocity. In the experiments of Clement and Garland k varied from 730 with a velocity of 1.45 ft. per sec. to 2500 with a velocity of 17.13 ft. per sec.

4. Air at Rest. Take h as the difference between the temperature of the surface and the mean temperature of the air in the room. Then, for vertical surfaces,

$k = 0.62 + 0.009h$ for $h < 20$ deg, and $k = 0.385\sqrt[4]{h}$ for $h > 20$ deg., or

for h =	0	5	10	25	50	100	200	300	400
k =	0.62	0.665	0.71	0.86	1.04	1.22	1.45	1.60	1.72

5. Transmission of Heat from the Surface of a Rotating Cylinder to Air, as in flywheels, armatures, etc. According to the experiments of Hinlein, for

Smooth Surfaces (polished copper).

Velocity w, ft. per sec.	0	20	40	60	80	100
Conductance k	0.47	1.97	2.79	3.14	3.30	3.38

Rough surfaces (dull black, varnished).

w =	0	20	40	60	80	100
k =	0.47	2.49	3.32	3.95	4.50	5.10

Table 20 Thermal Conductivities of Metals

Substance	Temp., deg. fahr.	K	Substance	Temp., deg. fahr.	K
Metals:			Magnesium	32–212	92.0
Aluminum	64	116.0	Mercury	32	3.6
Aluminum	212	119.0	Mercury	122	4.6
Antimony	32	10.6	Nickel	64	34.4
Antimony	212	9.7	Nickel	212	33.4
Bismuth	64	4.7	Platinum	64	40.2
Bismuth	212	3.9	Platinum	212	41.9
Cadmium	64	53.7	Silver	64	244.0
Cadmium	212	52.2	Silver	212	240.0
Copper	64	222.0	Tin	64	37.6
Copper	212	220.0	Tin	212	35.0
Gold	64	169.0	Zinc	64	64.1
Gold	212	170.0	Zinc	212	63.5
Iron, pure	64	39.0	Alloys:		
Iron, pure	212	36.6	Brass	63	63.0
Iron, wrought	64	34.9	Constantan (60 Cu, 40 Ni)	64	13.1
Iron, wrought	212	34.6	Constantan (60 Cu, 40 Ni)	212	15.5
Iron, cast	129	27.6	German silver	32	16.9
Iron, cast	216	26.8	German silver	212	21.5
Steel (1 per cent C)	64	26.2	Manganin (84 Cu, 4 Ni, 12 Mn)	64	12.8
Steel (1 per cent C)	212	25.9		212	15.2
Lead	64	20.1	Platinoid	64	14.5
Lead	212	19.8			

Table 21. Conductivities of Various Insulating Materials

(Poensgen, *Zeit. Ver. Deutsch. Ing.*, Oct. 12, 1912)

(See also Tables 22 and 23)

Materials	Wt., lb. per cu. ft.	K at 68 deg. fahr.	Materials	Wt., lb. per cu. ft.	K at 68 deg. fahr.
Cork plate No. 1	3.8	0.024	Burnt infusorial earth No. 2	20.8	0.046
Cork plate No. 5	11.2	0.028	Burnt infusorial earth No. 3	22.8	0.044
Cork plate No. 10	13.4	0.031	Burnt infusorial earth No. 4	28.2	0.050
Cork plate No. 14	16.6	0.034	Cork linoleum	33.4	0.046
Cork plate No. 16	21.8	0.038	Cork stone 2.6 in. thick with 0.2 in. of cement on each side	27.8	0.040
Charcoal, bound in plates	12.7	0.032			
Burnt infusorial earth No. 1	18.5	0.039			

Thermal Conductivity. The resistance of the plate to heat transmission is b/K, the conductance K/b. The number K in this expression is called the **thermal conductivity** of the material of the plate. The thermal conductivity of a substance may be defined as the quantity of heat (B.t.u.) that flows in a unit of time (1 hr.) through unit area of plate (1 sq. ft.) of unit thickness (1 ft.) having unit difference of temperature (1 deg. fahr.) between its faces. This definition follows from the relation

$$Q = KAs(t'_1 - t'_2)/b, \text{ or } K = Qb/As(t'_1 - t'_2).$$

It will be observed that the conductivity K involves an element of length, while the conductance k does not.

The thermal conductivity of different materials varies greatly. For metals and alloys K is high, while for certain insulating materials, as asbestos, cork, and silk, K is very low. In general, K **varies with the temperature.** In the case of metals K usually decreases with rising temperature, while for most other substances the reverse is true. Tables 20 to 24, collected from various sources, give values of thermal conductivity.

Table 22. Thermal Conductivities of Miscellaneous Solid Substances

(The values of K in the table below are to be regarded as rough average values for the temperature range indicated)

Substance	Temp. range, deg. fahr.	K	Substance	Temp. range, deg. fahr.	K
Cardboard		0.120	Glass:		
Cement		0.170	Crown, window		0.6
Cotton wool		0.010	Flint, window		0.48
Ebonite		0.100	Jena, window		0.25–0.5
Felt		0.022	Soda, window		0.3–0.44
Gas carbon		2.400	Firebrick	32–2400	0.75
Graphite		2.900	Alumina brick	32–1300	0.5
Marble, white		1.720	Clinker, granular	32–1300	0.27
Mica		0.440	Pumice stone	70–150	0.135
Paper		0.075	Carborundum, coarse	70–212	0.122
Paraffin wax		0.145	Carborundum, fine	70–212	0.121
Porcelain		0.600	Magnesia, fused	70–212	0.114
Quarts:			Magnesia, powder		0.04
Parallel to axis		7.250	Magnesia, calcined	70–212	0.109
Perpendicular to axis		3.900	Coke, powdered	32–212	0.106
Rubber, Para		0.109	Graphite, powdered	70–212	0.097
Sand		0.031	Coarse brickdust	32–212	0.094
Sawdust	70–275	0.037	Silicate enamel	70–208	0.097
Silicate cotton		0.046	Fused quarts	70–212	0.094
Slate		1.140	Infusorial earth	32–1200	0.092
Blast-furnace slag	75–260	0.064	Serpentine (Cornwall)		1.07
Cork stone (asphalted)	50–135	0.041	Strawboard		0.80
Lime	70–212	0.070	Vulcanite		0.21
Firebrick, powdered	70–212	0.068	Beeswax		0.022
Chalk		0.480	Haircloth		0.01
Charcoal, powdered	32–212	0.053	Leather (cowhide)		0.10
Wood ashes	32–212	0.041	Linen		0.05
Lava		0.019	Mill shavings		0.05
Granite		1.300	Hair felt		0.03
Magnesia bricks	32–2400	1.5	Mineral wool		0.035

For steam-pipe coverings, see also pp. 1348 to 1359.

Table 23. Conductivities of Insulating Materials (Nusselt)

Material	Weight, lb. per cu. ft.	Temperatures, deg. fahr.						
		32	100	200	300	400	600	800
Asbestos	36.0	0.087	0.097	0.110	0.117	0.121	0.125	0.130
Burnt infusorial earth for pipe coverings	12.5	0.043	0.046	0.052	0.057	0.062	0.073	0.085
Insulating composition (loose)	25.0	0.040	0.046	0.050	0.053	0.055	.....	.....
Same mixed with water and dried	42.5	.....	.....	.....	0.067	0.081 (at 428°)	.....	.....
Cotton	5.0	0.032	0.035	0.039	.....	.....	.....	.....
Silk hair	9.1	0.026	0.030	0.034	.....	.....	.....	.....
Silk	6.3	0.025	0.028	0.034	.....	.....	.....	.....
Wool	8.5	0.022	0.027	0.033	.....	.....	.....	.....
Pulverised cork	10.0	0.021	0.026	0.032	.....	.....	.....	.....
Infusorial earth (loose)	22.0	0.035	0.039	0.045	0.047	0.050	0.053	.....
Same mixed with water and dried	36.0	.....	.....	.....	0.056	.....	.....	(at 662°) 0.083

Table 24. Conductivities of Liquids and Gases

Substance	Temp., deg. fahr.	K	Substance	Temp., deg. fahr.	K
Alcohol	77	0.104	Ammonia	32.0	0.0111
Aniline	54	0.099	Ammonia	212.0	0.0172
Glycerine	77	0.165	Carbon monoxide	32.0	0.0121
Benzol	41	0.081	Carbon monoxide	44.6	0.0123
Ether	48–59	0.073	Carbon dioxide	32.0	0.0079
Oil, olive	.....	0.096	Carbon dioxide	212.0	0.0122
Oil, castor	.....	0.103	Ethylene	32.0	0.0096
Oil, paraffin	63	0.085	Helium	32.0	0.0082
Oil, petroleum	55	0.086	Hydrogen	32.0	0.0775
Oil, turpentine	55	0.079	Hydrogen	212.0	0.0895
Vaseline	77	0.106	Methane	46.0	0.0156
Water	63	0.032	Nitrogen	45.0	0.0127
Water	52	0.036	Nitrous oxide	32.0	0.0085
Water	77	0.033	Nitrous oxide	212.0	0.0122
Air	32	0.0126	Nitric oxide	46.0	0.0101
Argon	32	0.0094	Oxygen	45.0	0.0136

The conductivities of air and steam are given as functions of the temperature by the relations: K (for air) $= 0.01221\ (1 + 0.00132\ t)$ and K (for steam) $= 0.00882\ (1 + 0.00219\ t)$.

For t (deg. fahr.) =	32	50	100	200	300
K (for air) =	0.0127	0.0130	0.0138	0.0154	0.0170
K (for steam) =	0.0095	0.0098	0.0107	0.0127	0.0146
For t (deg. fahr.) =	400	500	600	700	800
K (for air) =	0.0187	0.0203	0.0219	0.0235	0.0250
K (for steam) =	0.0165	0.0184	0.0204	0.0223	0.0242

Heat Transmission Between Fluids Separated by a Plate. The following are important examples of heat transmission through plates or tubes. (Lucke's "Engineering Thermodynamics," p. 550):

Fluid giving up heat	Fluid receiving heat	Examples of heat transmission
1. Liquid	(a) Liquid	Liquid heat exchangers, etc.
	(b) Gas	Hot-water radiators; cooling-tower surfaces.
	(c) Boiling liquid	Brine coolers; hot-liquid evaporators.
2. Gas	(a) Liquid	Brine coolers in cold-storage rooms; air coolers with water or brine coils; economizers.
	(b) Gas	Steam superheaters; air-cooled motors.
	(c) Boiling liquid	Steam boilers; direct-expansion ammonia coils.
3. Condensing vapor	(a) Liquid	Condensers; feed-water heaters.
	(b) Gas	Steam radiators.
	(c) Boiling liquid	Vacuum evaporators.

Let Q = heat transmitted per hour, k_0 = coefficient of heat transfer = conductance of wall, with scale, soot, and fluid films, A = area in sq. ft. through which transfer is in process, and h_m = mean temperature difference for the process. Then $Q = k_0 A h_m$.

If one of the fluids is condensing or boiling under constant pressure the temperature remains constant during the heat transfer. In case 3(c) both fluids remain at constant temperature and h_m is simply the difference. In all other cases the temperature of one fluid or of both fluids changes during the process. If h_1 is the initial temperature difference and h_2 the final temperature difference, then the mean temperature difference for the process is given by $h_m = (h_1 - h_2)/\log_e(h_1/h_2)$.

Coefficient of Heat Transmission

The value of the coefficient k_0 depends on a number of conditions, viz.: the character of the fluids, the velocity of the fluids along the separating surfaces, the condition of the surface, and the shape of the surface. In general, k_0 is small when one of the fluids is a gas. The following rough values are given by Lucke for the cases stated in the preceding paragraph:

Case	Fluids	Values of k_0 (B.t.u. per hour per sq. ft. per deg. fahr.)
1 (*a*)	Liquid—Liquid	50–75
1 (*b*)	Liquid—Gas	2–6
1 (*c*)	Liquid—Boiling liquid	10–100
2	Gas—Liquid, gas, or boiling liquid	2–5
3 (*a*)	Condensing vapor—Liquid	150–350, 1000 under special conditions
3 (*b*)	Condensing vapor—Gas	2–4
3 (*c*)	Condensing vapor—Boiling liquid	400–600

For some of the most important cases the variation of k_0 with external conditions has been investigated. The following is a résumé of some of the results.

1. Transmission Between Steam and Water (Condensers, etc.). According to the exhaustive experiments of Orrok (*Trans. A. S. M. E.*, vol. 32) the value of k_0 depends upon the following factors:

(*a*) k_0 is approximately proportional to the square root of the velocity of the cooling water.

(*b*) k_0 is inversely proportional to the eighth root of the mean difference of steam and water temperatures.

(*c*) k_0 depends upon the material of the tube and upon the cleanliness of the tube.

(*d*) k_0 is reduced in a marked degree by the presence of air in the condenser. Thus if p_s denotes the pressure of the steam alone and p_t the total pressure ($= p_s +$ pressure of air), then k_0 varies as $(p_s/p_t)^5$.

Orrok (1914) gives the following equation as representing the results of **tests on condensers** under various conditions: $k_0 = 350\, c_1 c_2 (p_s/p_t)^2 \times \sqrt{w}$, in which c_1 ($= 0.5$ to 1.0) is the cleanliness coefficient, and c_2 a coefficient depending on the material of the tubes. For copper, $c_2 = 1.0$; for Admiralty mixture, 0.98 (oxidised, 0.97); for Munts metal, 0.95; for aluminum bronze, 0.92. The exponent for (p_s/p_t) is now taken as 2 instead of 5. w denotes the velocity of the water in ft. per sec.

The variation of k_0 with the water velocity has received attention from a number of experimenters. The following are some of the relations obtained (*Trans. A. S. M. E.*, vol. 32):

Ser, $k_0 = 520 \sqrt[3]{w}$; Hagemann, $282 \sqrt{w}$; Josse, $487 \sqrt[3]{w}$; Allen, $220 \sqrt{w}$; Stanton, $340 \sqrt[3]{w}$; Orrok, $308 \sqrt{w}$; Clement and Garland, $270 \sqrt[3]{w}$.

Carrier (*Trans. A. S. M. E.*, vol. 33), gives the following formula as representing the results of tests on condensers: $k_0 = 1/[0.000394 + (0.00255/w)]$.

2. Steam and Boiling Liquids. Where **steam coils are used for evaporation** of water or other liquids the following coefficients of transmission may be used (Hausbrand, "Evaporating, Condensing and Cooling Apparatus"):

Let d = diameter of pipe and l = length, both in feet; then $k_0 = 1250/\sqrt{dl}$. This value holds for copper pipe, when the boiling liquid is water. For wrought-iron pipe take 0.75 and for cast-iron pipe 0.5 of this value. In practice, ¾ of these values should be assumed.

In the case of thick, viscous liquids, the size of pipe has little influence on the rate of transmission, and the following values of k_o may be taken: For long heating coils, 130–150; for short heating coils, 160–180; for thin steam pipes, 200.

3. Steam to Air. Carrier (*Trans. A. S. M. E.*, vol. 33) deduces the following formula for the coefficient of transmission under the conditions that obtain in **hot-blast heating** systems: $k_o = 1/[a + (b/w)]$, in which w denotes the velocity of the air, ft. per min., and a and b are constants depending on the type of heater. Tests on a Buffalo standard heater with 50 lb. steam pressure, gave $a = 0.0447$, $b = 50.66$. For Vento cast-iron heaters the test showed $a = 0.047$, $b = 61.0$.

Conduction of Heat through Steam Pipes.—Pipe Coverings. The coefficient of heat transmission through bare steam pipes varies with the steam pressure. Assuming the air to be still, k_o may be taken as about 2.1 for low steam pressure (5 or 6 lb. gage), and 3.0 or 3.1 for a pressure of 90 or 100 lb. gage. Table 25 gives the results of tests by the Johns-Manville Co. on heat conduction through bare pipe and pipe with various insulating coverings. Outside temperature, 72 deg. fahr.

Table 25. Heat Conduction through Bare and Covered Steam Pipes

(Johns-Manville Co., New York)

Nature and thickness of covering	Conductance, B.t.u. per sq. ft. per hr. per deg. fahr.	Heat transmitted in B.t.u. per sq. ft. per hour at steam pressures (per sq. in.) of		
		30 lb.	25 lb.	50 lb.
Low-pressure Steam Pipe				
Asbestocel, 1 in.	0.564	77.4	109.8	127.2
Air cell, 1 in.	0.798	111.6	156.0	179.4
Asbestos, molded, 1 in.	0.696	97.2	135.6	156.6
Asbestos, indent, 1 in.	0.684	96.0	133.2	154.2
Bare pipe	3.000	360.0	576.0	876.0

		100 lb.	150 lb.	200 lb.	250 lb.
High-pressure Steam Pipe					
Asbestos:					
Sponge-felted, 1 in.	0.4524	120.6	133.8	142.8	151.8
Sponge-felted, 2 in.	0.354	94.2	104.4	111.6	118.2
Sponge-felted, 3 in.	0.324	85.2	95.4	102.6	108.6
Magnesia, J. M., 1 in.	0.498	132.6	147.0	156.0	166.2
Magnesia, J. M., 1½ in.	0.450	119.4	132.6	142.2	150.0
Magnesia, J. M., 2 in.	0.378	100.2	111.6	119.4	126.6
Bare pipe	3.000	1170.0	1284.0	1360.0	.....

		At 500 deg. fahr.	At 600 deg. fahr.
Superheated Steam			
Asbestos:			
Sponge-felted, 1 in.	0.452	193.2	238.8
Sponge-felted, 2 in.	0.354	151.2	187.2
Sponge-felted, 3 in.	0.324	138.6	171.0
Magnesia, J. M., 1 in.	0.498	211.4	261.6
Magnesia, J. M., 1½ in.	0.450	192.0	237.6
F. F. and A. S. F. 2 in.	0.378	161.4	200.0
Bare pipe	3.000	2000.0	3120.0

Transmission of Heat by Radiation

The **absorption capacity** of a body for radiation is the ratio of the heat absorbed to the entire radiation received on the surface of the body. The **absolute "black body"** absorbs all the radiation received; hence its absorption capacity is 1. Bodies that are good absorbers have also high capacity for radiation. Bodies that are good reflectors, as polished steel or silver, have correspondingly small absorption capacity and radiating power. Table 26 gives the relative absorbing, radiating and reflecting properties of certain bodies.

Table 26. Relative Radiating and Reflecting Capacities

Substance	Absorbing or radiating capacity	Reflecting power	Substance	Absorbing or radiating capacity	Reflecting power
Water	100	.....	Steel, polished	17	83
Marble	93–98	7–2	Tin	15	85
Glass	90	10	Brass, dead polished	11	89
Ice	85	15	Brass, bright	7	93
Cast iron, polished	25	75	Copper, hammered	7	93
Wrot. iron, polished	23	77	Gold	5	95
Mercury	23	77	Silver, polished	3	97
Zinc	19	81			

The Stefan-Boltzmann Law. The radiating power of a body is proportional to the fourth power of the absolute temperature of the body. This law holds exactly for an absolute black body; for other bodies it holds with sufficient exactness for practical purposes.

Let A = area of radiating surface, sq. ft., z = time in hours, T = absolute temperature on the Fahrenheit scale, and Q = heat radiated in B.t.u. Then

$$Q = CAz\,(T/100)^4$$

The radiation constant C depends on the substance and on the character of the radiating surface. For the absolute black body, $C = 0.1618$. Values of C for various substances are given in Table 27.

Table 27. Radiation Constants of Various Materials

Material	Temp. range of experiment, deg. fahr.	C	Material	Temp. range of experiment, deg. fahr.	C
Glass, smooth	70	0.154	Red sandstone*	140–400	0.10
Brass, dull	100–660	0.0362	Italian marble*	140–400	0.095
Lampblack	32–100	0.154	Granite*	140–400	0.0745
Copper, slightly polished	100–540	0.0278	Dolomite*	140–400	0.0685
Wrought-iron, dull, oxidised	70–670	0.154	Clay	140–400	0.065
Wrought-iron, clean, bright	85–225	0.0562	Field soil	140–400	0.063
Wrought-iron, highly polished	105–480	0.0467	Chalk	140–400	0.051
Zinc, dull	120–545	0.034	Gravel	140–400	0.0481
Cast iron, rough, highly oxidized	105–480	0.157	Water	140	0.112
Lime plaster, rough, white	50–195	0.151	Ice	32	0.106
Basalt*	140–400	0.120	Gold plate, shining but not polished	70	0.082
Slate*	140–400	0.115			
Humus	140–400	0.110			

* Finished smooth, but not shining.

The simplest case of heat radiation is that in which a body having an external surface A of uniform temperature t_1 is enclosed within a second surface having a uniform temperature t_2. Let C_1, C_2 and C be the radiation

constants for the two surfaces and for the absolute black body, respectively. Then the heat transmitted by radiation in z hours is

$$Q = Az[(T_1/100)^4 - (T_2/100)^4]/[(1/C_1) + (1/C_2) - (1/C)]$$

The following example (Dalby, *Inst. M. E.*, Oct., 1909) shows an application of this formula to steam boilers. Let the temperature of the furnace be 3000 deg. and that of the boiler plate or tubes, 800 deg. fahr., abs. The constant C_1 of the incandescent carbon may be taken as equal to the constant C of the black body, and C_2 for iron may be taken as 0.154. The heat radiated per sq. ft. of the boundary per hour is consequently

$$0.154[(3000/100)^4 - (800/100)^4] = 124{,}100 \text{ B.t.u.}$$

Ray and Kreisinger (Breckenridge, "Study of Four Hundred Steaming Tests," U. S. Geol. Survey, 1907) have studied the **relations between heat transmitted by radiation and by convection** in a Heine boiler. It was found that the heat transmitted by radiation increased with the temperature, as indicated by the Stefan law. The ratio of heat received by radiation to that received by convection varied from 3 to 15 per cent.

GENERAL PRINCIPLES OF THERMODYNAMICS

Notation:

Q, q = quantity of heat absorbed, B.t.u.
p = absolute pressure, lb. per sq. ft.
M = weight of substance under consideration, lb.
V, v = volume, cu. ft.
t = temperature, deg. fahr.
$T = t + 459.6$ = absolute temperature.
U, u = internal energy, B.t.u.
I, i = heat content at constant pressure, B.t.u.
S, s = entropy.
$J = 777.7$ = mechanical equivalent of heat.
$A = 1/J$ = reciprocal of mechanical equivalent.
c_p = specific heat at constant pressure.
c_v = specific heat at constant volume.
W = external work performed during a change of state.

In this notation small letters denote magnitudes referred to unit weight of the substance, capital letters corresponding magnitudes referred to M units of weight. Thus, v denotes the volume of 1 lb., $V = Mv$, the volume of M lb. Similarly, $U = Mu$, $S = Ms$, etc. Subscripts are used to indicate different states; thus, p_1, v_1, T_1, u_1, s_1, refer to the initial state, p_2, v_2, T_2, u_2, s_2 refer to the final state. Q_{12} is used to denote the heat absorbed by a body during the change from state 1 to state 2, and W_{12} denotes the external work done during the same change.

State of a Substance.—Characteristic Equations. Assuming that the substance under consideration (as air or steam) is homogeneous and of uniform density and temperature, and that it is subjected to a uniform pressure, the state of the substance is given by the values of p, v, and T. In general, two of these three variables may be changed independently, and the value of the third will then depend upon the values assigned to these two. That is, if v and T are taken to vary independently, then $p = f(v, T)$. The equation that expresses the relation between p, v, and T is called the **characteristic equation** of the substance in question.

For **perfect gases** the characteristic equation has the simple form

$$pv = RT.$$

This is approximately the equation of air, oxygen, nitrogen, and hydrogen. For certain gases, the equation of van der Waals

$$p = \frac{BT}{v - b} - \frac{a}{v^2}$$

represents quite accurately the states of the substance. The empirical equation $v - c = (BT/p) - [(1 + 3ap^{1/2})\, m/T^n]$ applies to **superheated steam.**

Any magnitude that depends upon the state of the substance may be used as a variable to describe the state. In addition to p, v, and T, the entropy s, energy u, and heat content i may be so used.

Fundamental Laws of Thermodynamics

Transformations of Energy are subject to two general laws: (1) The law of **conservation of energy**, which may be stated as follows: The total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. (2) The law of **degradation of energy**, according to which the result of any transformation of energy is a reduction in the quantity of energy available for transformation into useful work.

The following are familiar examples of degradation: Work transformed into heat through friction; electrical energy transformed into heat in the conducting system; flow of heat from a body of higher temperature to one of lower temperature; throttling or wire-drawing processes.

The First Law of Thermodynamics is the conservation law applied to the transformation of heat into work, or *vice versa*. When work is expended in producing heat, the quantity of heat generated is equivalent to the work done; conversely, when heat is employed to do work, a quantity of heat precisely equivalent to the work done disappears.

The first law is expressed symbolically by the equation $W = JQ$, in which W denotes the work and J the **mechanical equivalent of heat.** The following are the values of the constant J in various units:

1 gram-calorie = 4.184 joules
1 kg.-calorie = 426.65 m.-kg.
1 B.t.u. = 777.64 ft.-lb.

For ordinary calculations the values 427 and 778 are sufficiently accurate. Another useful relation is

1 h.p.-hr. = 2546 B.t.u.

In writing the equations of thermodynamics it is frequently convenient to use the reciprocal of J, which is denoted by A; thus $A = 1/J$.

The Energy Equation. The first law applied to a change of state of a substance is expressed by the equation

$$JQ_{12} - W_{12} = E_2 - E_1$$

in which E_1 denotes the total energy of the substance in the initial state and E_2 the total energy in the final state. The total energy E is made up of the internal energy U and the external kinetic energy. The latter comes into consideration in the flow of fluids. In case the kinetic energy remains constant or = 0, the energy equation becomes

$$Q_{12} = U_2 - U_1 + AW_{12}$$

If the work W arises from the overcoming of fluid pressure, then $W = \int pdV$ and the equation takes the form

$$dQ = dU + Apdv, \text{ or } Q_{12} = U_2 - U_1 + A\int_{V_1}^{V_2} pdV$$

The change of energy depends upon the initial and final states of the system only. The external work depends, however, on the relation between p and V during the change of state, that is, upon the path; hence the heat absorbed also depends upon the path.

If a system passes through a closed cycle of processes and returns to its initial state, the change of energy for the cycle is zero; hence for a closed cycle the heat absorbed by the system is the equivalent of the external work. This statement is expressed symbolically by the equation $J(Q) = (W)$, where (Q) denotes the heat absorbed for the cycle and (W) the net work performed. If the change of state is adiabatic, the heat absorbed is zero and the external work is gained at the expense of the intrinsic energy of the system. That is, $W_{12} = U_1 - U_2$.

The Second Law of Thermodynamics is essentially the law of degradation of energy. While the first law gives a relation that must be satisfied in any transformation of energy, it is the second law that gives information regarding the possibility of transformation and the availability of a given form of energy for transformation into work. The following is a general statement of the second law: **No change in a system of bodies that takes place of itself can increase the available energy of a system.**

A more concrete statement is that of Kelvin, namely: It is impossible by means of inanimate material agency to derive mechanical effect from any portion of matter by cooling it below the temperature of surrounding objects. In effect, Kelvin's statement denies the possibility of deriving work directly from the heat contained in the atmosphere.

The availability of a given quantity of heat energy for transformation into work is given by the efficiency of the **ideal Carnot engine**. Thus, if T denote the temperature of the source of heat and T_0 that of the refrigerator, or coldest body available, then the efficiency is $e = (T - T_0)/T = 1 - (T_0/T)$. If heat Q is taken from the source the part $Q[1 - (T_0/T)]$ may be transformed into work under ideal conditions, but the part $Q(T_0/T)$ at least must be rejected.

Entropy. Experience shows that every actual physical process is irreversible and accompanied by frictional effects. As a result the actual irreversible process is accompanied by a decrease of the quantity of energy available for transformation into work, or, what is the same thing, an increase of unavailable energy. The increase of unavailable energy is the product of two factors: one is T_0, the lowest absolute temperature available (usually the temperature of the atmosphere), the other is a term of the form Q/T or $\int dQ/T$. To this second factor the name **increase of entropy** is given.

When the conception of increase of entropy is applied to the system composed of all the bodies involved in a change, that is, to an isolated system, it appears that the increase of entropy is a measure of the thermodynamic degeneration produced by the change. According to the law of degradation every natural change is accompanied by thermodynamic degeneration, therefore it is accompanied by an increase of entropy. The following important principle is evident: The direction of a process, physical or chemical, that occurs of itself is such as will bring about an increase of entropy of the system. This principle is the foundation of the application of thermodynamics to chemistry.

The conception of increase of entropy may be extended to a body not isolated but in thermal communication with other bodies. In this case the change of entropy is given by the relation

$$S_2 - S_1 = \int_{T_1}^{T_2} \frac{dQ}{T} + \int_{T_1}^{T_2} \frac{dH}{T}$$

in which Q denotes heat absorbed by the body from outside and H heat generated within the system through friction. For a reversible frictionless change, the increase of entropy is simply $\int_{T_1}^{T_2} \frac{dQ}{T}$.

The entropy of the body, as thus defined, depends on the state only; hence S may be used with p, V, and T as a variable defining the state. For reversible frictionless changes, the defining equation gives $dQ = TdS$; hence the energy equation may be written $TdS = dU + ApdV$.

General Thermodynamic Relations. The first law gives the **energy equation**

$$dq = du + Apdv \tag{1}$$

the second law gives the **entropy equation**

$$dq = Tds \tag{2}$$

A third general equation is obtained by the introduction of a magnitude analogous to the energy u and defined by the relation

$$i = u + Apv \quad (i = U + ApV)$$

This is called the **heat content at constant pressure**, for the reason that $i_2 - i_1 = Q_{12}$ for a change at constant pressure. By the introduction of i, Eq. (1) takes the form

$$dq = di - Avdp \tag{3}$$

By various transformations the following relations are derived from these three fundamental equations:

1. **Maxwell's thermodynamic relations:**

$$\left(\frac{\partial T}{\partial v}\right)_s = -A\left(\frac{\partial p}{\partial s}\right)_v \qquad \left(\frac{\partial s}{\partial v}\right)_T = A\left(\frac{\partial p}{\partial T}\right)_v$$

$$\left(\frac{\partial T}{\partial p}\right)_s = A\left(\frac{\partial v}{\partial s}\right)_p \qquad \left(\frac{\partial s}{\partial p}\right)_T = -A\left(\frac{\partial v}{\partial T}\right)_p$$

2. **Relations involving specific heats:**

$$c_v = \left(\frac{\partial q}{\partial T}\right)_v = T\left(\frac{\partial s}{\partial T}\right)_v \qquad c_v = \left(\frac{\partial u}{\partial T}\right)_v$$

$$c_p = \left(\frac{\partial q}{\partial T}\right)_p = T\left(\frac{\partial s}{\partial T}\right)_p \qquad c_p = \left(\frac{\partial i}{\partial T}\right)_p$$

$$c_p - c_v = AT\left(\frac{\partial v}{\partial T}\right)_p\left(\frac{\partial p}{\partial T}\right)_v$$

$$\left(\frac{\partial c_v}{\partial v}\right)_T = AT\left(\frac{\partial^2 p}{\partial T^2}\right)_v \qquad \left(\frac{\partial c_p}{\partial p}\right)_T = -AT\left(\frac{\partial^2 v}{\partial T^2}\right)_p$$

3. Relations involving q, u, i, and s:

$$dq = c_v dT + AT\left(\frac{\partial p}{\partial T}\right)_v dv \qquad dq = c_p dT - AT\left(\frac{\partial v}{\partial T}\right)_p dp$$

$$du = c_v dt + A\left[T\left(\frac{\partial p}{\partial T}\right)_v - p\right]dv, \quad di = c_p dT - A\left[T\left(\frac{\partial v}{\partial T}\right)_p - v\right]dp$$

$$ds = c_v\frac{dT}{T} + A\left(\frac{\partial p}{\partial T}\right)_v dv = c_p\frac{dT}{T} - A\left(\frac{\partial v}{\partial T}\right)_p dp$$

The derivatives involved in these relations are found from the characteristic equation of the substance under investigation. Thus, for a perfect gas,

$pv = BT$, whence $\left(\frac{\partial p}{\partial T}\right)_v = \frac{B}{v}$, $\left(\frac{\partial v}{\partial T}\right)_p = \frac{B}{p}$. Substitution of these expressions in the preceding equations gives the following results for gases:

$$c_p - c_v = AB$$

$$dq = c_v dT + Apdv \qquad dq = c_p dT - Avdp$$

$$du = c_v dT \qquad di = c_p dT$$

$$ds = c_v \frac{dT}{T} + AB\frac{dv}{v} = c_p \frac{dT}{T} - AB\frac{dp}{p}.$$

As an example of the use of the Clausius relation $\left(\frac{\partial c_p}{\partial p}\right)_T = -AT\left(\frac{\partial^2 v}{\partial T^2}\right)_p$, consider the case of superheated steam, the equation of which is $v - c = \frac{BT}{p} - (1+3ap^{\frac{1}{2}})\frac{m}{T^n}$. $\left(\frac{\partial^2 v}{\partial T^2}\right)_p = -\frac{mn(n+1)}{T^{n+2}}(1+3ap^{\frac{1}{2}})$, whence $\left(\frac{\partial c_p}{\partial p}\right)_T = \frac{Amn(n+1)}{T^{n+1}}(1+3ap^{\frac{1}{2}})$, and $c_p = \frac{Amn(n+1)}{T^{n+1}}p(1+2ap^{\frac{1}{2}}) + f(T)$.

It follows that c_p for superheated steam varies with the temperature and also with the pressure.

Graphical Representation. The change of state of a substance may be shown graphically by taking any two of the six variables p, V, T, S, U, I, as independent co-ordinates and drawing a curve to represent the successive values of these two variables as the change proceeds. While any pair may be chosen, there are three systems of graphical representation that are specially useful.

1. p and V. The curve (Fig. 2) represents the simultaneous values of p and V during the change from state 1 to state 2. The area between the curve and the axis OV is given by the integral $\int_{V_1}^{V_2} pdV$ and therefore represents the external work W_{12} done by the gas during the change. The area included by a closed cycle represents the work of the cycle (as in the indicator diagram of the steam engine).

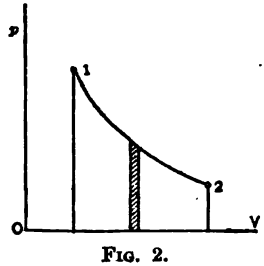


FIG. 2.

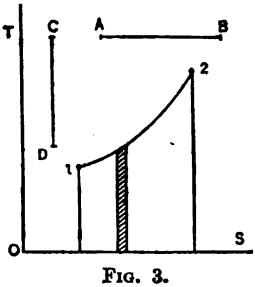


FIG. 3.

2. T and S (Fig. 3). The absolute temperature T is taken as the ordinate; the entropy S as the abscissa. The area between the curve of change of state and the S-axis is given by the integral $\int_{S_1}^{S_2} TdS$, and it therefore represents the heat Q_{12} absorbed by the substance from external sources provided there are no irreversible frictional effects. On the T-S diagram an isothermal is a straight line, as AB, parallel to the S-axis; a reversible adiabatic is a straight line, as CD, parallel to the T-axis.

In the case of internal generation of heat through friction, as in steam turbines, the increase of entropy is given by $\int_{T_1}^{T_2} \frac{dH}{T}$ (see p. 197) and the area under the curve represents the heat H thus generated. In this case an adiabatic is *not* a straight line parallel to the T-axis.

3. I and S. In the system of representation devised by Dr. Mollier, the heat content I is taken as the ordinate and the entropy S as the abscissa. If on this diagram (Fig. 4) a line of constant pressure, as 12, be drawn, the heat absorbed during the change at constant pressure is given by $Q_{12} = I_2 - I_1$, and this is represented by the line segment 23. The **Mollier diagram** is specially useful in problems that involve the flow of fluids, throttling, and the action of steam in turbines. The advantage lies in the fact that heat and work are represented by line segments instead of areas. A Mollier diagram for steam is given on p. 215, and for ammonia on p. 219.

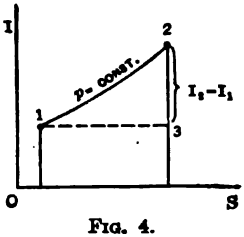


FIG. 4.

PERFECT GASES

Laws of Gases. The so-called perfect gases are those that obey very closely the laws of Boyle and Gay-Lussac. The two laws are combined in the characteristic equations $pv = RT$, $pV = MRT$, in which R is a constant for any gas. The value of R for a gas is inversely proportional to the molecular weight m of the gas and is given by the relation $1544/m$.

Since for a perfect gas $\left(\frac{\partial p}{\partial T}\right)_v = \frac{R}{v} = \frac{T}{p}$, the general equation (p. 198) becomes $\left(\frac{\partial u}{\partial v}\right)_T = AT\left(\frac{\partial p}{\partial T}\right)_v - Ap = 0$; that is, the internal energy u is independent of the volume and depends on the temperature only.

Specific Heats. According to the experiments of Regnault and others the specific heats c_p and c_v of diatomic gases (oxygen, nitrogen, etc.) are practically constant for considerable ranges of temperature, and in ordinary applications that involve only moderate ranges of temperature they may be taken as constant. Other gases, as superheated steam, ammonia, and carbon dioxide, show a considerable variation of specific heat with the temperature. The values of c_p and c_v in Table 28 are therefore to be considered as approximately correct for the range 32 deg.–400 deg. fahr. For further discussion of specific heats of superheated steam, see p. 212.

The difference of the specific heats c_p and c_v is given by the relation $c_p - c_v = AR = 1.9855/m$, or approximately $2/m$. The ratio c_p/c_v is denoted by k and the value of k for diatomic gases is about 1.4.

Specific Weight and Specific Volume. Referred to standard atmospheric pressure and a temperature of 32 deg. fahr., the weight of a cubic foot of gas is $0.002788\,m$ lb. and the volume of one pound in cubic feet is $358.65/m$. The **mol** is frequently taken as the unit of weight for gases. The mol is defined as m lb. where m denotes the molecular weight; thus, for oxygen, 1 mol = 32 lb. For any gas, therefore, the volume of 1 mol at 32 deg. fahr. and atmospheric pressure is 358.65 cu. ft. If M' denotes the weight in mols, then $M = M'm$, and the characteristic equation becomes

$$pV = 1544\,M'T, \quad \text{or} \quad ApV = 1.9855\,M'T$$

The specific heats referred to the mol as the unit of weight are mc_p and mc_v, respectively, and $mc_p - mc_v = 1.9855$ (= 2, approx.); $mc_p = 1.9855k/(k-1)$; $mc_v = 1.9855/(k-1)$.

Taking $k = 1.4$ for diatomic gases, then approximately $mc_p = 7$, $mc_v = 5$.

Table 28. Properties of Gases

Gas	Chemical symbol	Number of atoms	Molecular weight: Approximate	Molecular weight: Exact, O₂ = 32	Weight in lb. of 1 cu. ft. at atmos. pressure: At 62 deg. fahr.	Weight in lb. of 1 cu. ft. at atmos. pressure: At 32 deg. fahr.	Density relative to air	Gas constant, R	Specific heat per lb.: c_p	Specific heat per lb.: c_v	Specific heat per cu. ft. at atmos. pressure and 62 deg. fahr.: c_p	Specific heat per cu. ft. at atmos. pressure and 62 deg. fahr.: c_v	$k = c_p/c_v$
Helium...	He	1	4.0	4.0	0.0105	0.0112	0.137	386.0	1.25	0.75	0.0131	0.0079	1.66
Argon....	Ar	1	40.0	39.9	0.1048	0.1112	1.378	38.70	0.124	0.075	0.0131	0.0079	1.66
Air.....	.....	.....	29.0	28.95	0.0761	0.0807	1	53.34	0.241	0.171	0.0183	0.0130	1.40
Oxygen...	O_2	2	32.0	32	0.0840	0.0892	1.105	48.25	0.217	0.155	0.0182	0.0130	1.40
Nitrogen..	N_2	2	28.0	28.08	0.0737	0.0783	0.970	54.99	0.247	0.176	0.0182	0.0130	1.40
Hydrogen	H_2	2	2.0	2.016	0.00529	0.00562	0.0696	765.86	3.42	2.44	0.0181	0.0129	1.40
Nitric oxide.....	NO	2	30.0	30.04	0.0789	0.0838	1.038	51.40	0.231	0.165	0.0183	0.0130	1.40
Carbon monoxide	CO	2	28.0	28.00	0.0734	0.0780	0.968	55.14	0.243	0.172	0.0180	0.0126	1.41
Hydrochloric acid.....	HCl	2	36.5	36.45	0.0958	0.1017	1.260	42.35	0.191	0.136	0.0183	0.0130	1.40
Carbon dioxide..	CO_2	3	44.0	44.00	0.1156	0.1227	1.520	35.09	0.210	0.160	0.0243	0.0185	1.31
Nitrous oxide....	N_2O	3	44.0	44.08	0.1157	0.1229	1.522	35.03	0.221	0.171	0.0256	0.0198	1.26
Sulphur dioxide..	SO_2	3	64.0	64.06	0.1684	0.1786	2.213	24.10	0.154	0.123	0.0260	0.0207	1.25
Ammonia.	NH_3	4	17.0	17.06	0.04483	0.0476	0.590	90.50	0.523	0.399	0.0234	0.0178	1.31
Acetylene.	C_2H_2	4	26.0	26.02	0.0684	0.0725	0.899	59.34	0.350	0.270	0.024	0.0185	1.28
Methyl chloride..	CH_3Cl	5	50.5	50.47	0.1326	0.1407	1.744	30.59	0.24	0.20	0.032	0.0265	1.20
Methane..	CH_4	5	16.0	16.03	0.0421	0.0447	0.554	96.31	0.593	0.450	0.025	0.019	1.32
Ethylene..	C_2H_4	6	28.0	28.03	0.0738	0.0780	0.969	55.08	0.40	0.33	0.029	0.024	1.20

Gas Mixtures. Let V denote the total volume of the mixture, M_1, M_2, M_3 . . . the weights of the constituent gases, R_1, R_2, R_3 . . . the corresponding gas constants, and R_m the constant for the mixture. The **partial pressures** of the constituents, that is, the pressures that the constituents would have if occupying the total volume V, are $p_1 = M_1R_1T/V$, $p_2 = M_2R_2T/V$, etc.

According to Dalton's law, the **total pressure** p **of the mixture** is the sum of the partial pressures; that is, $p = p_1 + p_2 + p_3 + \ldots$ Let $M = M_1 + M_2 + M_3 + \ldots$ denote the total weight of the mixture; then $pV = MR_mT$ and $R_m = \Sigma(M_iR_i)/M$. Also $p_1/p = M_1R_1/MR_m$, $p_2/p = M_2R_2/MR_m$, etc.

Let V_1, V_2, V_3, . . . denote the volumes that would be occupied by the constituents at pressure p and temperature T (these are given by the volume composition of the gas). Then $V = V_1 + V_2 + V_3 + \ldots$ and the apparent molecular weight m_m of the mixture is $m_m = \Sigma(m_iV_i)/V$. Then $R_m = 1544/m_m$.

Volume of 1 lb. at 32 deg. fahr. and atmos. pressure $= 358.65/m_m$.

Weight of 1 cu. ft. at 32 deg. fahr. and atmos. pressure $= 0.002788m_m$.

The specific heats of the mixture are, respectively

$$c_p = \Sigma(M_ic_{pi})/M, \qquad c_v = \Sigma(M_ic_{vi})/M.$$

Energy, Heat Content, and Entropy. If a gas changes from an initial state p_1, V_1, T_1, to a final state p_2, V_2, T_2, the following equations hold:

$$U_2 - U_1 = Mc_v (T_2 - T_1) = A(p_2V_2 - p_1V_1)/(k - 1)$$
$$I_2 - I_1 = Mc_p (T_2 - T_1) = Ak(p_2V_2 - p_1V_1)/(k - 1).$$
$$S_2 - S_1 = M\left[c_v \log_e \frac{T_2}{T_1} + AR \log_e \frac{V_2}{V_1}\right] = M\left[c_p \log_e \frac{T_2}{T_1} - AR \log_e \frac{p_2}{p_1}\right]$$
$$= M\left[c_p \log_e \frac{V_2}{V_1} + c_v \log_e \frac{p_2}{p_1}\right].$$

In general, the energy per unit weight is $u = c_vT + u_0$,
the heat content is $i = c_pT + i_0$,
and the entropy is $s = c_v \log T + AR \log v + s_0$
$= c_p \log T - AR \log p + s'_0 = c_p \log v + c_v \log p + s''_0$

The two fundamental equations for gases are

$$dq = c_vdT + Apdv, \qquad dq = c_pdT - Avdp.$$

Special Changes of State

In the following formulæ the subscripts 1 and 2 refer to the initial and final states, respectively.

1. Constant Volume: $p_2/p_1 = T_2/T_1$. $Q_{12} = U_2 - U_1 = Mc_v (t_2 - t_1)$. $Q_{12} = A V(p_2 - p_1)/(k - 1)$. $W = 0$. $s_2 - s_1 = Mc_v \log_e(T_2/T_1)$.

2. Constant Pressure: $V_2/V_1 = T_2/T_1$. $W_{12} = p(V_2 - V_1) = MR(t_2 - t_1)$. $Q_{12} = Mc_p (t_2 - t_1) = AkW_{12}/(k - 1)$. $s_2 - s_1 = Mc_p \log_e(T_2/T_1)$.

3. Isothermal. (Constant Temperature): $p_2/p_1 = V_1/V_2$. $U_2 - U_1 = 0$. $W_{12} = MRT \log_e(V_2/V_1) = p_1V_1 \log_e(V_2/V_1)$. $Q_{12} = AW_{12}$. $s_2 - s_1 = Q_{12}/T = MAR \log_e(V_2/V_1)$.

4. Adiabatic: $T_2/T_1 = (V_1/V_2)^{k-1} = (p_2/p_1)^{(k-1)/k}$. $p_1V_1^k = p_2V_2^k$.

$$AW_{12} = U_1 - U_2 = Mc_v(t_1 - t_2). \quad Q_{12} = 0. \quad s_2 - s_1 = 0.$$
$$W_{12} = (p_1V_1 - p_2V_2)/(k - 1) = p_1V_1[1 - (p_2/p_1)^{(k-1)/k}]/(k - 1).$$

5. Polytropic. This name is given to the change of state which is represented by the equation $pV^n = \text{const.}$ The polytropic curve usually represents sufficiently well actual expansion and compression curves in motors and air compressors. By giving n different values the preceding changes are made special cases of the polytropic change, thus,

for $n = 1$,	$pv = \text{const.}$	isothermal
$n = k$,	$pv^k = \text{const.}$	adiabatic
$n = 0$,	$p = \text{const.}$	constant pressure
$n = \infty$,	$v = \text{const.}$	constant volume

For a polytropic change the specific heat is constant and is given by the relation $c_n = c_v(n - k)/(n - 1)$; hence for $1 < n < k$, c_n is negative. This is the case in **air compression.** The following are the principal formulæ: $p_1V_1^n = p_2V_2^n$. $T_2/T_1 = (V_1/V_2)^{n-1} = (p_2/p_1)^{(n-1)/n}$.

$$W_{12} = (p_1V_1 - p_2V_2)/(n - 1) = p_1V_1[1 - (p_2/p_1)^{(n-1)/n}]/(n - 1).$$
$$Q_{12} = Mc_n(t_2 - t_1).$$
$$AW_{12} : U_2 - U_1 : Q_{12} = k - 1 : 1 - n : k - n$$

Table 29. Adiabatic and Polytropic Expansion

$\frac{p_1}{p_2}$	n =				n =			
	1.4 (Adiabatic)	1.3	1.2	1.1	1.4 (Adiabatic)	1.3	1.2	1.1
	$V_2/V_1 = (p_1/p_2)^{1/n} =$				$T_1/T_2 = (p_1/p_2)^{(n-1)/n} =$			
1.1	1.070	1.076	1.083	1.090	1.028	1.022	1.016	1.009
1.2	1.139	1.151	1.164	1.180	1.053	1.043	1.031	1.017
1.3	1.206	1.224	1.244	1.269	1.078	1.062	1.045	1.024
1.4	1.271	1.295	1.323	1.358	1.101	1.081	1.058	1.031
1.5	1.336	1.366	1.401	1.445	1.123	1.098	1.070	1.038
1.6	1.399	1.436	1.479	1.533	1.144	1.115	1.081	1.044
1.7	1.461	1.504	1.557	1.620	1.164	1.130	1.092	1.050
1.8	1.522	1.571	1.633	1.706	1.183	1.145	1.103	1.055
1.9	1.581	1.638	1.706	1.791	1.201	1.160	1.113	1.060
2.0	1.641	1.705	1.782	1.879	1.219	1.174	1.123	1.065
2.5	1.924	2.023	2.145	2.300	1.299	1.235	1.165	1.087
3.0	2.193	2.330	2.498	2.715	1.369	1.289	1.201	1.105
3.5	2.449	2.624	2.842	3.126	1.431	1.336	1.232	1.121
4.0	2.692	2.907	3.177	3.505	1.487	1.378	1.260	1.134
4.5	2.926	3.187	3.500	3.925	1.526	1.415	1.285	1.147
5.0	3.156	3.449	3.824	4.320	1.583	1.449	1.307	1.157
5.5	3.378	3.712	4.142	4.710	1.627	1.482	1.328	1.167
6.0	3.598	3.970	4.447	5.100	1.668	1.512	1.348	1.177
6.5	3.809	4.218	4.760	5.483	1.707	1.540	1.366	1.186
7.0	4.012	4.467	5.058	5.861	1.742	1.566	1.383	1.194
7.5	4.217	4.710	5.360	6.250	1.778	1.591	1.399	1.201
8.0	4.415	4.950	5.650	6.620	1.811	1.616	1.414	1.208
9.0	4.800	5.420	6.240	7.370	1.873	1.660	1.442	1.221
10.0	5.188	5.885	6.820	8.120	1.931	1.701	1.468	1.233

Construction of Polytropic Curves (Fig. 5). Through the origin O a line OA is drawn at any convenient angle a with the V-axis. Then a second line OB is drawn making the angle b with the p-axis, where angle b is determined by the relation $(1 + \tan b) = (1 + \tan a)^n$. From the initial point 1 (p_1, V_1) locate points C and D on the axes. Then, by drawing 45-deg. lines between the axes and the auxiliary lines, as shown in the figure, the successive points on the curve, 2, 3, 4, etc., are obtained.

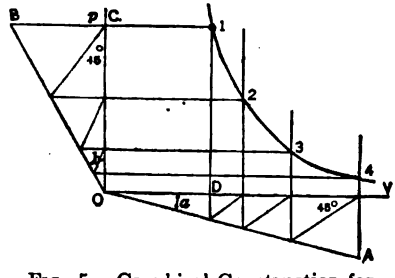


FIG. 5.—Graphical Construction for Polytropic Curves.

Determination of Exponent n. Lay off successive values of p and V, measured at chosen points on the curve under investigation, on logarithmic cross-section paper; or, lay off values of log p and log V on ordinary cross-section paper. If n is a constant, the points will lie in a straight line, and the slope of the line gives the value of n.

If two representative points (p_1, V_1 and p_2, V_2) be chosen, then

$$n = (\log p_1 - \log p_2)/(\log V_1 - \log V_2).$$

Several pairs of points should be used in order to test the constancy of n.

Changes of State with Variable Specific Heat. In case of a considerable range of temperature, as in the internal-combustion motor, the assumption of constant specific heat is not permissible, and the equations referring to changes of state must be suitably modified. Experiments on the specific heat of various gases show that the specific heat may be taken as a linear function of the temperature: thus, $c_v = a + bT$; $c_p = a' + bT$.

From the fundamental equation for gases, $dq = c_v dT + Apdv$, the following expressions are found for the change of energy and entropy, respectively:

$$U_2 - U_1 = M[a(T_2 - T_1) + 0.5b(T_2^2 - T_1^2)]$$

$$S_2 - S_1 = M[a \log_e (T_2/T_1) + b(T_2 - T_1) + AR \log_e (V_2/V_1)].$$

For an adiabatic change,

$$W_{12} = J(U_1 - U_2)$$

$$AR \log_e (p_2/p_1) = (a + AR) \log_e (T_2/T_1) + b(T_2 - T_1)$$

$$AR \log_e (V_1/V_2) = a \log_e (T_2/T_1) + b(T_2 - T_1)$$

Ideal Cycles with Perfect Gases

Gases are used as heat media in several important types of motors. In air compressors, air engines, and air refrigerating machines, atmospheric air is the medium. In the internal-combustion engine the medium is a mixture of products of combustion. Motors using gases are operated in certain well-defined cycles, which are described in the following sections. In the analyses given ideal conditions that cannot be attained by actual motors are assumed. However, conclusions derived from such analyses are usually valid for the modified actual cycle.

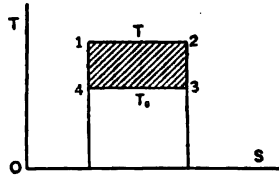


FIG. 6.—Carnot Cycle.

In the following the subscripts 1, 2, 3, etc., refer to corresponding points shown in the figures. The work of the cycle is denoted by (W) and the net heat absorbed by (Q).

Carnot's Cycle. The Carnot cycle (Fig. 6) has historical interest only. It consists of two isothermals and two adiabatics. The heat absorbed along the upper isothermal 12 is $Q_{12} = MART \log_e (V_2/V_1)$, and the heat transformed into work, represented by the cycle area is $A(W) = Q_{12}[1 - (T_0/T)]$.

Hence $(W) = MR(T - T_0) \log_e (V_2/V_1)$

If the cycle is traversed in the reverse sense, $Q_{43} = MART_0 \log_e (V_3/V_4)$ is the heat absorbed from the cold body (brine), and the ratio Q_{43}: $A(W) = T_0 : (T - T_0)$ is the **coefficient of performance** of the refrigerating machine.

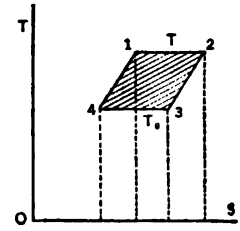


FIG. 7.—Stirling Cycle.

Stirling and Ericsson Cycles. In the Stirling air engine (Fig. 7) the adiabatics of the Carnot cycle are replaced by constant-volume curves; in the Ericsson engine by constant-pressure curves. By the use of a regenerator the

heat Q_{23} rejected during the operation 23 is stored and is given back to the medium during the operation 41. In the ideal case, $Q_{23} = Q_{41}$, hence the heat absorbed from the source is $Q_{12} = MART \log_e V_2/V_1$, as in the Carnot cycle, and the efficiency is identical with that of the Carnot cycle.

Air engines of the Stirling and Ericsson type, in which the medium is separated from the furnace by a metal wall, have been failures, and have been replaced by the internal-combustion type, in which the air comes into direct contact with the fuel inside of the working cylinder. The rapid chemical action supported by the medium itself makes possible the rapid heating of large quantities of air to a very high temperature. And by proper cooling of the outside surface of the metal walls, the deterioration of the metal is prevented even at high temperatures.

The ideal cycles usually employed for internal-combustion engines may be classified in two groups: 1. Explosive—Otto. 2. Non-explosive—Diesel, Joule.

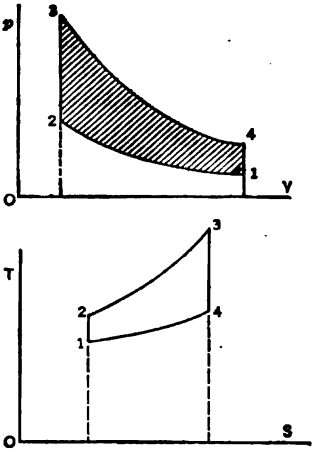


FIG. 8.—Otto Cycle.

The Otto Cycle (Fig. 8). Adiabatic (or polytropic) compression 12 is followed by ignition and rapid heating at constant volume, 23. This is followed by adiabatic expansion, 34. Assuming compression and expansion to be adiabatic, the following relations hold:

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \left(\frac{p_3}{p_4}\right)^{\frac{k-1}{k}} = \left(\frac{V_1}{V_2}\right)^{k-1}.$$

$$Q_{23} = Mc_v(T_3 - T_2).$$

$$(W) = JQ_{23}[1 - (T_1/T_2)] = JMc_v(T_3 - T_4 - T_2 + T_1)$$

$$\text{Efficiency} = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{V_2}{V_1}\right)^{k-1} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}.$$

If the compression and expansion curves are polytropics with the same value of n, replace k by n in the first relation above. In this case,
$(W) = [(p_3V_3 - p_4V_4) - (p_2V_2 - p_1V_1)]/(n-1) = MR(T_3 - T_4 - T_2 + T_1)/(n-1)$.

The mean effective pressure of the diagram is given by

$$p_m = ap_1[(p_3/p_2) - 1],$$

where a has the values given in the following table.

$p_2/p_1 =$	3	4	5	6	8	10	12	14	16
$(n = 1.4)$..... $a =$	1.70	1.94	2.13	2.31	2.62	2.88	3.10	3.31	3.50
$(n = 1.3)$..... $a =$	1.69	1.92	2.11	2.28	2.57	2.81	3.08	3.22	3.39
$(n = 1.2)$..... $a =$	1.68	1.90	2.08	2.25	2.51	2.74	2.94	3.12	3.27

The Joule Cycle (Fig. 9) consists of two adiabatics and two constant-pressure lines. The following relations hold:

$$V_3/V_2 = V_4/V_1 = T_3/T_2 = T_4/T_1.$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left(\frac{V_1}{V_2}\right)^{k-1} = \left(\frac{V_4}{V_3}\right)^{k-1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}$$

$(W) = JMc_p(T_3 - T_2 - T_4 + T_1)$. Efficiency $= (W)/JQ_{23} = 1 - (T_1/T_2)$.

For the use of the Joule cycle in refrigeration, see p. 227.

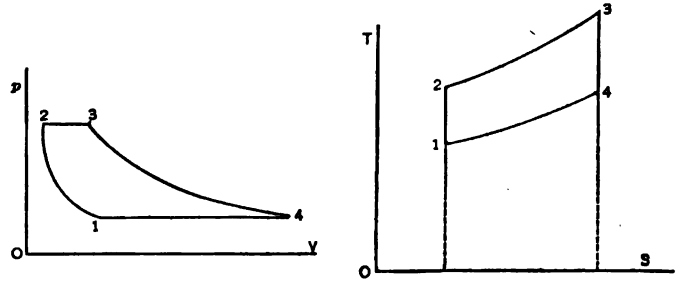


FIG. 9.—Joule Cycle.

The Diesel Cycle. In the Diesel oil engine air is compressed to a pressure of about 500 lb. per sq. in. Fuel is then injected into the air and, as the temperature is above the ignition point, burns at nearly constant pressure (23, in Fig. 10). Adiabatic expansion of the products of combustion is followed by exhaust and suction of fresh air, as in the Otto cycle.

The work obtained is

$$(W) = JM[c_p(T_3 - T_2) - c_v(T_4 - T_1)],$$

and the efficiency of the ideal cycle is

$$1 - [(T_4 - T_1)/k(T_3 - T_2)].$$

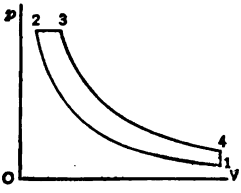


FIG. 10.—Diesel Cycle.

Air Compression. It is assumed that the compressor works under ideal conditions without clearance and without friction losses. If the compression from p_1 to p_2 (Fig. 11*a*) follows the law $pV^n = \text{const.}$, the work represented by the indicator diagram is

$$W = n(p_2V_2 - p_1V_1)/(n - 1) = np_1V_1[(p_2/p_1)^{(n-1)/n} - 1]/(n - 1).$$

The temperature at the end of compression is given by $T_2/T_1 = (p_2/p_1)^{(n-1)/n}$. The work W is smaller the smaller the value of n, and to reduce n from the adiabatic value 1.4 is the office of the water jacket. Under usual working conditions, n is about 1.3.

When the pressure p_2 is high it is advantageous to divide the process into two or more stages and cool the air between the cylinders. The saving effected is best shown on the T-S plane (Fig. 11*b*). With single-stage compression, 12 represents the compression from p_1 to p_2, and if the constant-pressure line 23 is drawn cutting the isothermal through point 1 in point 3, the area 1′1233′ represents the work W. When two stages are used, 14 represents the compression from p_1 to an intermediate pressure p', 45 cooling at constant pressure in the intercooler between the cylinders, and 56 the compression in

the second stage. The area under 14563 represents the work of the two stages and the area 2456 the saving effected by compounding. This saving is a maximum when $T_4 = T_6$, and this is the case when the intermediate pressure p' is given by $p' = \sqrt{p_1 p_2}$.

The total work in two-stage compression is

$$n\, p_1 V_1 [(p'/p_1)^{(n-1)/n} + (p_2/p')^{(n-1)/n} - 2] /(n - 1).$$

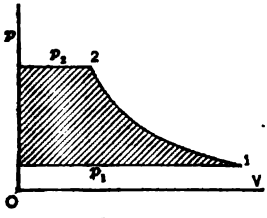


FIG. 11a.

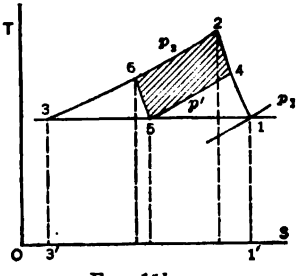


FIG. 11b.

Air-compressor Cycle.

VAPORS

General Characteristics of Vapors. Let a gas be compressed at constant temperature; then, provided this temperature does not exceed a certain critical value, the gas begins to liquefy at a definite pressure, which depends upon the temperature. At the beginning of liquefaction, a unit weight of gas will also have a definite volume v'', depending on the temperature. In Fig. 12, AB represents the compression and the point B gives the **saturation** pressure and volume. If the compression is continued the pressure remains constant with the temperature, as indicated by BC, until at C the substance is in the liquid state with the volume v'.

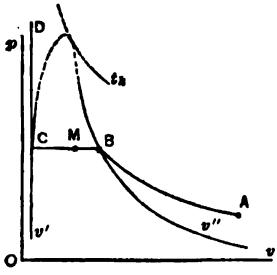


FIG. 12.

The curves v' and v'' giving the volumes for various temperatures at the end and beginning of liquefaction, respectively, may be called the **limit curves.** A point B on curve v'' represents the state of **saturated vapor**; a point C on the curve v' represents the liquid state; and a point M between B and C represents a mixture of vapor and liquid of which the part $x = MC/BC$ is vapor and the part $1 - x = BM/BC$ is liquid. The ratio x is called the **quality of the mixture.** The region between the curves v' and v'' is thus the region of liquid and vapor mixtures. The region to the right of curve v'' is the region of **superheated vapor.** The curve v'' dividing these regions represents the so-called **saturated vapor.**

For saturated vapor or a mixture of vapor and liquid, the pressure is a function of the temperature only, and the volume of the mixture depends upon the temperature and quality x. That is, $p = f(t)$, $v = F(t, x)$.

For the vapor in the superheated state the volume depends on pressure and temperature [$v = F_1(p, t)$], and these may be varied independently.

Critical States. If the temperature of the gas lies above a definite temperature t_k called the **critical temperature**, the gas cannot be liquefied by compression alone. The saturation pressure corresponding to t_k is the **critical pressure** and is denoted by p_k. At the critical state the limit curves v' and v'' merge; hence for temperatures above t_k it is impossible to have a mixture of vapor and liquid. Table 30 gives the critical data for various gases; also the boiling temperature t_b corresponding to atmospheric pressure.

Table 30. Critical Data for Various Gases

Substance	t_b, deg. fahr.	t_k, deg. fahr.	p_k, atmospheres	Substance	t_b, deg. fahr.	t_k, deg. fahr.	p_k, atmospheres
Acetylene, C_2H_2	−117.4	95.0	68.0	Ethylene	−157.0	50.0	54
Air	.....	−220.0	39.0	Helium	−450.4	−448.6	3
Alcohol, C_2H_6O	172.4	421.0	65.0	Hydrogen	−423.0	−402.0	20
Ammonia, NH_3	− 27.4	266.0	115.0	Hydrogen chloride	−112.0	125.6	87
Benzol, C_6H_6	176.0	554.0	50.0	Hydrogen sulphide	− 61.6	212.0	94
Bromine	142.0	512.0	.....	Methane, CH_4	−263.0	−115.6	57
Carbon dioxide	−110.0	88.2	77.0	Nitric oxide, NO	−238.0	−137.2	73
Carbon monoxide	−310.0	−222.0	35.9	Nitrous oxide, N_2O	−134.0	96.8	80
Carbon disulphide	115.0	468.0	78.1	Nitrogen	−321.0	−236.0	35
Chloroform	141.0	500.0	54.9	Oxygen	−297.0	−180.4	50
Chlorine	− 27.4	289.0	92.0	Pentane, C_5H_{12}	96.8	386.6	34
Ether, $C_4H_{10}O$	95.0	381.2	37.0	Sulphur dioxide	14.0	314.6	80
Ethane	−135.0	31.0	45.2	Water	212.0	706.1	218

Thermal Properties of Saturated Vapors and of Vapor and Liquid Mixtures

Notation:

v' = volume in cu. ft. of 1 lb. of liquid.
v'' = volume in cu. ft. of 1 lb. of saturated vapor, at given temperature.
c' = specific heat of liquid.
c'' = specific heat of saturated vapor, at given temperature.
r = latent heat, or heat required to vaporise 1 lb. of liquid at given constant pressure and temperature.
$AL = Ap(v'' - v')$ = external latent heat, *i.e.*, the heat equivalent of the external work performed during vaporization.
$l = r - Ap(v'' - v')$ = internal latent heat.
i', i'' = heat content of liquid and saturated vapor, respectively.
u', u'' = internal energy of liquid and saturated vapor, respectively.
s' = entropy of the liquid.
s'' = entropy of saturated vapor.

For a unit weight (1 lb.) of saturated vapor the following relations exist:

$$i'' = i' + r. \qquad u'' = u' + l. \qquad s' = \int_{491.6}^{T} c' dT/T. \qquad s'' = s' + (r/T).$$

$r/T = A\,(v'' - v')\,dp/dt$ (Clapeyron's equation), the derivative dp/dt being determined from the relation $p = f(t)$ between temperature and pressure at saturation.

For a unit weight of mixture of vapor and liquid, quality x, these relations become $i = i' + xr$, $u = u' + xl$, $s = s' + (xr/T)$;
also $v = v' + x(v'' - v')$.

Tables of Thermal Properties of Vapors

Water Vapor. The experimental data relating to the properties of water vapor have been correlated by Marks and Davis; and the values given in the Marks and Davis tables may be accepted as standard at the present time.

Table 21. Properties of Saturated Steam

Condensed from the Steam Tables of Marks and Davis by permission of the publishers, Longmans, Green and Co.

Abs. press., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, q'	Latent heat of evap., r	Heat content of steam, q''	Internal energy (B.t.u.) evap., I	Entropy	
								Water, s'	Evap., r/T
1	101.83	333.0	0.00300	69.8	1034.6	1104.4	972.9	0.1327	1.8427
2	126.15	173.5	0.00576	94.0	1021.0	1115.0	956.7	0.1749	1.7431
3	141.52	118.5	0.00845	109.4	1012.3	1121.6	946.4	0.2008	1.6840
4	153.01	90.5	0.01107	120.9	1005.7	1126.5	938.6	0.2198	1.6416
5	162.28	73.33	0.01364	130.1	1000.3	1130.5	932.4	0.2348	1.6084
6	170.06	61.89	0.01616	137.9	995.8	1133.7	927.0	0.2471	1.5814
7	176.85	53.56	0.01867	144.7	991.8	1136.5	922.4	0.2579	1.5582
8	182.86	47.27	0.02115	150.8	988.2	1139.0	918.2	0.2673	1.5380
9	188.27	42.36	0.02361	156.2	985.0	1141.1	914.4	0.2756	1.5202
10	193.22	38.38	0.02606	161.1	982.0	1143.1	910.9	0.2832	1.5042
11	197.75	35.10	0.02849	165.7	979.2	1144.9	907.8	0.2902	1.4895
12	201.96	32.36	0.03090	169.9	976.6	1146.5	904.8	0.2967	1.4760
13	205.87	30.03	0.03330	173.8	974.2	1148.0	902.0	0.3025	1.4639
14	209.55	28.02	0.03569	177.5	971.9	1149.4	899.3	0.3081	1.4523
14.7	212.00	26.79	0.03732	180.0	970.4	1150.4	897.6	0.3118	1.4447
15	213.0	26.27	0.03806	181.0	969.7	1150.7	896.8	0.3139	1.4416
16	216.3	24.79	0.04042	184.4	967.6	1152.0	894.4	0.3183	1.4311
17	219.4	23.38	0.04277	187.5	965.6	1153.1	892.1	0.3229	1.4215
18	222.4	22.16	0.04512	190.5	963.7	1154.2	889.9	0.3273	1.4127
19	225.2	21.07	0.04746	193.4	961.8	1155.2	887.8	0.3315	1.4045
20	228.0	20.08	0.04980	196.1	960.0	1156.2	885.8	0.3355	1.3965
21	230.6	19.18	0.05213	198.8	958.3	1157.1	883.9	0.3393	1.3887
22	233.1	18.37	0.05445	201.3	956.7	1158.0	882.0	0.3430	1.3811
23	235.5	17.62	0.05676	203.8	955.1	1158.8	880.2	0.3465	1.3739
24	237.8	16.93	0.05907	206.1	953.5	1159.6	878.5	0.3499	1.3670
25	240.1	16.30	0.0614	208.4	952.0	1160.4	876.8	0.3532	1.3604
26	242.2	15.72	0.0636	210.6	950.6	1161.2	875.1	0.3564	1.3542
27	244.4	15.18	0.0659	212.7	949.2	1161.9	873.5	0.3594	1.3483
28	246.4	14.67	0.0682	214.8	947.8	1162.6	872.0	0.3623	1.3425
29	248.4	14.19	0.0705	216.8	946.4	1163.2	870.5	0.3652	1.3367
30	250.3	13.74	0.0728	218.8	945.1	1163.9	869.0	0.3680	1.3311
31	252.2	13.32	0.0751	220.7	943.8	1164.5	867.6	0.3707	1.3257
32	254.1	12.93	0.0773	222.6	942.5	1165.1	866.2	0.3733	1.3205
33	255.8	12.57	0.0795	224.4	941.3	1165.7	864.8	0.3759	1.3155
34	257.6	12.22	0.0818	226.2	940.1	1166.3	863.4	0.3784	1.3107
35	259.3	11.89	0.0841	227.9	938.9	1166.8	862.1	0.3808	1.3060
36	261.0	11.58	0.0863	229.6	937.7	1167.3	860.8	0.3832	1.3014
37	262.6	11.29	0.0886	231.3	936.6	1167.8	859.5	0.3855	1.2969
38	264.2	11.01	0.0908	232.9	935.5	1168.4	858.3	0.3877	1.2925
39	265.8	10.74	0.0931	234.5	934.4	1168.9	857.1	0.3899	1.2882
40	267.3	10.49	0.0953	236.1	933.3	1169.4	855.9	0.3920	1.2841
41	268.7	10.25	0.0976	237.6	932.2	1169.8	854.7	0.3941	1.2800
42	270.2	10.02	0.0998	239.1	931.2	1170.3	853.6	0.3962	1.2759
43	271.7	9.80	0.1020	240.5	930.2	1170.7	852.4	0.3982	1.2720
44	273.1	9.59	0.1043	242.0	929.2	1171.2	851.3	0.4002	1.2681
45	274.5	9.39	0.1065	243.4	928.2	1171.6	850.3	0.4021	1.2644
46	275.8	9.20	0.1087	244.8	927.2	1172.0	849.2	0.4040	1.2607
47	277.2	9.02	0.1109	246.1	926.3	1172.4	848.1	0.4059	1.2571
48	278.5	8.84	0.1131	247.5	925.3	1172.8	847.1	0.4077	1.2536
49	279.8	8.67	0.1153	248.8	924.4	1173.2	846.1	0.4095	1.2502

Table 31. Properties of Saturated Steam—(*continued*)

Condensed from the Steam Tables of Marks and Davis by permission of the publishers, Longmans, Green and Co.

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol. cu. ft. per lb., v''	Density, lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''	Internal energy (B.t.u.) evap., I	Entropy	
								Water, s'	Evap., r/T
50	281.0	8.51	0.1175	250.1	923.5	1173.6	845.0	0.4113	1.2468
51	282.3	8.35	0.1197	251.4	922.6	1174.0	844.0	0.4130	1.2435
52	283.5	8.20	0.1219	252.6	921.7	1174.3	843.1	0.4147	1.2402
53	284.7	8.05	0.1241	253.9	920.8	1174.7	842.1	0.4164	1.2370
54	285.9	7.91	0.1263	255.1	919.9	1175.0	841.1	0.4180	1.2339
55	287.1	7.78	0.1285	256.3	919.0	1175.4	840.2	0.4196	1.2309
56	288.2	7.65	0.1307	257.5	918.2	1175.7	839.3	0.4212	1.2278
57	289.4	7.52	0.1329	258.7	917.4	1176.0	838.3	0.4227	1.2248
58	290.5	7.40	0.1350	259.8	916.5	1176.4	837.4	0.4242	1.2218
59	291.6	7.28	0.1372	261.0	915.7	1176.7	836.5	0.4257	1.2189
60	292.7	7.17	0.1394	262.1	914.9	1177.0	835.6	0.4272	1.2160
61	293.8	7.06	0.1416	263.2	914.1	1177.3	834.8	0.4287	1.2132
62	294.9	6.95	0.1438	264.3	913.3	1177.6	833.9	0.4302	1.2104
63	295.9	6.85	0.1460	265.4	912.5	1177.9	833.1	0.4316	1.2077
64	297.0	6.75	0.1482	266.4	911.8	1178.2	832.2	0.4330	1.2050
65	298.0	6.65	0.1503	267.5	911.0	1178.5	831.4	0.4344	1.2024
66	299.0	6.56	0.1525	268.5	910.2	1178.8	830.5	0.4358	1.1998
67	300.0	6.47	0.1547	269.6	909.5	1179.0	829.7	0.4371	1.1972
68	301.0	6.38	0.1569	270.6	908.7	1179.3	828.9	0.4385	1.1946
69	302.0	6.29	0.1590	271.6	908.0	1179.6	828.1	0.4398	1.1921
70	302.9	6.20	0.1612	272.6	907.2	1179.8	827.3	0.4411	1.1896
71	303.9	6.12	0.1634	273.6	906.5	1180.1	826.5	0.4424	1.1872
72	304.8	6.04	0.1656	274.5	905.8	1180.4	825.8	0.4437	1.1848
73	305.8	5.96	0.1678	275.5	905.1	1180.6	825.0	0.4449	1.1825
74	306.7	5.89	0.1699	276.5	904.4	1180.9	824.2	0.4462	1.1801
75	307.6	5.81	0.1721	277.4	903.7	1181.1	823.5	0.4474	1.1778
76	308.5	5.74	0.1743	278.3	903.0	1181.4	822.7	0.4487	1.1755
77	309.4	5.67	0.1764	279.3	902.3	1181.6	822.0	0.4499	1.1732
78	310.3	5.60	0.1786	280.2	901.7	1181.8	821.3	0.4511	1.1710
79	311.2	5.54	0.1808	281.1	901.0	1182.1	820.6	0.4523	1.1687
80	312.0	5.47	0.1829	282.0	900.3	1182.3	819.8	0.4535	1.1665
81	312.9	5.41	0.1851	282.9	899.7	1182.5	819.1	0.4546	1.1644
82	313.8	5.34	0.1873	283.8	899.0	1182.8	818.4	0.4557	1.1623
83	314.6	5.28	0.1894	284.6	898.4	1183.0	817.7	0.4568	1.1602
84	315.4	5.22	0.1915	285.5	897.7	1183.2	817.0	0.4579	1.1581
85	316.3	5.16	0.1937	286.3	897.1	1183.4	816.3	0.4590	1.1561
86	317.1	5.10	0.1959	287.2	896.4	1183.6	815.6	0.4601	1.1540
87	317.9	5.05	0.1980	288.0	895.8	1183.8	815.0	0.4612	1.1520
88	318.7	5.00	0.2001	288.9	895.2	1184.0	814.3	0.4623	1.1500
89	319.5	4.94	0.2023	289.7	894.6	1184.2	813.6	0.4633	1.1481
90	320.3	4.89	0.2044	290.5	893.9	1184.4	813.0	0.4644	1.1461
91	321.1	4.84	0.2065	291.3	893.3	1184.6	812.3	0.4654	1.1442
92	321.8	4.79	0.2087	292.1	892.7	1184.8	811.7	0.4664	1.1423
93	322.6	4.74	0.2109	292.9	892.1	1185.0	811.0	0.4674	1.1404
94	323.4	4.69	0.2130	293.7	891.5	1185.2	810.4	0.4684	1.1385
95	324.1	4.65	0.2151	294.5	890.9	1185.4	809.7	0.4694	1.1367
96	324.9	4.60	0.2172	295.3	890.3	1185.6	809.1	0.4704	1.1348
97	325.6	4.56	0.2193	296.1	889.7	1185.8	808.5	0.4714	1.1330
98	326.4	4.51	0.2215	296.8	889.2	1186.0	807.9	0.4724	1.1312
99	327.1	4.47	0.2237	297.6	888.6	1186.2	807.2	0.4733	1.1295

Table 31. Properties of Saturated Steam—(*continued*)

Condensed from the Steam Tables of Marks and Davis by permission of the publishers, Longmans, Green and Co.

Abs. pres. lb.,	Temp., deg. fahr.,	Sp. vol., cu. ft. per lb.,	Density lb. per cu. ft.,	Heat of the liquid,	Latent heat of evap.,	Heat content of steam,	Internal energy (B.t.u.) evap.,	Entropy	
								Water,	Evap.,
p	t	v''	$1/v''$	i'	r	i''	i	s'	r/T
100	327.8	4.429	0.2258	298.3	888.0	1186.3	806.6	0.4743	1.1277
102	329.3	4.347	0.2300	299.8	886.9	1186.7	805.4	0.4762	1.1242
104	330.7	4.268	0.2343	301.3	885.8	1187.0	804.2	0.4780	1.1208
106	332.0	4.192	0.2386	302.7	884.7	1187.4	803.0	0.4798	1.1174
108	333.4	4.118	0.2429	304.1	883.6	1187.7	801.9	0.4816	1.1141
110	334.8	4.047	0.2472	305.5	882.5	1188.0	800.7	0.4834	1.1108
112	336.1	3.978	0.2514	306.9	881.4	1188.4	799.6	0.4852	1.1076
114	337.4	3.912	0.2556	308.3	880.4	1188.7	798.5	0.4869	1.1045
116	338.7	3.848	0.2599	309.6	879.3	1189.0	797.4	0.4886	1.1014
118	340.0	3.786	0.2641	311.0	878.3	1189.3	796.3	0.4903	1.0984
120	341.3	3.726	0.2683	312.3	877.2	1189.6	795.2	0.4919	1.0954
122	342.5	3.668	0.2726	313.6	876.2	1189.8	794.2	0.4935	1.0924
124	343.8	3.611	0.7769	314.9	875.2	1190.1	793.1	0.4951	1.0895
126	345.0	3.556	0.2812	316.2	874.2	1190.4	792.0	0.4967	1.0865
128	346.2	3.504	0.2854	317.4	873.3	1190.7	791.0	0.4982	1.0837
130	347.4	3.452	0.2897	318.6	872.3	1191.0	790.0	0.4998	1.0809
132	348.5	3.402	0.2939	319.9	871.3	1191.2	789.0	0.5013	1.0782
134	349.7	3.354	0.2981	321.1	870.4	1191.5	788.0	0.5028	1.0755
136	350.8	3.308	0.3023	322.3	869.4	1191.7	787.0	0.5043	1.0728
138	352.0	3.263	0.3065	323.4	868.5	1192.0	786.0	0.5057	1.0702
140	353.1	3.219	0.3107	324.6	867.6	1192.2	785.0	0.5072	1.0675
142	354.2	3.175	0.3150	325.8	866.7	1192.5	784.1	0.5086	1.0649
144	355.3	3.133	0.3192	326.9	865.8	1192.7	783.2	0.5100	1.0624
146	356.3	3.092	0.3234	328.0	864.9	1192.9	782.2	0.5114	1.0599
148	357.4	3.052	0.3276	329.1	864.0	1193.2	781.3	0.5128	1.0574
150	358.5	3.012	0.3320	330.2	863.2	1193.4	780.4	0.5142	1.0550
152	359.5	2.974	0.3362	331.4	862.3	1193.6	779.4	0.5155	1.0525
154	360.5	2.938	0.3404	332.4	861.4	1193.8	778.5	0.5169	1.0501
156	361.6	2.902	0.3446	333.5	860.6	1194.1	777.6	0.5182	1.0477
158	362.6	2.868	0.3488	334.6	859.7	1194.3	776.7	0.5195	1.0454
160	363.6	2.834	0.3529	335.6	858.8	1194.5	775.8	0.5208	1.0431
162	364.6	2.801	0.3570	336.7	858.0	1194.7	775.0	0.5220	1.0409
164	365.6	2.769	0.3612	337.7	857.2	1194.9	774.1	0.5233	1.0387
166	366.5	2.737	0.3654	338.7	856.4	1195.1	773.2	0.5245	1.0365
168	367.5	2.706	0.3696	339.7	855.5	1195.3	772.4	0.5257	1.0343
170	368.5	2.675	0.3738	340.7	854.7	1195.4	771.5	0.5269	1.0321
172	369.4	2.645	0.3780	341.7	853.9	1195.6	770.7	0.5281	1.0300
174	370.4	2.616	0.3822	342.7	853.1	1195.8	769.8	0.5293	1.0278
176	371.3	2.588	0.3864	343.7	852.3	1196.0	769.0	0.5305	1.0257
178	372.2	2.560	0.3906	344.7	851.5	1196.2	768.2	0.5317	1.0235
180	373.1	2.533	0.3948	345.6	850.8	1196.4	767.4	0.5328	1.0215
182	374.0	2.507	0.3989	346.6	850.0	1196.6	766.6	0.5339	1.0195
184	374.9	2.481	0.4031	347.6	849.2	1196.8	765.8	0.5351	1.0174
186	375.8	2.455	0.4073	348.5	848.4	1196.9	765.0	0.5362	1.0154
188	376.7	2.430	0.4115	349.4	847.7	1197.1	764.2	0.5373	1.0134
190	377.6	2.406	0.4157	350.4	846.9	1197.3	763.4	0.5384	1.0114
192	378.5	2.381	0.4199	351.3	846.1	1197.4	762.6	0.5395	1.0095
194	379.3	2.358	0.4241	352.2	845.4	1197.6	761.8	0.5405	1.0076
196	380.2	2.335	0.4283	353.1	844.7	1197.8	761.1	0.5416	1.0056
198	381.0	2.312	0.4325	354.0	843.9	1197.9	760.3	0.5426	1.0038

Table 31. Properties of Saturated Steam—(*continued*)

Condensed from the Steam Tables of Marks and Davis by permission of the publishers, Longmans, Green and Co.

Abs. pres., lb., p	Temp., deg. fahr., t	Sp. vol., cu. ft. per lb., v''	Density lb. per cu. ft., $1/v''$	Heat of the liquid, i'	Latent heat of evap., r	Heat content of steam, i''	Internal energy (B.t.u.) evap., I	Entropy: Water, s'	Entropy: Evap., r/T
200	381.9	2.290	0.437	354.9	843.2	1198.1	759.5	0.5437	1.0019
205	384.0	2.237	0.447	357.1	841.4	1198.5	757.6	0.5463	0.9973
210	386.0	2.187	0.457	359.2	839.6	1198.8	755.8	0.5488	0.9928
215	388.0	2.138	0.468	361.4	837.9	1199.2	754.0	0.5513	0.9885
220	389.9	2.091	0.478	363.4	836.2	1199.6	752.3	0.5538	0.9841
225	391.9	2.046	0.489	365.5	834.4	1199.9	750.5	0.5562	0.9799
230	393.8	2.004	0.499	367.5	832.8	1200.2	748.8	0.5586	0.9758
235	395.6	1.964	0.509	369.4	831.1	1200.6	747.0	0.5610	0.9717
240	397.4	1.924	0.520	371.4	829.5	1200.9	745.4	0.5633	0.9676
245	399.3	1.887	0.530	373.3	827.9	1201.2	743.7	0.5655	0.9638
250	401.1	1.850	0.541	375.2	826.3	1201.5	742.0	0.5676	0.9600
260	404.5	1.782	0.561	378.9	823.1	1202.1	738.9	0.5719	0.9525
270	407.9	1.718	0.582	382.5	820.1	1202.6	735.8	0.5760	0.9454
280	411.2	1.658	0.603	386.0	817.1	1203.1	732.7	0.5800	0.9385
290	414.4	1.602	0.624	389.4	814.2	1203.6	729.7	0.5840	0.9316
300	417.5	1.551	0.645	392.7	811.3	1204.1	726.8	0.5878	0.9251
350	431.9	1.334	0.750	408.2	797.8	1206.1	713.3	0.6053	0.8949
400	444.8	1.17	0.86	422.0	786.0	1208.0	701.0	0.621	0.868
450	456.5	1.04	0.96	435.0	774.0	1209.0	690.0	0.635	0.844
500	467.3	0.93	1.08	448.0	762.0	1210.0	678.0	0.648	0.822
660	486.6	0.76	1.32	469.0	741.0	1210.0	658.0	0.670	0.783

The **relation between pressure and temperature** is satisfactorily represented by Marks's equation

$$\log p = 10.515354 - (4873.71/T) - 0.00405096T + 0.0000013929647^2.$$

For the **heat content i'' of saturated steam** Davis has deduced the equation

$$i'' = 1150.3 + 0.3745(t - 212) - 0.00055(t - 212)^2.$$

This applies to the range 212 deg. to 400 deg. fahr.

The **heat content i' of water between 32 deg. and 212 deg. fahr.** is deduced from the experiments of Barnes and Dieterici, and above 212 deg. from the experiments of Dieterici and Regnault. With these fundamental data the remaining properties are readily calculated. For an exhaustive discussion of sources and methods see Marks and Davis, "Steam Tables and Diagrams," pp. 87–106. For density and specific heat of water at saturation pressures see Table 4.

For **condenser calculations** a more detailed table of the properties of saturated steam at temperatures from 50 deg. to 130 deg. fahr. is given in Table 32, which is calculated from the Marks and Davis tables. Vacuums are stated with reference to a 30-in. barometer; the standard atmosphere is equal to 30 in. of mercury when the mercury is at 58.4 deg. fahr. Absolute pressures are given in lb. per sq. in. and also in in. of mercury at 32 deg. fahr.

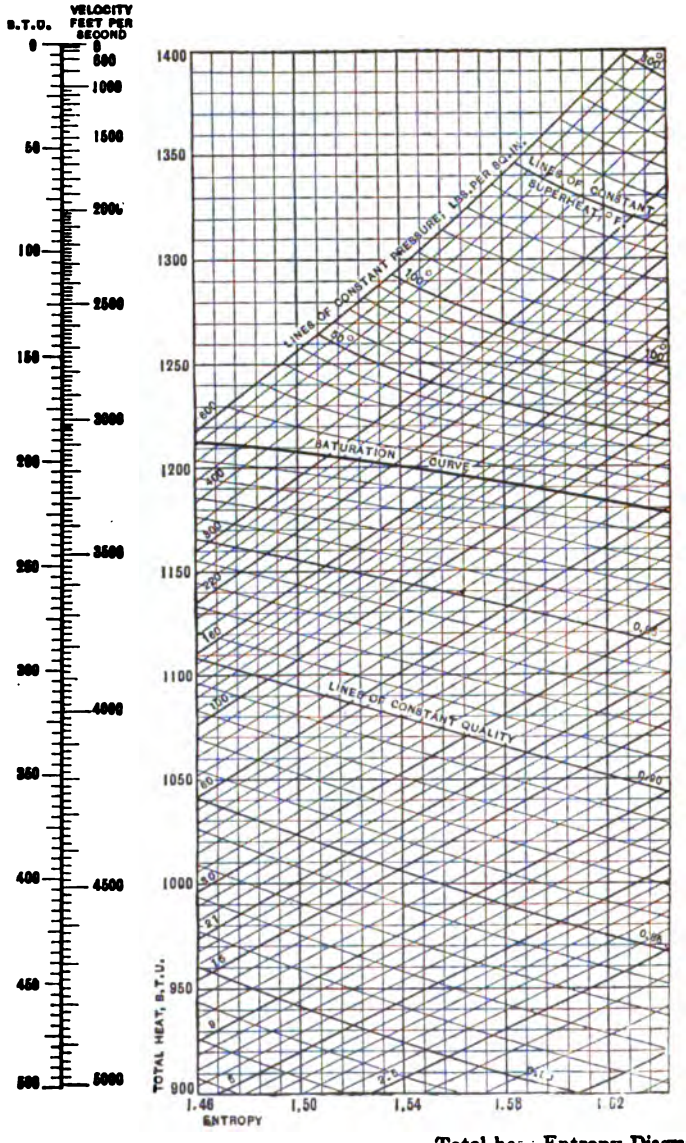
Properties of Superheated Steam. An experimental basis for the properties of superheated steam is furnished by the researches carried on in the Munich laboratory. Volume measurements have been given by Knoblauch, Linde, and Klebe; specific-heat measurements by Knoblauch and

Jakob and by Knoblauch and Mollier. These two sets of measurements are connected by the Clausius thermodynamic relation

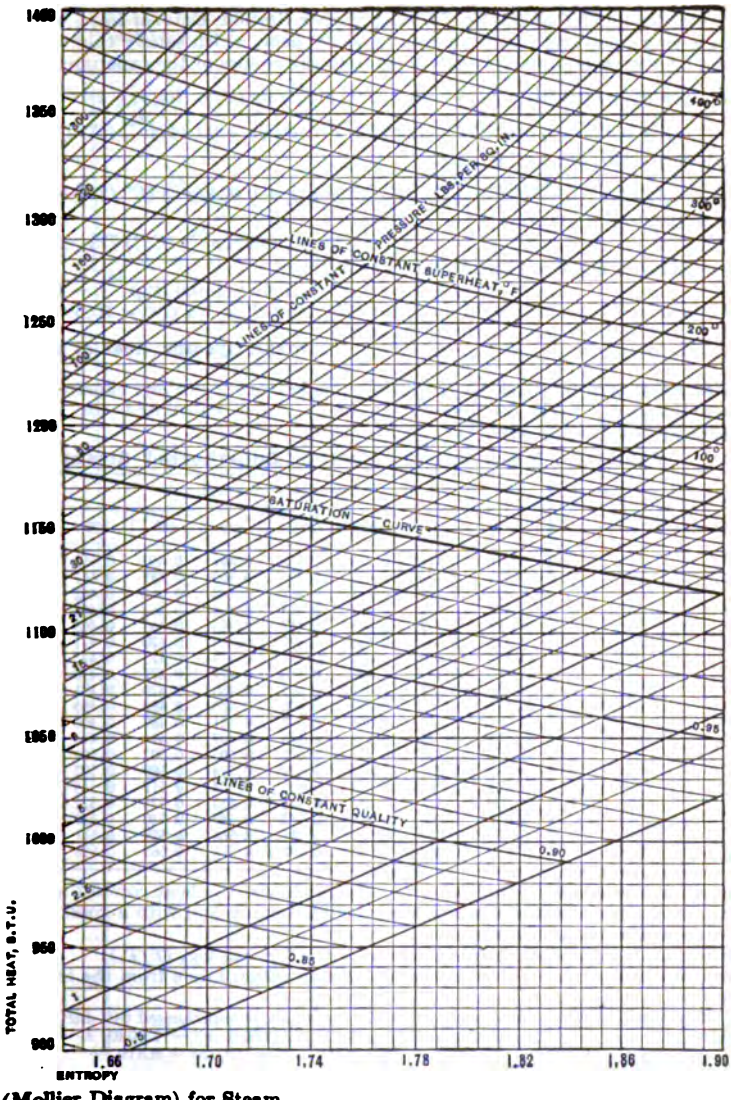
$$\left(\frac{\partial c_p}{\partial p}\right)_T = -AT\frac{\partial^2 v}{\partial T^2}.$$ (See p. 198.)

Table 22. Steam Table for Use in Condenser Calculations
(Calculated from the Steam Tables of Marks and Davis)

Temperature, deg. fahr. t	Vacuum in in. of mercury referred to a 30 in. bar. (Mercury at 58.4 deg. fahr.)	Pressure, lb. per sq. in. absolute p	Pressure, inches of mercury, with mercury at 32 deg. fahr.	Specific volume, cu. ft. per lb. v''	Heat of the liquid i'	Total heat of the steam i''	Internal energy of evaporation l	Entropy of water s'	Entropy of steam s''
50	29.637	0.1780	0.363	1702.0	18.08	1081.4	1007.3	0.0361	2.1226
52	29.609	0.1917	0.390	1586.0	20.08	1082.3	1006.0	0.0401	2.1164
54	29.579	0.2063	0.420	1480.0	22.08	1083.2	1004.6	0.0440	2.1100
56	29.547	0.2219	0.452	1381.0	24.08	1084.1	1003.3	0.0478	2.1037
58	29.513	0.2385	0.486	1291.0	26.08	1085.0	1002.0	0.0517	2.0975
60	29.477	0.2562	0.522	1208.0	28.08	1085.9	1000.7	0.0555	2.0913
62	29.439	0.2749	0.560	1130.0	30.08	1086.8	999.3	0.0593	2.0851
64	29.398	0.2949	0.601	1058.0	32.07	1087.6	998.0	0.0631	2.0791
66	29.354	0.3161	0.644	991.0	34.07	1088.5	996.7	0.0669	2.0731
68	29.308	0.3386	0.690	928.0	36.07	1089.4	995.4	0.0707	2.0672
70	29.259	0.3626	0.739	871.0	38.06	1090.3	994.0	0.0745	2.0613
72	29.208	0.3880	0.790	817.0	40.05	1091.2	992.7	0.0783	2.0556
74	29.153	0.4148	0.845	767.0	42.05	1092.1	991.4	0.0821	2.0499
76	29.095	0.4432	0.903	720.0	44.04	1093.0	990.1	0.0858	2.0443
78	29.034	0.4735	0.964	677.0	46.04	1093.9	988.7	0.0895	2.0386
80	28.968	0.505	1.029	636.8	48.03	1094.8	987.4	0.0932	2.0330
82	28.899	0.539	1.098	598.7	50.03	1095.6	986.1	0.0969	2.0275
84	28.826	0.575	1.171	562.9	52.02	1096.5	984.8	0.1005	2.0220
86	28.749	0.613	1.248	529.5	54.01	1097.4	983.4	0.1041	2.0165
88	28.666	0.654	1.331	498.4	56.01	1098.3	982.1	0.1078	2.0112
90	28.580	0.696	1.417	469.3	58.00	1099.2	980.8	0.1114	2.0058
92	28.489	0.741	1.508	442.2	60.00	1100.1	979.4	0.1151	2.0007
94	28.392	0.789	1.605	417.0	61.99	1101.0	978.1	0.1187	1.9954
96	28.290	0.838	1.706	393.4	63.98	1101.8	976.8	0.1223	1.9903
98	28.183	0.891	1.813	371.4	65.98	1102.8	975.5	0.1259	1.9851
100	28.070	0.946	1.926	350.8	67.97	1103.6	974.1	0.1295	1.9800
102	27.951	1.005	2.045	331.5	69.96	1104.5	972.8	0.1330	1.9750
104	27.825	1.066	2.171	313.3	71.96	1105.3	971.5	0.1365	1.9700
106	27.692	1.131	2.303	296.4	73.95	1106.2	970.1	0.1401	1.9651
108	27.550	1.199	2.443	280.5	75.95	1107.1	968.8	0.1436	1.9602
110	27.404	1.271	2.589	265.5	77.94	1108.0	967.5	0.1471	1.9553
112	27.250	1.346	2.740	251.4	79.93	1108.8	966.2	0.1506	1.9506
114	27.088	1.426	2.904	238.2	81.93	1109.7	964.8	0.1541	1.9458
116	26.919	1.509	3.073	225.8	83.92	1110.6	963.5	0.1576	1.9412
118	26.739	1.597	3.252	214.1	85.92	1111.5	962.2	0.1611	1.9366
120	26.553	1.689	3.438	203.1	87.91	1112.3	960.8	0.1645	1.9319
122	26.355	1.785	3.635	192.8	89.91	1113.2	959.5	0.1679	1.9273
124	26.149	1.886	3.841	183.1	91.90	1114.1	958.2	0.1713	1.9228
126	25.931	1.992	4.057	173.9	93.90	1115.0	956.8	0.1747	1.9183
128	25.706	2.103	4.282	165.3	95.89	1115.8	955.5	0.1781	1.9139
130	25.48	2.219	4.52	157.1	97.89	1116.7	954.1	0.1816	1.9095



Total-heat-Entropy Diagram



(Mollier Diagram) for Steam.

Goodenough (*Bulletin* No. 75, Engr. Exper. Station, Univ. of Ill.) gives the following equations for volume and specific heat, which satisfy this relation and at the same time represent accurately the results of the experiments:

$$v - 0.017 = 0.59465\,(T/p) - (1 + 0.0513p^{\frac{1}{2}})\,(m/T^4). \quad \log m = 10.82500.$$

$$c_p = 0.320 + 0.000126T + (23{,}583/T^2) + p(1 + 0.0342p^{\frac{1}{2}})\,(C'/T^5),$$

where p = lb. per sq. in. Also

$$i = 0.320T + 0.000063T^2 - (23{,}583/T) - p(1 + 0.0342p^{\frac{1}{2}})\,(C''/T^4) + 948.7.$$

$$u = 0.2099T + 0.000063T^2 - (23{,}583/T) - p(1 + 0.02992p^{\frac{1}{2}})\,(C'''/T^4) + 948.7.$$

$$s = 0.73683 \log T + 0.000126T - (11{,}792/T^2) - 0.254 \log p - (C^{iv}p/T^5)\,(1 + 0.0342p^{\frac{1}{2}}) - 0.0809.$$

$$\log C' = 11.39361; \quad \log C'' = 10.79155; \quad \log C''' = \log C^{iv} = 10.69464.$$

The quotient $\int_{t_s}^{t} c_p dt/(t - t_s)$ gives the **mean specific heat of superheated steam** between the saturation temperature t_s and any chosen higher temperature. Table 33 gives values of c_{pm} thus calculated for various pressures and superheats.

Table 33. Mean Specific Heat of Superheated Steam

Pressure, lb. per sq. in.	Superheat, deg. fahr.							
	0	50	100	200	300	400	500	600
1	0.476	0.471	0.466	0.462	0.461	0.460	0.462	0.465
2	0.474	0.470	0.465	0.461	0.460	0.459	0.461	0.464
3	0.475	0.470	0.465	0.461	0.460	0.461	0.462	0.465
5	0.477	0.472	0.467	0.463	0.461	0.462	0.464	0.467
10	0.483	0.477	0.471	0.467	0.465	0.466	0.468	0.470
15	0.490	0.483	0.476	0.471	0.469	0.470	0.471	0.474
20	0.496	0.488	0.480	0.475	0.473	0.473	0.475	0.477
25	0.500	0.493	0.486	0.480	0.477	0.477	0.478	0.480
30	0.506	0.498	0.490	0.483	0.480	0.479	0.481	0.483
40	0.515	0.506	0.498	0.490	0.486	0.484	0.485	0.487
50	0.524	0.515	0.506	0.497	0.491	0.489	0.490	0.491
60	0.534	0.523	0.513	0.503	0.497	0.494	0.494	0.495
80	0.550	0.536	0.525	0.512	0.505	0.502	0.501	0.501
100	0.566	0.550	0.538	0.523	0.514	0.509	0.507	0.507
125	0.582	0.566	0.553	0.535	0.524	0.517	0.515	0.514
150	0.600	0.581	0.566	0.546	0.533	0.525	0.522	0.521
200	0.627	0.606	0.588	0.564	0.549	0.540	0.535	0.533
250	0.653	0.629	0.609	0.581	0.564	0.553	0.547	0.543
300	0.676	0.650	0.627	0.597	0.578	0.565	0.558	0.553

With the aid of the table of mean specific heats, the heat content and entropy of superheated steam may be calculated easily and with a good degree of approximation. Thus,

$$i = i'' + c_{pm}(t - t_s), \qquad s = s'' + c_{pm} \log_e (T/T_s),$$

where i'' and s'' [$= s' + (r/T)$] are saturation values taken from the table on p. 209.

Example. Steam at a pressure of 100 lb. per sq. in. is superheated 240 deg. From Table 33, $c_{pm} = 0.519$, and from Table 31, $i'' = 1186.3$, $s'' = 1.6020$, $t_s = 327.8$. Hence, $i = 1186.3 + 0.519 \times 240 = 1310.9$ B.t.u., $s = 1.6020 + 0.519 \log_e [(567.8 + 459.6)/(327.8 + 459.6)] = 1.7402$.

Table 34 gives properties of superheated steam condensed from the steam tables of Marks and Davis. A Mollier diagram for steam is given on pp. 214 and 215.

Table 34. Properties of Superheated Steam

(Condensed from Marks and Davis's "Steam Tables and Diagrams")

v = Specific volume in cu. ft. per lb.; i = Total heat in B.t.u. per lb.; s = Entropy

Pressure, lb. per sq. in. abs.	Temp. of saturated steam, deg. fahr.		Superheat, deg. fahr.								
			20	40	60	80	100	150	200	250	300
20	228.0	v	20.73	21.37	22.01	22.63	23.25	24.80	26.33	27.85	29.37
		i	1165.7	1175.2	1184.6	1194.1	1203.5	1227.1	1250.6	1274.1	1297.6
		s	1.7456	1.7587	1.7716	1.7840	1.7961	1.8251	1.8524	1.8781	1.9026
40	267.3	v	10.83	11.16	11.50	11.82	12.13	12.93	13.70	14.48	15.25
		i	1179.3	1189.1	1198.9	1208.7	1218.4	1242.4	1266.4	1290.3	1314.1
		s	1.6895	1.7025	1.7151	1.7273	1.7392	1.7674	1.7940	1.8189	1.8427
60	292.7	v	7.40	7.63	7.86	8.08	8.30	8.84	9.36	9.89	10.41
		i	1187.3	1197.5	1207.7	1217.7	1227.6	1252.1	1276.4	1300.4	1324.3
		s	1.6568	1.6698	1.6823	1.6944	1.7062	1.7342	1.7603	1.7849	1.8081
80	312.0	v	5.65	5.83	6.00	6.18	6.34	6.75	7.17	7.56	7.95
		i	1193.0	1203.6	1214.0	1224.2	1234.3	1259.0	1283.6	1307.8	1331.9
		s	1.6338	1.6469	1.6594	1.6716	1.6833	1.7110	1.7368	1.7612	1.7840
100	327.8	v	4.58	4.72	4.86	5.00	5.14	5.47	5.80	6.12	6.44
		i	1197.5	1208.4	1219.1	1229.5	1239.7	1264.7	1289.4	1313.6	1337.8
		s	1.6160	1.6294	1.6420	1.6541	1.6658	1.6933	1.7188	1.7428	1.7656
120	341.3	v	3.85	3.98	4.10	4.22	4.33	4.62	4.89	5.17	5.44
		i	1201.1	1212.4	1223.3	1233.8	1244.1	1269.3	1294.1	1318.4	1342.7
		s	1.6016	1.6152	1.6279	1.6400	1.6517	1.6789	1.7041	1.7280	1.7505
140	353.1	v	3.32	3.44	3.54	3.65	3.75	4.00	4.24	4.48	4.71
		i	1204.3	1215.8	1226.8	1237.5	1248.0	1273.3	1298.2	1322.6	1346.9
		s	1.5894	1.6031	1.6159	1.6280	1.6395	1.6666	1.6916	1.7152	1.7376
160	363.6	v	2.93	3.03	3.12	3.21	3.30	3.53	3.74	3.95	4.15
		i	1207.0	1218.8	1230.0	1240.8	1251.3	1276.8	1301.7	1326.2	1350.6
		s	1.5789	1.5928	1.6056	1.6177	1.6292	1.6561	1.6810	1.7043	1.7266
180	373.1	v	2.62	2.71	2.80	2.88	2.96	3.16	3.35	3.54	3.72
		i	1209.4	1221.5	1232.8	1243.8	1254.3	1279.9	1304.8	1329.5	1353.9
		s	1.5697	1.5838	1.5767	1.6088	1.6201	1.6468	1.6716	1.6948	1.7169
200	381.9	v	2.37	2.45	2.53	2.61	2.68	2.86	3.04	3.21	3.38
		i	1211.6	1223.9	1235.5	1246.5	1257.1	1282.6	1307.7	1332.4	1357.0
		s	1.5614	1.5757	1.5886	1.6007	1612.0	1.6385	1.6632	1.6862	1.7082
220	389.9	v	2.16	2.24	2.31	2.38	2.45	2.62	2.78	2.94	3.10
		i	1213.6	1226.2	1237.9	1248.9	1259.6	1285.2	1310.3	1335.1	1359.8
		s	1.5541	1.5686	1.5816	1.5936	1.6049	1.6312	1.6558	1.6787	1.7005
240	397.4	v	1.99	2.06	2.13	2.20	2.26	2.42	2.57	2.71	2.85
		i	1215.4	1228.3	1240.1	1251.3	1261.9	1287.6	1312.8	1337.6	1362.3
		s	1.5476	1.5623	1.5753	1.5873	1.5985	1.6246	1.6492	1.6720	1.6937
260	404.5	v	1.84	1.91	1.97	2.04	2.10	2.24	2.39	2.52	2.65
		i	1217.1	1230.3	1242.3	1253.4	1264.1	1289.9	1315.1	1340.0	1364.7
		s	1.5416	1.5564	1.5695	1.5815	1.5926	1.6186	1.6430	1.6658	1.6874
280	411.2	v	1.72	1.78	1.84	1.90	1.95	2.09	2.22	2.35	2.48
		i	1218.7	1232.2	1244.3	1255.5	1266.2	1291.9	1317.2	1342.2	1367.0
		s	1.5362	1.5512	1.5643	1.5762	1.5873	1.6133	1.6375	1.6603	1.6818
300	417.5	v	1.60	1.66	1.72	1.78	1.83	1.96	2.09	2.21	2.33
		i	1220.2	1234.1	1246.2	1257.4	1268.2	1294.0	1319.3	1344.3	1369.2
		s	1.5310	1.5462	1.5594	1.5713	1.5824	1.6082	1.6323	1.6550	1.6765

Ammonia Vapor. The tables of Goodenough and Mosher (*Bulletin* No. 66, Engineering Exper. Station, University of Illinois), represent the latest and probably the most accurate values of the thermal properties of saturated vapor of ammonia. See Table 35.

Table 35. Properties of Ammonia

Temp., deg. fahr.	Pressure, lb. per sq. in., abs.	Specific volume		Heat content		Heat of vaporization	Internal energy of vaporization	Entropy	
		of liquid, cu. ft. per lb.	of sat. vapor, cu. ft. per lb.	of liquid	of sat. vapor			of liquid	of vaporization
t	p	v'	v''	i'	i''	r	l	s'	r/T
−40	10.12	0.0234	25.45	−75.3	526.6	601.9	554.2	−0.1653	1.4343
−35	11.74	0.0235	22.14	−70.2	528.2	598.3	550.2	−0.1531	1.4090
−30	13.56	0.0236	19.35	−65.0	529.8	594.7	546.2	−0.1410	1.3842
−25	15.61	0.0238	16.95	−59.8	531.3	591.1	542.1	−0.1290	1.3598
−20	17.91	0.0239	14.89	−54.6	532.8	587.4	538.0	−0.1171	1.3360
−15	20.46	0.0240	13.15	−49.4	534.3	583.6	533.9	−0.1054	1.3126
−10	23.30	0.0241	11.63	−44.2	535.7	579.9	529.8	−0.0938	1.2896
− 5	26.46	0.0242	10.32	−38.9	537.1	576.1	525.6	−0.0824	1.2671
0	29.95	0.0244	9.19	−33.7	538.5	572.2	521.4	−0.0709	1.2449
5	33.79	0.0245	8.20	−28.4	539.9	568.3	517.1	−0.0595	1.2231
10	38.02	0.0246	7.34	−23.2	541.2	564.4	512.9	−0.0483	1.2017
15	42.67	0.0248	6.583	−17.9	542.5	560.4	508.6	−0.0372	1.1806
20	47.75	0.0249	5.920	−12.6	543.7	556.3	504.2	−0.0262	1.1599
25	53.30	0.0250	5.336	− 7.3	545.0	552.2	499.8	−0.0153	1.1395
30	59.35	0.0252	4.820	− 1.9	546.2	548.1	495.4	−0.0044	1.1194
35	65.91	0.0253	4.364	+ 3.5	547.4	543.9	491.0	+0.0065	1.0996
40	73.03	0.0255	3.959	8.9	548.5	539.7	486.5	0.0173	1.0801
45	80.75	0.0256	3.599	14.3	549.7	535.3	481.9	0.0280	1.0609
50	89.09	0.0258	3.278	19.8	550.8	531.0	477.3	0.0387	1.0419
55	98.03	0.0259	2.992	25.3	551.9	526.5	472.7	0.0494	1.0231
60	107.7	0.0261	2.734	30.9	552.9	522.0	468.0	0.0601	1.0046
65	118.1	0.0263	2.503	36.5	554.0	517.5	463.3	0.0708	0.9863
70	129.2	0.0264	2.296	42.1	555.0	512.8	458.5	0.0813	0.9683
75	141.1	0.0266	2.109	47.8	556.0	508.1	453.7	0.0919	0.9504
80	153.9	0.0268	1.940	53.6	557.0	503.4	448.8	0.1025	0.9328
85	167.4	0.0270	1.788	59.4	557.9	498.5	443.9	0.1132	0.9153
90	181.8	0.0271	1.650	65.3	558.9	493.5	438.9	0.1238	0.8980
95	197.3	0.0273	1.524	71.3	559.8	488.5	433.9	0.1344	0.8808
100	213.8	0.0275	1.408	77.3	560.7	483.4	428.7	0.1450	0.8638
105	231.2	0.0277	1.305	83.4	561.6	478.2	423.5	0.1557	0.8469
110	249.6	0.0280	1.210	89.6	562.5	472.9	418.3	0.1664	0.8302
115	269.2	0.0282	1.122	95.9	563.3	467.4	412.9	0.1772	0.8135
120	289.9	0.0284	1.042	102.2	564.2	461.9	407.5	0.1881	0.7969
125	311.6	0.0286	0.970	108.7	565.0	456.3	402.0	0.1990	0.7805

The relation between pressure and temperature is derived from the vapor-pressure law, using as a basis the Marks equation (p. 212) connecting the pressure and temperature of saturated steam. For a given pressure p, let T_w denote the saturation temperature (absolute) of water vapor and T_a the corresponding temperature of ammonia vapor. Then

$$T_a = 1/[(1.70356/T_w) - 0.0002242].$$

Below 160 deg. fahr. the volume v' of liquid ammonia is given by

$$v' = 0.06335 - 0.016 \log (273.2 - t).$$

The difference $v'' - v'$ is found from the Clapeyron relation (p. 208), and from this v'', the specific volume of the saturated vapor, is determined. The latent heat is calculated from the empirical formula

$$\log r = 1.856064 + 0.37 \log (273.2 - t).$$

Superheated Ammonia. The specific volume of superheated ammonia is given by the characteristic equation of Mosher, viz.:

$$v + 0.10 = 0.6321\,(T/p) - (79{,}433 \times 10^6/T^5).$$

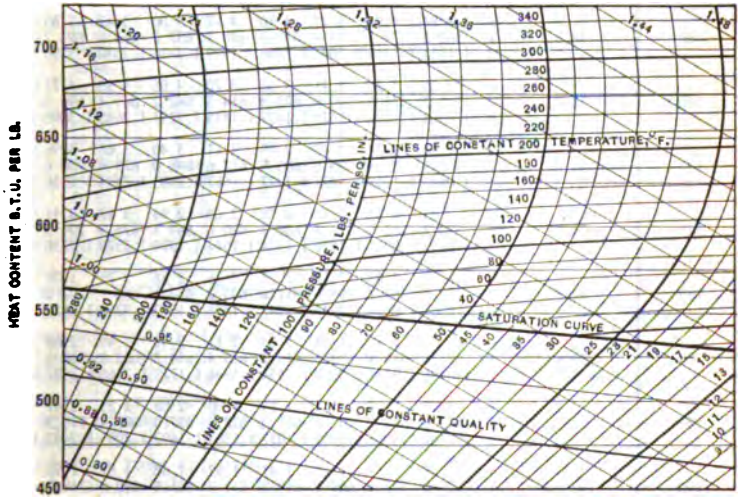
The specific heat at constant pressure is given by the empirical formula

$$c_p = 0.382 + 0.000174T + (Cp/T^6). \quad \log C = 13.644705.$$

For the heat content and entropy the following formulæ are deduced from the preceding:

$$i = 0.382T + 0.000087T^2 - (C'p/T^5) - 0.0185p + 358. \quad \log C' = 12.945734.$$

$$s = 0.8796 \log T + 0.000174T - 0.2695 \log p - (C''p/T^6) - 0.8266.$$
$$\log C'' = 12.866554.$$



Total-heat-Entropy Diagram (Mollier Diagram) for Ammonia.

Table 36 gives these properties of superheated NH_3 for various pressures and degrees of superheat. A Mollier diagram for ammonia is given above. In using this diagram it should be noted that the constant-entropy lines are diagonals (not verticals), and that adiabatics have the same slope as these diagonals.

Sulphur Dioxide (SO_2). The properties of saturated vapor of SO_2 given in Table 37 are based on the researches of Cailletet and Mathias. Certain of these properties are given by the following empirical formulæ:

$$c' = 0.3194 + 0.00065(t - 32). \quad v' = 0.0113.$$

$$s' = 0.00065(t - 32). \quad r/T = 0.3327 - 0.00129(t - 32).$$

Table 36. Properties of Superheated Ammonia

(Condensed from the Tables of Goodenough and Mosher)

v = Specific volume in cubic feet per pound; i = Total heat in B.t.u. per pound; s = Entropy

Pressure, lb. per sq. in. Abs.	Temp. of saturated vapor, deg. fahr.		Superheat, deg. fahr.								
			20	40	60	80	100	150	200	250	300
20	−15.9	v	14.2	14.9	15.6	16.3	16.9	18.6	20.2	21.8	23.4
		i	545.2	556.1	566.7	577.1	587.4	612.7	638.0	663.4	689.1
		s	1.2339	1.2569	1.2784	1.2986	1.3178	1.3625	1.4033	1.4413	1.4771
40	12.2	v	7.40	7.76	8.12	8.46	8.80	9.64	10.46	11.26	12.07
		i	553.9	565.5	576.8	587.7	598.3	624.6	650.6	676.6	702.7
		s	1.1742	1.1974	1.2188	1.2390	1.2581	1.3021	1.3422	1.3795	1.4145
50	30.5	v	5.04	5.30	5.54	5.77	6.01	6.57	7.12	7.66	8.20
		i	559.1	571.2	582.9	594.2	605.2	632.1	658.6	684.9	711.3
		s	1.1396	1.1629	1.1846	1.2047	1.2238	1.2676	1.3074	1.3442	1.3787
80	44.5	v	3.84	4.04	4.22	4.40	4.57	5.00	5.42	5.83	6.23
		i	562.8	574.3	587.4	599.0	610.3	637.7	664.6	691.2	717.9
		s	1.1154	1.1389	1.1606	1.1808	1.1999	1.2436	1.2830	1.3196	1.3538
100	56.0	v	3.10	3.26	3.41	3.56	3.70	4.05	4.38	4.71	5.03
		i	565.7	578.6	590.9	602.8	614.3	642.2	669.4	696.3	723.2
		s	1.0968	1.1204	1.1422	1.1625	1.1815	1.2251	1.2645	1.3009	1.3348
120	65.8	v	2.61	2.74	2.87	2.99	3.11	3.40	3.68	3.96	4.23
		i	568.2	581.4	593.9	605.9	617.6	646.0	673.4	700.6	727.6
		s	1.0818	1.1055	1.1274	1.1477	1.1667	1.2103	1.2494	1.2856	1.3195
140	74.5	v	2.25	2.37	2.48	2.58	2.69	2.94	3.17	3.41	3.65
		i	570.2	583.6	596.4	608.7	620.6	649.3	676.9	704.3	731.5
		s	1.0693	1.0931	1.1150	1.1354	1.1544	1.1979	1.2370	1.2730	1.3067
160	82.3	v	1.98	2.08	2.18	2.27	2.36	2.58	2.79	3.00	3.21
		i	572.0	589.7	598.6	611.1	623.2	652.1	680.1	707.6	735.0
		s	1.0585	1.0824	1.1043	1.1247	1.1438	1.1872	1.2263	1.2622	1.2957
180	89.4	v	1.77	1.86	1.94	2.03	2.11	2.31	2.49	2.68	2.86
		i	573.6	587.5	600.6	613.2	625.4	654.7	682.8	710.5	738.1
		s	1.0491	1.0731	1.0950	1.1154	1.1346	1.1779	1.2168	1.2527	1.2861
200	95.9	v	1.59	1.67	1.75	1.83	1.90	2.08	2.25	2.41	2.57
		i	575.0	589.1	602.4	615.2	627.5	657.0	685.4	713.3	741.0
		s	1.0407	1.0648	1.0868	1.1072	1.1263	1.1697	1.2085	1.2442	1.2776
250	110.1	v	1.28	1.35	1.41	1.47	1.53	1.68	1.81	1.95	2.08
		i	578.0	592.4	606.1	619.3	631.9	661.9	690.8	719.0	747.0
		s	1.0232	1.0474	1.0695	1.0899	1.1090	1.1524	1.1911	1.2266	1.2598
300	122.4	v	1.07	1.13	1.18	1.23	1.28	1.41	1.52	1.63	1.74
		i	580.5	595.3	609.3	622.7	635.5	666.1	695.4	723.9	752.1
		s	1.0093	1.0335	1.0556	1.0761	1.0953	1.1386	1.1772	1.2125	1.2455

Carbon Dioxide (CO_2). The properties of CO_2 vapor have been investigated experimentally by Amagat. The results are given in Table 38. The following are the empirical formulæ devised by Mollier:

Critical temperature, T_k = 547.83 deg. fahr. absolute.

$$p = 42.2056\,[(T/180) - 1]^{4.526} \qquad r = 0.68167^{0.43}(T_k - T)^{0.43}$$

$$c' = 0.000185T + 0.285\,(r/T) + 0.215\,r/(T_k - T)$$

$$s' = 0.10155 + 0.000185(t - 32) - r/2T$$

Table 37. Properties of Sulphur Dioxide

Temp., deg. fahr. t	Pressure, lb. per sq. in. p	Specific volume of sat. vapor v''	Heat content of liquid t'	Heat content of vapor t''	Heat of vaporisation r	Internal energy of vaporisation l	Entropy of liquid s'	Entropy of vaporisation r/T
−25	5.02	16.250	−17.14	159.32	176.46	163.10	−0.0368	0.4057
−20	5.92	12.980	−15.69	159.94	175.63	162.04	−0.0338	0.3996
−15	6.86	10.910	−14.28	160.52	174.80	161.04	−0.0308	0.3935
−10	7.90	9.510	−12.85	161.02	173.87	160.00	−0.0274	0.3869
− 5	9.03	8.410	−11.39	161.50	172.89	158.90	−0.0240	0.3803
0	10.35	7.490	− 9.907	161.92	171.83	157.70	−0.0208	0.3739
5	11.80	6.660	− 8.388	162.29	170.68	156.43	−0.0176	0.3675
10	13.40	5.910	− 6.842	162.59	169.43	154.95	−0.0141	0.3608
15	15.15	5.210	− 5.321	162.87	168.19	153.59	−0.0110	0.3546
20	17.09	4.635	− 3.780	163.12	166.90	152.26	−0.0079	0.3483
25	19.17	4.135	− 2.212	163.30	165.51	150.84	−0.0046	0.3418
30	21.46	3.716	− 0.622	163.43	164.05	149.36	−0.0012	0.3351
35	24.04	3.350	0.968	163.51	162.54	147.75	0.0020	0.3287
40	26.82	3.005	2.565	163.55	160.99	146.11	0.0051	0.3224
45	29.90	2.700	4.156	163.53	159.37	144.40	0.0083	0.3160
50	33.38	2.433	5.850	163.45	157.60	142.68	0.0117	0.3094
55	36.96	2.210	7.510	163.33	155.82	140.92	0.0150	0.3029
60	40.53	2.003	9.203	163.19	153.99	138.95	0.0182	0.2965
65	44.92	1.817	10.900	162.99	152.09	137.12	0.0215	0.2900
70	49.56	1.650	12.580	162.73	150.15	135.29	0.0248	0.2835
75	54.33	1.499	14.310	162.44	148.13	133.28	0.0280	0.2771
80	59.58	1.362	16.090	162.09	146.00	131.16	0.0312	0.2707
85	65.25	1.238	17.830	161.70	143.87	129.04	0.0345	0.2642
90	71.25	1.137	19.640	161.25	141.61	126.85	0.0377	0.2578
95	77.66	1.041	21.420	160.74	139.32	124.63	0.0410	0.2513
100	84.75	0.950	23.170	160.19	137.02	122.39	0.0443	0.2442
105	92.18	0.867	24.960	159.60	134.64	120.02	0.0475	0.2380

Properties of Other Refrigerating Fluids. In addition to the three fluids whose properties have been given in the preceding tables, the following fluids have been used to some extent:

Ethyl chloride (C_2H_5Cl); methyl chloride (CH_3Cl); nitrous oxide (N_2O) and Pictet's fluid, a mixture of SO_2 and CO_2 in the proportion of 64 parts SO_2 to 44 parts CO_2 by weight.

The vapor pressures of these fluids are given in Table 39. The other thermal properties are not at present well known and the values here given must be regarded as rude approximations. Nitrous oxide closely resembles carbon dioxide in its behavior: the vapor pressure and the density of the two fluids are nearly identical.

The latent heat of ethyl chloride is given as 174.5 B.t.u., and that of methyl chloride as 174 B.t.u., temperature not specified, but presumably under atmospheric pressure. Cailletet and Mathias give the following formula for the latent heat of N_2O: $r^2 = 237.15\,(97.5 - t) - 0.928(97.5 - t)^2$.

From this formula the following values are found:

for t =	−20	0	20	40	60	80	deg. fahr.
r =	122.7	120.4	118.2	102.8	87.1	62.2	

The specific heat of N_2O vapor at constant pressure is 0.225. The specific heat of the vapor of ethyl chloride is given by Regnault as 0.274 between 66 deg. and 340 deg.; the specific heat of the liquid is 0.4276 between −18 deg. and 40 deg.

The density of N_2O may be taken the same as that of CO_2. According to Regnault, the density of ethyl chloride gas is 2.2268 times the density of air.

Table 38. Properties of Carbon Dioxide

Temp. deg. fahr. t	Pressure, lb. per sq. in. p	Specific volume of liquid, cu. ft. per lb. v'	Specific volume of vapor, cu. ft. per lb. v''	Heat content of liquid i'	Heat content of vapor i''	Heat of vaporisation r	Internal energy of vaporisation l	Entropy of liquid s'	Entropy of vaporisation r/T
−25	203.4	0.01551	0.4575	−26.91	100.22	127.13	110.8	−0.0561	0.2925
−20	221.0	0.01556	0.4173	−24.75	100.50	125.25	109.0	−0.0513	0.2851
−15	240.5	0.01565	0.3810	−22.72	100.74	123.46	107.3	−0.0467	0.2778
−10	261.8	0.01578	0.3481	−20.56	100.88	121.44	105.4	−0.0419	0.2702
− 5	284.1	0.01594	0.3185	−18.31	100.99	119.30	103.4	−0.0372	0.2626
0	308.0	0.01612	0.2918	−16.00	101.00	117.00	101.3	−0.0325	0.2549
5	334.2	0.01631	0.2672	−13.73	100.97	114.70	99.2	−0.0276	0.2470
10	362.5	0.01652	0.2450	−11.36	100.89	112.35	96.9	−0.0227	0.2391
15	391.0	0.01675	0.2244	− 8.94	100.70	109.64	94.3	−0.0176	0.2309
20	421.6	0.01656	0.2060	− 6.40	100.43	106.83	91.8	−0.0126	0.2228
25	454.7	0.01720	0.1882	− 3.74	100.08	102.82	88.4	−0.0074	0.2143
30	488.8	0 01747	0.1724	− 1.04	99.43	100.47	86.7	−0.0021	0.2014
35	525.5	0.01776	0.1580	+ 1.74	99.00	97.26	83.6	+0.0032	0.1968
40	564.5	0.01805	0.1444	4.36	98.25	93.89	80.4	0.0087	0.1876
45	606.0	0.01835	0.1323	7.54	97.32	89.78	77.05	0.0145	0.1780
50	650.0	0.01870	0.1205	10.76	96.30	85.54	73.31	0.0205	0.1679
55	696.0	0.01918	0.1090	14.18	95.00	80.82	69.16	0.0268	0.1572
60	744.0	0.01986	0.0986	17.85	93.54	75.69	64.90	0.0334	0.1461
65	794.0	0.02052	0.0890	21.50	91.67	70.17	60.08	0.0406	0 1334
70	847.0	0.02136	0.0816	26.02	89.35	63.33	54.03	0.0483	0.1201
75	906.0	0.02230	0.0706	30.96	86.36	55.40	47.20	0.0576	0.1037
80	965.0	0.02365	0.0614	36.80	82.80	46.00	39.16	0.0684	0.0843
85	1026.0	0.02620	0.0500	44.67	76.60	30.23	26.80	0.0828	0.0559
87	1052.0	0.02782	0.0440	48.98	71.80	23.82	18.25	0.0923	0.0393
88.43	1071.0	0.0346	0.0346	61.45	61.45	0.00	0.00	0.1120	0.000

Table 39. Vapor Pressures of Refrigerating Fluids (Lb. per Sq. In.)

Temp., deg. fahr.	Ethyl chloride C_2H_5Cl	Methyl chloride CH_3Cl	Nitrous oxide N_2O	Pictet's fluid 64 SO_2 + 44 CO_2	Temp., deg. fahr.	Ethyl chloride C_2H_5Cl	Methyl chloride CH_3Cl	Nitrous oxide N_2O	Pictet's fluid 64 SO_2 + 44 CO_2
−20	2.26	11.73	.....	11.7	75	22.0	79.8	881	55.8
−15	2.64	13.25	296	12.7	80	24.2	86.7	933	60.1
−10	3.03	14.86	315	13.5	85	26.6	94.2	988	64.5
− 5	3.50	16.69	336	14.4	90	29.1	102.0	1046	69.2
0	4.04	18.73	358	15.6	95	31.8	110.1	1107	74.2
5	4.65	20.87	381	17.0	100	34.7	.....	1171	79.5
10	5.31	23.24	405	18.6	105	37.8	.....	1239	84.7
15	6.02	25.86	431	20.3	110	41.1	.....	.....	89.6
20	6.79	28.68	458	22.2	115	44.6	.....	.....	94.4
25	7.63	31.78	487	24.2	120	48.3	.....	.....	98.9
30	8.58	35.13	518	26.3	125	52.3	.....	.....	.....
35	9.62	38.83	550	28.6	130	56.5	.....	.....	.....
40	10.76	42.7	584	31.6	135	61.0	.....	.....	.....
45	12.04	47.0	620	34.5	140	65.8	.....	.....	.....
50	13.35	51.5	658	37.6	145	70.7	.....	.....	.....
55	14.73	56.4	698	41.0	150	76.1	.....	.....	.....
60	16.33	61.7	740	44.4	155	81.6	.....	.....	.....
65	18.15	67.3	785	47.8	160	87.6	.....	.....	.....
70	20.02	73.3	832	51.6					

Special Changes of State. Mixtures of Vapor and Liquid

1. Isothermal or Constant Pressure: t = const. p = const.
If the initial and final qualities are x_1 and x_2, respectively, then

$$Q = Mr(x_2 - x_1). \qquad U_2 - U_1 = Ml(x_2 - x_1).$$
$$L = p(V_2 - V_1) = Mp\,(v'' - v')(x_2 - x_1).$$

2. Constant Volume:

$$x_2 = x_1\,(v''_1 - v'_1)/(v''_2 - v'_2) = x_1\,v''_1/v''_2, \text{ approx.}$$
$$Q = U_2 - U_1 = M\,[(i'_2 + x_2l_2) - (i'_1 + x_1l_1)]$$

3. Adiabatic: s = const.

$$s'_1 + (x_1r_1/T_1) = s'_2 + (x_2r_2/T_2). \qquad Q = 0.$$
$$L = J(U_1 - U_2) = JM[(i'_1 + x_1l_1) - (i'_2 + x_2l_2)].$$

The relation between p and v during an adiabatic change may be represented approximately by the equation pv^n = const. The exponent n is not constant, however, but varies with the initial quality and initial pressure, as shown by Table 40.

Table 40. Values of n (Water Vapor)

Initial quality	Initial pressure, lb. per sq. in. absolute											
	20	40	60	80	100	120	140	160	180	200	220	240
1.00	1.131	1.132	1.133	1.134	1.136	1.137	1.138	1.139	1.141	1.142	1.143	1.145
0.95	1.127	1.128	1.129	1.130	1.131	1.131	1.132	1.133	1.134	1.135	1.136	1.137
0.90	1.123	1.123	1.124	1.124	1.125	1.125	1.126	1.126	1.127	1.127	1.128	1.129
0.85	1.119	1.119	1.119	1.119	1.120	1.120	1.120	1.120	1.120	1.120	1.120	1.121
0.80	1.115	1.115	1.114	1.114	1.114	1.114	1.113	1.113	1.113	1.113	1.112	1.112
0.75	1.111	1.110	1.110	1.109	1.109	1.108	1.107	1.106	1.106	1.105	1.104	1.104

The volume at the end of expansion (or compression) is $V_2 = V_1\,(p_1/p_2)^{1/n}$, and the external work

$$W = (p_1V_1 - p_2V_2)/(n - 1) = p_1V_1[1 - (p_2/p_1)^{(n-1)/n}]/(n - 1)$$

4. Constant Quality: For a change of state with the quality x constant, an approximate relation between p and v for water vapor, is

$$pv^{1.0631} = 484.2\,x^{1.0631}.$$

For dry steam $x = 1$, and the relation becomes $p^{0.9406}v'' = 327.7$.

This formula may be used for calculating approximately the volume v'' of saturated steam for a given pressure p.

Mixture of Gases and Vapors

Moisture in the Atmosphere. Atmospheric air always contains a certain amount of water vapor mixed with it. The pressure of the vapor cannot exceed the saturation pressure corresponding to the temperature of the atmosphere. Generally, the vapor is in a superheated state, and the pressure is less than the saturation pressure. For a mixture of water vapor and air it may be assumed that Dalton's law holds. Thus, the pressure p_a indicated by the barometer is the sum of p', the pressure of the vapor, and p'', the pressure of the air. On the T-S plane, Fig. 13, let A denote the state of superheated water vapor at the temperature t of the atmosphere, and let m denote the weight of the vapor per

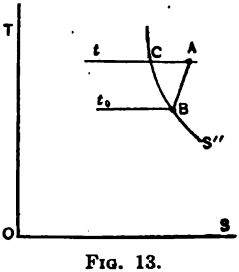


FIG. 13.

cubic foot. **Point** C represents vapor in the saturated condition at the same temperature t. The density or weight per cubic foot is greater in state C than in state A. Denoting by m_1 the weight per cubic foot in state C, the ratio m/m_1 is the **relative humidity** of the air under the given conditions. If the mixture be cooled at constant pressure, the curve AB represents the change of state of the vapor, and at some lower temperature t_0 the vapor becomes saturated and upon further cooling begins to condense. The temperature t_0 is called the **dew point** corresponding to the state A. If p'_B and p'_C denote the vapor pressures corresponding to the temperatures t_0 and t, respectively (at points B and C), then the relative humidity is approximately the ratio p'_B/p'_C.

Properties of Moist Air.

Let t = temperature of air, deg. fahr.
$T = t + 459.6$, the corresponding absolute temperature.
h = relative humidity.
B = barometric pressure, in inches of mercury.
e = pressure of *saturated* vapor corresponding to temperature t, in inches of mercury.
D' = weight per cu. ft. of saturated vapor at temperature t, lb.
D'' = weight per cu. ft. of dry air at temperature t and pressure B, lb.
D = weight per cu. ft. of mixture at temperature t, and with relative humidity h, lb.
R = 53.34, the characteristic constant of dry air.
R_m = characteristic constant of mixture.

The **partial pressure of the vapor** is he, that of the air $B - he$. Hence, by **volume**, the mixture has he/B parts of vapor, and $(B - he)/B$ parts of air; that is, the addition of moisture reduces the effective volume of air 100 he/B per cent.

The constant R_m of the mixture is given by $R_m = 53.34/[1 - 0.378(he/B)]$.

The **weight per cu. ft.** of the mixture is $D = hD' + [D''(B - he)/B]$, and this reduces to $D = 1.326(B/T) - 0.501(he/T)$.

These formulæ apply only when the vapor pressure is small relative to the total pressure. Table 41 facilitates the use of the preceding equations.

Example. The temperature of the air is 84 deg. fahr., the relative humidity 0.70, and the barometric pressure 29.94 in.

From Table 41 the vapor pressure e for 84 deg. is 1.171 in. of mercury and weight of 1 cu. ft. of saturated vapor is 12.37 gr. Hence the air contains $12.37 \times 0.70 = 8.69$ gr. of moisture per cu. ft. and can absorb $12.37 - 8.69 = 3.68$ gr. more before condensation ensues. If the air is cooled the air will become saturated at the pressure $he = 1.171 \times 0.70 = 0.82$ in. of mercury, which corresponds to the temperature 73.5 deg. fahr. This is the dew point. The weight of the mixture per cu. ft. is obtained from the factors in the last two columns of the table: thus, $D = (29.94 \times 0.002439) - (0.70 \times 0.00108) = 0.07226$ lb. The volume of 1 lb. is therefore $1/0.07226 = 13.84$ cu. ft.

Table 41. Properties of Air Mixed with Water Vapor

Temp. deg. fahr. t	Vapor pressure: Inches of mercury e	Vapor pressure: Pounds per sq. in.	Weight of water vapor per cu. ft.: Grains	Weight of water vapor per cu. ft.: Pounds D'	$\frac{1.326}{T} \times 1000$	$0.501\frac{e}{T} \times 1000$
32	0.1804	0.0886	2.129	0.000304	2.697	0.184
34	0.1955	0.0960	2.297	0.000328	2.686	0.198
36	0.2117	0.1040	2.477	0.000354	2.676	0.214
38	0.2290	0.1125	2.669	0.000381	2.665	0.231
40	0.2477	0.1217	2.872	0.000410	2.654	0.248
42	0.2677	0.1315	3.091	0.000442	2.644	0.267
44	0.2891	0.1420	3.322	0.000475	2.633	0.287
46	0.3119	0.1532	3.570	0.000510	2.623	0.309
48	0.3366	0.1653	3.831	0.000547	2.612	0.332
50	0.3627	0.1781	4.110	0.000587	2.602	0.357
52	0.3905	0.1918	4.411	0.000630	2.592	0.383
54	0.4202	0.2064	4.720	0.000674	2.582	0.410
56	0.4518	0.2219	5.054	0.000722	2.572	0.439
58	0.4856	0.2385	5.410	0.000773	2.562	0.470
60	0.5217	0.2562	5.785	0.000827	2.552	0.503
62	0.5598	0.2749	6.043	0.000883	2.542	0.538
64	0.6005	0.2949	6.604	0.000943	2.532	0.575
66	0.644	0.3161	7.048	0.001007	2.523	0.614
68	0.689	0.3386	7.52	0.001074	2.513	0.655
70	0.738	0.3625	8.02	0.001145	2.504	0.698
72	0.789	0.3879	8.54	0.001220	2.494	0.744
74	0.844	0.4148	9.10	0.001300	2.485	0.793
76	0.905	0.4433	9.68	0.001383	2.476	0.845
78	0.964	0.4735	10.30	0.001471	2.466	0.899
80	1.029	0.5054	10.95	0.001564	2.457	0.956
82	1.098	0.539	11.64	0.001663	2.448	1.016
84	1.171	0.575	12.37	0.001768	2.439	1.080
86	1.248	0.613	13.15	0.001879	2.430	1.146
88	1.329	0.653	13.98	0.001997	2.422	1.216
90	1.415	0.695	14.85	0.002121	2.413	1.290
92	1.506	0.739	15.77	0.002253	2.404	1.368
94	1.602	0.787	16.74	0.002391	2.395	1.450
96	1.704	0.837	17.76	0.002537	2.387	1.537
98	1.812	0.890	18.79	0.002690	2.378	1.629
100	1.925	0.946	19.95	0.002850	2.370	1.724
102	2.044	1.004	21.12	0.00 017	2.361	1.825
104	2.171	1.066	22.35	0.003193	2.353	1.931
106	2.303	1.131	23.63	0.003376	2.344	2.041
108	2.441	1.199	24.96	0.003566	2.336	2.155
110	2.588	1.271	26.36	0.003765	2.328	2.274
120	3.439	1.689	34.42	0.004916	.....	.....
130	4.518	2.219	44.45	0.00635	.....	.....
140	5.874	2.885	56.86	0.00812	.....	.....
150	7.56	5.714	72.0	0.01029	.....	.....
160	9.64	4.737	90.5	0.01293	.....	.....
170	12.19	5.988	112.7	0.01611	.....	.....
180	15.28	7.506	139.4	0.01991	.....	.....
190	20.01	9.335	171.0	0.02443	.....	.....
200	23.46	11.523	208.3	0.02976	.....	.....
210	28.75	14.122	252.1	0.03601	.....	.....
212	29.92	14.697	261.7	0.03738	.....	.....

The Steam Engine

The Rankine Cycle. The ideal Rankine cycle is generally employed by engineers as a standard of reference by which the performance of steam engines and steam turbines is measured. Fig. 14 shows this cycle on the T-S and p-V planes. AB represents the heating of the water in the boiler, BC represents evaporation (and superheating if there is any), CD the assumed adiabatic expansion in the engine cylinder, and DA condensation in the condenser.

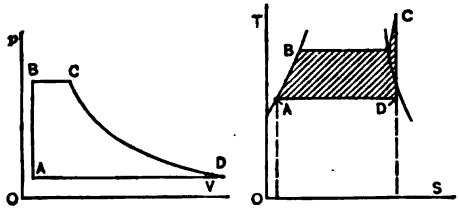


FIG. 14.—Rankine Cycle.

Let i_a, i_b, i_c, i_d represent the heat content per lb. of steam in the four states A, B, C, and D, respectively. Then the heat transformed into work, represented by the area $ABCD$, is $Q_a = i_c - i_d$.

The heat expended on the fluid is $i_c - i_a$; hence the thermodynamic **efficiency of the cycle** is $E_t = (i_c - i_d)/(i_c - i_a)$.

The **steam consumption** of the ideal Rankine engine in lb. per h.p.-hr. is $N_r = 2546/Q_a = 2546/(i_c - i_d)$. For steam consumptions per kw.-hour (for use in steam turbine calculations) see p. 355.

The performance of an engine is frequently stated in terms of the heat used per h.p.-hr. For the ideal Rankine engine this is

$$Q_r = 2546/E_t = 2546\ (i_c - i_a)/(i_c - i_d).$$

Efficiency of the Actual Engine. Let Q denote the heat transformed into work per lb. of steam by the actual engine; then if Q_1 is the heat furnished by the boiler per lb. of steam, the thermodynamic **efficiency of the engine** is $E_t = Q/Q_1$.

The efficiency thus defined is however misleading, as it takes no account of the conditions of operation, as regards boiler and condenser pressures, superheat, or quality of steam. It is customary therefore to define the efficiency as the ratio Q/Q_a, where Q_a is the **available heat**, or the heat that might possibly be transformed under ideal conditions. This efficiency for engines and turbines ranges from 0.50 to 0.80. The efficiency may also be expressed in terms of steam consumed; thus, if N_a is the steam consumption of the actual engine and N_r is the steam consumption of the ideal Rankine engine under similar conditions, then $E = N_r/N_a$.

Example. Suppose the boiler pressure to be 180 lb. per sq. in. absolute, superheat 150 deg., and the condenser pressure 3 in. of mercury. From the steam tables or diagram the following values are found: $i_c = 1279.9$, $i_d = 940$, $i_a = 82.9$. The available heat is $Q_a = 1279.9 - 940 = 339.9$ B.t.u., and the thermodynamic efficiency of the cycle is $339.9/(1279.9 - 82.9) = 0.284$. The steam consumption per h.p.-hr. is $2546/339.9 = 7.5$ lb., and the heat used per h.p.-hr. is $2546/0.284 = 8960$ B.t.u. If an actual engine working under the same conditions has a steam consumption of 11.4 lb. per h.p.-hr., its efficiency is $7.5/11.4 = 0.658$, and its heat consumption per h.p.-hr. is $8960/0.658 = 13{,}600$ B.t.u.

Calorimetric Analysis of the Steam Engine. The first attempt to determine quantitatively the thermal actions in the steam-engine cylinder is due to Hirn. By **Hirn's method**, equations are derived for the heat interchanges between the steam and the cylinder walls, and the quantities involved are such as may be determined from an accurate test. In Fig. 15, points 1, 2, 3 and 0 represent cut-off, release, compression, and admission, respectively. Let U_0, U_1, U_2, U_3 represent the energy per lb. of fluid at the corresponding points. Let M denote the weight of fluid supplied per stroke, and M_0 the weight of fluid caught in the clearance space. Let Q_a, Q_b, Q_c, Q_d denote respectively the quantities of heat absorbed by the cylinder walls from the fluid during the processes represented by a, b, c, and d on the diagram; and let W_a, W_b, W_c, and W_d denote the external work performed during the corresponding parts of the cycle. Let Q denote the heat brought into the cylinder per stroke by the working fluid; thus, $Q = Mi$, where i is the heat content of the entering steam. Finally, let i' denote the heat content of the liquid at the pressure of the exhaust steam, G the weight of condensing water used per stroke, and t', t'' the initial and final temperatures of the cooling water. Then the heat interchanges are expressed by the following equations:

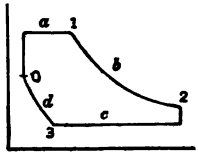


FIG. 15.

$$Q_a = Q + U_0 - U_1 - AW_a. \qquad Q_b = U_1 - U_2 - AW_b.$$
$$Q_c = U_2 - U_3 - Mi - G(t'' - t') + AW_c. \qquad Q_d = U_3 - U_0 + AW_d.$$

In getting the expressions for U_0, U_1, etc., there are five unknown quantities, namely M_0, and the four qualities x_0, x_1, x_2, x_3. Between these there are four relations. Let V_0 denote the volume of the clearance space, and V_1, V_2, V_3 the spaces swept through by the piston up to the points 1, 2, 3. Also let $s = v'' - v'$, the difference between the volume of vapor and liquid. Then

$$V_0 = M_0(x_0 s_0 + v'). \qquad V_0 + V_1 = (M + M_0)(x_1 s_1 + v').$$
$$V_0 + V_2 = (M + M_0)(x_2 s_2 + v'). \qquad V_0 + V_3 = M_0(x_3 s_3 + v').$$

If one quality be known or assumed, these four equations give M_0 and the remaining qualities. It is usually assumed that the steam in the cylinder is nearly or quite dry at the beginning of compression, hence $x_3 = 1$.

For an exhaustive treatment of the calorimetric analysis of steam engines, see Peabody's "Thermodynamics," 5th ed., chap. xi.

REFRIGERATION

Air Refrigeration. When air is used as a medium for refrigeration the reversed Joule cycle is employed (see p. 206). The refrigerating machine has four essential organs: 1. A compressor in which the air is compressed. 2. A cooling coil surrounded by water. 3. An expansion cylinder. 4. A brine coil. Fig. 16 represents ideal p-V and T-S diagrams. Point A represents the state of the air entering the compressor from the brine coil or cold room, AB represents the compression, BC the cooling of the air at constant pressure, CD the expansion of air in the expansion cylinder, and DA the absorption of heat by the air during the passage through the brine coils. In the p-V diagram $ABEF$ represents the indicator diagram of the compressor, $ECDF$ that of the expansion cylinder. The difference, area $ABCD$, represents the work that must be furnished from external sources. In the T-S diagram area B_1BCC_1 represents the heat absorbed from the air by the cooling water, area C_1DAB_1, the heat absorbed by the air from the brine, and area $ABCD$ the heat

equivalent of the work required to drive the machine. The temperature at point A must be somewhat lower than the brine temperature, the temperature at C somewhat above the temperature of the cooling water. With the open-cycle machine the lower pressure p_1 is atmospheric pressure; with a closed-cycle machine, as the Allen dense-air machine, p_1 may be 40 to 60 lb. per sq. in. and p_2 as high as 200 lb. per sq. in.

Let Q = heat in B.t.u. absorbed from brine (or cold room) per min.
M = weight of air circulated per min., lb.
c_p = 0.24, specific heat of air at constant pressure
W = work required per min., ft.-lb.
V_c = displacement volume of compressor cylinder, cu. ft.
V_e = displacement volume of expansion cylinder, cu. ft.
n = number of working strokes per min.

With assumed adiabatic compression and expansion the following relations hold:

$$T_b/T_a = T_c/T_d = (p_2/p_1)(k-1)/k \qquad M = Q/c_p(T_a - T_d).$$
$$W = 778Q(T_b - T_a)/T_a. \qquad H.P. = W/33{,}000.$$
$$V_c = MBT_a/144p_1n. \qquad V_e = MBT_d/144p_1n.$$

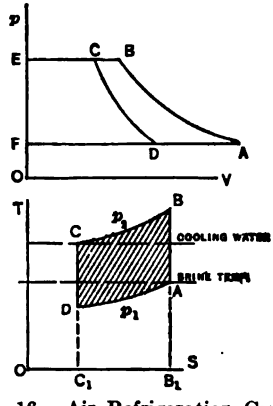


FIG. 16.—Air Refrigeration Cycle.

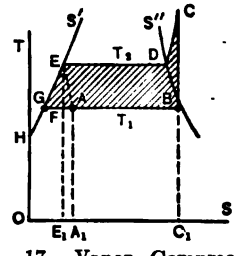


FIG. 17.—Vapor Compression Refrigeration Cycle.

Vapor Compression Machines. The essential organs of a vapor compression system are the same as in the system using air, except that the expansion cylinder is replaced by an expansion valve through which the liquefied medium flows from the high-pressure condensing coils to the low-pressure brine coils. The cycle of operation is best shown on the T-S plane (Fig. 17). The point B represents the state of the refrigerating medium leaving the brine coils and entering the compressor. Usually in this state the fluid is nearly dry saturated vapor, that is, point B is near the saturation curve S''. BC represents the adiabatic compression, during which the fluid is usually superheated. In the state C the superheated vapor passes into the cooling coils and is cooled at constant pressure, as indicated by CD, and then condensed at temperature T_2 as shown by DE. The liquid now flows through the expan-

sion valve into the brine coils. This is a throttling process and the final-state point A is located on the T_1-line in such a position as to make the heat content for state A (= area $OHGAA_1$) equal to the initial heat content at E (= area $0HEE_1$). The mixture of liquid and vapor now absorbs heat from the brine and vaporises, as indicated by AB.

The **heat absorbed from the brine**, represented by area A_1ABC_1, is

$$Q_1 = i_b - i_a = i_b - i_e$$

The **heat rejected to the cooling water**, represented by area C_1CDEE_1, is

$$Q_2 = i_c - i_e = c_p(T_c - T_d) + r_2$$

where r_2 denotes the latent heat at the upper temperature T_2, and c_p the specific heat of the superheated vapor. The **work** that must be **supplied per pound of fluid circulated** is $W = J(Q_2 - Q_1) = J(i_c - i_b)$.

The ratio $JQ_1/W = (i_b - i_e)(i_c - i_b)$ is sometimes called the **coefficient of performance.**

If Q denotes the heat that is required to be absorbed from the brine per hr., then the **weight of fluid circulated per hour** is $M = Q/(i_b - i_a)$; or, if B is taken on the saturation curve, $M = Q/(i''_1 - i'_2)$.

The work per hr. is $W = JM(i_c - i''_1) = JQ(i_c - i''_1)/(i''_1 - i'_2)$ ft.-lb., and the **h.p. required** is $H = Q(i'_c - i''_1)/2546(i''_1 - i'_2)$.

If v''_1 is the volume of the saturated vapor at the temperature T_1 in the brine coils, and n the number of working strokes per min., the **displacement volume of the compressor cylinder** is $V = Mv''_1/60n$.

The values of i''_1, i_2 in the preceding formulæ are found in the tables of saturated vapors. The heat content i_c of the superheated vapor may be determined in the case of ammonia from Table 36 for superheated ammonia.

Vapors Used in Refrigeration. The three vapors that are principally used for refrigeration are **ammonia** (NH_3), **carbon dioxide** (CO_2) and **sulphur dioxide** (SO_2). The temperatures of the working fluid in the condensing and brine coils are fixed by the temperature of cooling water and the temperature at which the brine is to be kept, respectively. Since for a vapor the pressure depends on the temperature only, the discharge and suction pressures that must be used are dependent on the properties of the fluid employed. Another consideration in the choice of a fluid is the volume of fluid required for a given amount of refrigeration. In general, the fluid for which the ratio (latent heat : specific volume) is greatest requires the smallest cylinder displacement per unit of refrigeration.

Taking 70 deg. fahr. as the temperature of the fluid in the condensing coils and 15 deg. as that in the cooling coils, the following are the conditions for the available vapors:

Pressures, lb. per sq.in.:	NH_3	CO_2	SO_2	Ethyl chloride	Methyl chloride	Nitrous oxide	Pictet's fluid
Suction.....	42.7	391	15.2	6.0	25.9	431	20.3
Discharge.....	129.2	847	49.6	20.0	73.3	832	51.6
Relative volume per unit weight of refrigeration.....	4.4	1	12.0	....	....	....	....

With SO_2 the pressures are low but the volume of medium is comparatively large, while with CO_2 small volume is accompanied by extremely high pressures.

Refrigeration Produced by Ammonia and Carbon Dioxide. Tables 42 and 43 give the refrigeration produced in B.t.u. per h.p.-hr. under different conditions. It is assumed that there are no losses in the cycle, except

that inherent in the use of the expansion valve. The temperature at the top of the column is that of the vapor in the brine coil, while at the side is given the temperature of the condensed fluid as it enters the expansion valve. For example, under a pressure p of 150 lb. with a temperature t_0 of 20 deg. fahr. in the brine coils and a temperature of 70 deg. fahr. at the expansion valve, the refrigeration produced in an ammonia machine without losses is 17,800 B.t.u. per h.p.-hr.; this is 90 per cent of the refrigeration that would be produced if the liquid temperature were reduced to 20 deg., the temperature in the brine coils.

Table 42. Refrigeration Produced by Ammonia in B.t.u. per H.p.-Hour

Temp. at expansion cock, degrees fahr. t'	Vapor temperature t_0 in deg. fahr.					
	30	20	10	0	−10	−20
	Condenser pressure p = 100 lb. per sq. in.					
t_0	45,400	30,900	23,350	21,000	17,150	12,900
50	43,600	29,300	21,950	19,100	15,500	11,350
	p = 150 lb.					
t_0	25,000	19,900	15,900	14,400	12,400	10,100
50	24,100	18,700	14,900	13,050	11,000	8,850
70	23,300	17,800	14,250	12,450	10,650	8,500
	p = 200 lb.					
t_0	18,300	15,200	13,050	11,700	10,350	8,600
50	17,800	14,400	12,200	10,650	9,150	7,550
70	17,000	13,800	11,600	10,100	8,750	7,250
90	16,100	13,000	11,150	9,650	8,250	6,850
	p = 250 lb.					
t_0	15,100	12,900	11,400	10,250	9,150	7,800
50	14,600	12,100	10,600	9,300	8,050	6,900
70	14,000	11,500	10,050	8,800	7,700	6,500
90	13,300	11,000	9,650	8,450	7,350	6,200
	p = 300 lb.					
t_0	13,400	11,200	10,200	9,250	8,300	7,150
50	13,000	10,950	9,450	8,450	7,400	6,300
70	12,500	10,400	8,995	8,050	7,100	6,000
90	11,900	10,000	8,650	7,650	6,750	5,700

Table 43. Refrigeration Produced by Carbon Dioxide in B. t. u. per H.p.-Hour

Temperature at expansion cock, deg. fahr. t'	Vapor temperature t_0 in deg. fahr.					
	30	20	10	0	−10	−20
	Condenser pressure p = 800 lb. per sq. in.					
t_0	32,600	26,200	21,750	18,650	15,600	13,300
50	29,650	23,650	19,000	16,150	13,450	11,050
60	27,600	21,600	17,150	14,350	11,650	9,550
	p = 1000 lb.					
t_0	22,600	18,800	16,050	13,750	12,200	10,700
50	20,800	17,300	14,250	12,000	10,250	8,650
60	19,350	15,850	12,900	10,750	9,150	7,500
70	17,800	14,750	12,050	10,050	8,400	6,950
	p = 1200 lb.					
t_0	16,600	14,700	13,050	11,550	10,350	9,250
50	15,350	13,550	11,500	9,950	8,750	7,500
60	14,450	12,500	10,600	9,050	7,850	6,500
70	13,600	11,700	9,850	8,550	7,250	6,150
80	12,650	10,800	9,150	8,000	6,750	5,600
90	11,450	9,750	8,300	7,100	6,050	5,150
	p = 1400 lb.					
t_0	14,250	12,700	11,400	10,350	9,250	8,250
50	13,200	11,650	10,200	8,950	7,900	6,850
60	12,400	10,900	9,350	8,100	7,050	6,150
70	11,750	10,150	8,750	7,600	6,600	5,650
80	10,900	9,600	8,200	7,200	6,200	5,400
90	10,000	8,750	7,550	6,600	5,650	5,000
	p = 1600 lb.					
t_0	13,150	11,600	10,250	9,950	8,500	7,550
50	12,200	10,600	9,300	8,300	7,400	6,500
60	11,450	10,000	8,600	7,500	6,600	5,850
70	10,700	9,250	8,150	7,000	6,250	5,500
80	9,800	8,750	7,650	6,700	5,850	5,200
90	9,150	8,200	7,150	6,250	5,850	4,950

FLOW OF COMPRESSIBLE FLUIDS

Important examples of the flow of compressible fluids are the following: 1. The flow of air and steam through orifices and short tubes or nozzles, as in the steam turbine. 2. The flow of compressed air, steam, and illuminating gas in long mains. 3. The flow of low-pressure gases, as furnace gases in flues and chimneys or air in ventilating ducts. 4. The flow of gases in moving channels, as in the centrifugal fan.

Notation.

Let M = weight in lb. of fluid flowing past a given section per sec.
F = area of section, sq. ft.
w = mean velocity in ft. per sec. at the given section.
h = height of cross-section above an assumed datum.
v = volume per unit weight (cu. ft. per lb.).
p = pressure of fluid at given section, lb. per sq. ft.
u = internal energy of fluid, B.t.u. per lb.
i = heat content, B.t.u. per lb.
Q_{12} = heat entering the flowing fluid between sections F_1 and F_2.
R_{12} = energy expended in overcoming internal and external friction between sections F_1 and F_2.

The cross-sections of the tube or channel are denoted by F_1, F_2, etc., Fig. 18, and the various magnitudes pertaining to these sections are denoted by corresponding subscripts. Thus, at section F_1, the velocity, volume, pressure, energy, are respectively w_1, v_1, p_1, u_1; at section F_2 they are w_2, v_2, p_2, u_2.

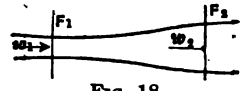


FIG. 18.

Fundamental Equations. The condition of continuity is expressed by the equation

$$M = F_1w_1/v_1 = F_2w_2/v_2, \quad \text{or} \quad dv/v = (dF/F) + (dw/w).$$

The principle of conservation of energy leads to the equation

$$[(w_2^2 - w_1^2)/2g] + J(u_2 - u_1) + p_2v_2 - p_1v_1 + h_2 - h_1 = JQ_{12};$$

or, introducing the expression for heat content $i = u + Apv$,

$$[(w_2^2 - w_1^2)/2g] + J(i_2 - i_1) + h_2 - h_1 = JQ_{12}.$$

If the sections are taken an infinitesimal distance apart, this energy equation takes the form

$$(wdw/g) + Jdi + dh = JdQ.$$

A second fundamental equation is obtained by the further application of the energy equation, taking the friction work R into account. It is

$$dQ + AdR = du + Apdv = di - Avdp.$$

Combining these last equations,

$$(wdw/g) + vdp + dR + dh = 0.$$

Usually $h_1 - h_2$ and Q_{12} are so small as to be negligible. In this case the two fundamental equations become

$$(w_2^2 - w_1^2)/2g = J(i_1 - i_2); \qquad (w_2^2 - w_1^2)/2g = -\int_1^2 vdp - R_{12}$$

For gases, $i_1 - i_2 = c_p(T_1 - T_2) = Ak(p_1v_1 - p_2v_2)/(k-1)$; hence

$$[(w_2^2 - w_1^2)/2g] = Jc_p(T_1 - T_2) = k(p_1v_1 - p_2v_2)/(k-1).$$

For a mixture of vapor and liquid, $i = i' + xr$; therefore

$$[(w_2^2 - w_1^2)/2g] = J[i'_1 + x_1r_1 - (i'_2 + x_2r_2)].$$

Flow Through Orifices

A case of special importance is the flow of a gas or vapor from a reservoir into a region of lower pressure through an orifice or short tube. In Fig. 19 the section F_1 is taken within the reservoir where the pressure is p_1, and the section F_2 is taken at the plane of the orifice. The pressure

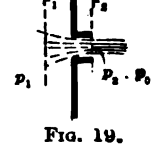


FIG. 19.

at this section is taken as p_2, and the pressure in the region into which the jet is flowing is taken as p_0. The initial velocity w_1 is so small that w_1^2 may be neglected in comparison with w_2^2. Hence

$$w_2^2/2g = J(i_1 - i_2), \text{ and } w_2 = 223.7\sqrt{i_1 - i_2}$$

The character of the flow depends on the ratio p_0/p_1, and two distinct cases may be noted. Let the law of frictionless adiabatic expansion of the fluid in question be $p_1v_1^n = p_0v_0^n = \text{const.}$ For gases, $n = k$, for steam n has the value given in Table 40, p. 223, and for superheated steam or ammonia n may be taken as 1.3. Take $p_m = p_1[2/(n+1)]^{n/(n-1)}$; then p_m is the **critical pressure.** For gases $p_m/p_1 = 0.53$, approx.; for saturated steam the ratio is about 0.575; and for moderately superheated steam, about 0.55.

Case 1. $p_0 > p_m$. Then $p_2 = p_0$, and the discharge M is given by the equation

$$M = \frac{F_2}{v_2}\sqrt{2gp_1v_1\frac{n}{n-1}\left[1 - \left(\frac{p_0}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

Thus the weight flowing per second depends upon the two pressures p_1 and p_0.

Case 2. $p_0 \leqq p_m$. Then $p_2 = p_m$, that is, p_2 remains constant whatever lower value the pressure p_0 may take. In this case the discharge is

$$M = F_2\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}}\sqrt{2g\frac{n}{n+1}\frac{p_1}{v_1}}.$$

Hence the discharge depends upon p_1 only and is not influenced by the pressure p_0.

Formulæ for Discharge of Air. When the back pressure p_0 exceeds the critical pressure $p_m = 0.53\ p_1$,

$$M = 2.05\,Fp_0\sqrt{\frac{1}{T}\left(\frac{p}{p_0}\right)^{0.286}\left[\left(\frac{p}{p_0}\right)^{0.286} - 1\right]}$$

in which p and T refer to the air in the reservoir. Values $(p/p_0)^{0.286}$ are given on p. 203. $[(n-1)/n = 0.286 \text{ when } n = 1.4]$

When p_0 is equal to or less than p_m, $M = 0.53Fp/\sqrt{T}$.

For a small difference of pressure, $M = 1.1F\sqrt{(p/T)(p - p_0)}$

In these three formulæ F is to be taken in sq. in. when p is taken in lb. per sq. in. The discharge as calculated by the formula should be multiplied by a **coefficient of discharge** to take account of friction, contraction, etc. From the experiments of Zeuner and others, the following mean values of this coefficient are obtained:

Orifices in a thin plate	0.64
Short cylindrical pipe, inner corners not rounded	0.75 to 0.84
Well-rounded mouthpiece	0.98

Formulæ for Discharge of Saturated Steam. When the back pressure p_0 is less than the critical pressure p_m, the discharge depends upon the area of orifice F and reservoir pressure p. There are three formulæ widely used to express the discharge G in terms of F and p, as follows:

1. Napier's equation, $M = Fp/70$.
2. Grashof's formula, $M = 0.0165F\ p^{0.97}$.
3. Rateau's formula, $M = Fp(16.367 - 0.96 \log p)/1000$.

In these formulæ F is to be taken in square inches, p in lb. per sq. in. Napier's formula is merely convenient as a rough check. The coefficient

of discharge may be taken as 1; that is, no correction is required.

Table 44. Values of $p^{0.97}$ for Use in Grashof's Formula

p	$p^{0.97}$	p	$p^{0.97}$	p	$p^{0.97}$	p	$p^{0.97}$
15	13.8	50	44.5	110	95.5	225	191.2
20	18.3	55	48.8	120	104.0	250	212.0
25	22.7	60	53.1	130	112.4	275	232.0
30	27.1	70	61.6	140	120.7	300	253.0
35	31.5	80	70.1	150	129.1		
40	35.8	90	78.6	175	150.0		
45	40.1	100	87.1	200	170.6		

When the back pressure p_0 is greater than the critical pressure p_m the velocity and discharge are found most conveniently from the general formulæ of flow. From the steam tables (p. 209) or from the **Mollier chart** (p. 214) find the initial heat content i_1 and the final heat content i_0 after adiabatic expansion; also the specific volume v_0 in the final state. See Fig. 20. Then

$$w = 223.7\sqrt{i_1 - i_0} \qquad \text{and} \qquad M = Fw/v_0.$$

The same method is used in the case of steam initially superheated.

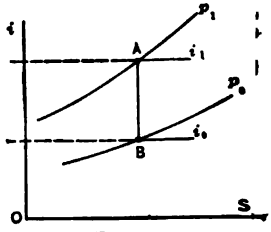


FIG. 20.

Example. Required the discharge through an orifice ½ in. in diameter of steam at 140 lb. per sq. in. superheated 110 deg.; back pressure, 90 lb. per sq. in.

From the Mollier chart, $i_1 = 1253$, and $i_0 = 1213$. Also $v_0 = 5.26$ cu. ft.

$$w = 223.7\sqrt{1253 - 1213} = 1413 \text{ ft. per sec.}$$
$$F = 0.1964 \text{ sq. in.} = (0.1964/144) \text{ sq. ft.,}$$
$$M = Fw/v_0 = (0.1964/144) \times (1413/5.26) = 0.366 \text{ lb. per sec.}$$

Flow Through Diverging Nozzles. At the throat, or smallest cross-section of the nozzle (Fig. 21) the pressure takes the value $p_m = 0.57\ p_1$. The weight discharged is fixed by the area F_m of the throat and the reservoir pressure p_1, and may be found from Grashof's or Rateau's formula. The diverging part of the nozzle permits further expansion to the back pressure p_0, the velocity of the jet meanwhile increasing from w_m, the critical velocity at the throat, to w_0 given by the fundamental equation $w_0 = 223.7\sqrt{i_1 - i_0}$.

The frictional resistances in the nozzle have the effect of decreasing the jet energy $w_0^2/2g$ and correspondingly increasing the energy and heat content of the flowing fluid. Thus, if i_0 is the heat content in the final state with frictionless expansion, $i'_0 (> i_0)$ is the heat content when friction is taken into account; hence $w'^2_0/2g = J(i_1 - i'_0)$ is less than $w_0^2/2g = J(i_1 - i_0)$. The loss of kinetic energy in B.t.u. is $i'_0 - i_0$, and the ratio of this loss to the available kinetic energy, that is $(i'_0 - i_0)/(i_1 - i_0)$, is denoted by y. For values of y, see p. 235.

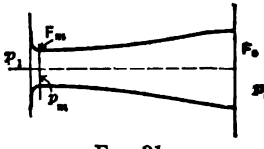


FIG. 21.

The **design of a nozzle for a given discharge** M with pressure p_1 and p_0 is most conveniently effected with the aid of the Mollier chart. Determine p_m, the critical pressure, and i_1, i_m, i_0, assuming frictionless flow. Then

$$w_m = 223.7\sqrt{i_1 - i_m}, \quad \text{and} \quad w'_0 = 223.7\sqrt{(1 - y)(i_1 - i_0)}.$$

Next find v_m and v'_0. Then, from the equation of continuity,

$$F_m = Gv_m/w_m, \text{ and } F'_0 = Gv'_0/w'_0.$$

The following example illustrates the method.

Example. Required the throat and end sections of a nozzle to deliver 0.7 lb. of steam per sec. The initial pressure is 160 lb., the back pressure 15 lb., and the steam is initially superheated 100 deg.; $y = 0.15$.

The critical pressure is $160 \times 0.55 = 88$ lb. On the Mollier chart (Fig. 22) the point A representing the initial state is located, and line of constant entropy (a frictionless adiabatic) is drawn from A. This cuts the curves $p = 88$ and $p = 15$ in the points B and C, respectively. The three values of i are found to be $i_1 = 1251$, $i_m = 1198$, $i_0 = 1065$. Of the available drop in heat content, $i_1 - i_0 = 186$ B.t.u., 15 per cent, or 27.9 B.t.u. is lost through friction. Hence, $CD = 27.9$ is laid off and D is projected horizontally to point C' on the curve $p = 15$. Then C' represents the final state of the steam, and the quality is found to be $x = 0.94$. The specific volume in the state C' is $26.27 \times 0.94 = 24.7$ cu. ft. Likewise the specific volume for the state B is found to be 5.2 cu. ft.

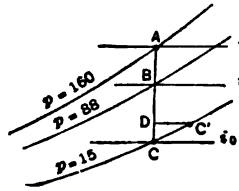


FIG. 22.

For the velocities at throat and end sections,

$$w_m = 223.7\sqrt{1251 - 1198} = 1629 \text{ ft. per sec.}$$
$$w_0 = 223.7\sqrt{186 - 27.9} = 2813 \text{ ft. per sec.}$$
$$F_m = (0.7 \times 5.2)/1629 = 0.00223 \text{ sq. ft.} = 0.322 \text{ sq. in.}$$
$$F_0 = (0.7 \times 24.7)/2813 = 0.0061 \text{ sq. ft.} = 0.88 \text{ sq. in.}$$

The diameters are $d_m = 0.64$ in. and $d_0 = 1.06$ in.

Velocity Coefficients. Loss of Energy y. On account of friction losses the actual velocity w attained by the jet is less than the velocity w_0 calculated under ideal conditions. That is, $w = xw_0$, where x (< 1) is a velocity coefficient. Evidently the coefficient x is connected with the coefficient y (see p. 234), giving the loss of energy by the relation $y = 1 - x^2$.

For **orifices in thin plates**, the experiments of Zeuner on air and of Gutermuth and Bendemann on saturated and superheated steam indicate that x has the value of 0.97 to 0.975.

For **flow through nozzles**, Stodola states that the energy loss varies from $y = 0.05$ for short tubes to $y = 0.15$ for long tubes. The most comprehensive experiments in this field are those of Briling (*Mitteil. über Forsch.-arb.*, 68) and of Josse and Christlein (*Zeit. Ver. Deutsch. Ing.*, 1911, p. 2081). Briling used slightly superheated steam and two nozzles, one cylindrical with a diam. of 0.254 in. and length of 1.18 in., the other a slightly converging nozzle with a minimum diam. of 0.34 in. For the cylindrical tube he found $x = 0.95$; for the convergent tube x increased from 0.92 to 0.965 as the velocity increased. These results indicate an energy loss of 10 per cent in the cylindrical tube and of 7 to 16 per cent in the convergent tube. The experiments of Josse and Christlein were made on 6 De Laval nozzles having ratios of maximum to minimum sectional areas ranging from 1.185 to 12.94. Under the most favorable conditions, with a tube properly proportioned for the pressure drop, the experiments showed a value of $x = 0.94$ for a velocity of 2000 ft. per sec. and a value around 0.95 for velocities of 2500 to 3500 ft. per sec. Taking $x = 0.945$ as a mean value, the energy loss is,

$$y = 1 - x^2 = 1 - 0.945^2 = 0.107, \text{ or } 10.7 \text{ per cent.}$$

Josse and Christlein also made experiments on the friction loss in a set of **three guide vanes**, which take the place of nozzles in turbines of the Rateau

type. Under favorable conditions with slightly superheated steam the mean value $x = 0.923$ was found. The corresponding loss of energy is $y = 0.148$ or 14.8 per cent. With high superheat and a velocity of 2200 ft. per sec., the value $x = 0.95$ was obtained, but at about 800 ft. per sec. x dropped to about 0.85. Rosenhain's experiments give values of x for several types of nozzles. Table 45 gives data and results (see Peabody's "Thermodynamics," p. 442).

Table 45. Values of the Velocity Coefficient x for Various Types of Nozzles

(Rosenhain, *Proc. Inst. C. E.*, vol. 140, p. 199)

Ratio of greatest to least diameter	Taper	Initial pressure	Velocity coefficient, x	Loss of energy, y
1.56	1:20	150.0	0.946	0.105
1.96	1:12	275.0		
1.36	1:12	97.5	0.972	0.045
1.28	1:12	80.0	0.903	0.185
1.39	1:30	105.0	0.913	0.166
1.32	1:30	90.0	0.929	0.137
1.26	1:30	77.5	0.901	0.188
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$$(w_2^2 - w_1^2)/2g = (i_1 - i_2)J.$$

The velocities w_2 and w_1 are practically equal, and it follows that $i_1 = i_2$; that is, in a throttling process the initial and final values of the heat content are equal.

For a mixture of liquid and vapor, $i = i' + xr$, hence the equation of throttling is $i'_1 + x_1r_1 = i'_2 + x_2r_2$. In the case of a perfect gas, $i = c_pT + i_o$, hence the equation of throttling is $c_pT_1 + i_o = c_pT_2 + i_o$, or $T_1 = T_2$.

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Example. The liquid ammonia at a temperature of 70 deg. fahr. passes through the valve into the brine coil in which the temperature is 20 deg. and the pressure is 47.75 lb. per sq. in. The initial heat content of the liquid ammonia is $i'_1 = 42.1$, and therefore the final heat content is $i'_2 + x_2r_2 = -12.6 + 556.3x_2 = 42.1$, whence $x_2 = (42.1 + 12.6)/556.3 = 0.0985$. The initial entropy is $s'_1 = 0.0813$. The final entropy is $s'_1 + (x_2r_2/T_2) = -0.0262 + 0.0985 \times 1.1599 = 0.0880$. $T_o = 20 + 460 = 480$; hence the loss of refrigerating effect is $480 \times (0.0880 - 0.0813) = 3.2$ B.t.u.

SECTION 3

FUELS AND COMBUSTION

FUEL OIL BURNING

BY

JOHN SHOBER BURROWS, Consulting Engineer for Castner, Curran & Bullitt; Mem. U. S. Fuel Administration, U. S. Geological Survey, U. S. Government Coal Testing Plant; Mem., A.S.T.M., Inter. Ry. Fuel Assn.

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CONTENTS

COAL

By J. S. BURROWS

	Page
Composition	241
Analysis	241–243
Classification	243
Marine Coals	243
Selection and Inspection of Coal for Marine Use	245
Spontaneous Combustion and Storage	245
U. S. Navy Coal	246
Standard Steaming	247
Sampling	247

OIL FUEL

By F. W. STERLING

Petroleum and Fuel Oil	250
Properties of Petroleum	251
Costs, Coal and Oil Fuel	252
Oil Fueling Stations	253
Advantages, Coal and Oil Fuel	254
Physical Properties	256
Heating Value	256
Viscosity	256
Viscosity Conversion Curves	260
Specific Gravity	262
Expansion	262
Specific Heat	267
Flash Point	268
Fire Point	269
Centrifuge	271
Conservation	273
Safety Precautions:	
Burning	274
Carrying	275
Loading	276

COMMERCIAL GASOLINE

By F. W. STERLING

Properties	277
Specifications	277
Tests	278

COMBUSTION

By F. W. STERLING

Air Required for	279
Excess Air	280
Volume Contraction	280
Combustion Products	280

OIL FUEL BURNING

By ALBERT M. PENN

Atomization and Atomizers	281
Air for Combustion	287
Air Registers	292
Boiler Furnace Design	297
Furnace Insulation	297
Relations Between Rate of Combustion, Heating Surface, Furnace Volume and Air Pressure	299
Fuel Oil Equipment	301
Boiler Room Instruments	315
Operation	322
Precautions	326

COAL

BY

J. S. BURROWS

REFERENCES: Publications of the U. S. Bureau of Mines on analyses of coal, combustion tests and firing methods. U. S. Geological Survey for descriptions and maps of U. S. coal fields. Bulletins of the Illinois Engineering Experiment Station on coal and combustion. Lewes, "Carbonisation of Coal." Dyson, "Practical Marine Engineering," Aldrich Publishing Co. The Coal Catalogue, Keystone Consolidated Publishing Co. for lists of producers. Welch Coal Annual, Cardiff, Wales, for foreign coals. Coal Age. "Black Diamond" and Coal Trade Journal for market reports.

Definition. Coal is an agglomerate of the solid degradation products of decayed vegetation which consisted of sedges, reeds, tree ferns, club mosses and trees akin to the pine, together with such of the original bodies as have resisted to a greater extent the actions of time, temperature and pressure to which the material was subjected in the transformation to coal.

Composition. The elementary **chemical constituents** of coal are carbon, hydrogen, oxygen, nitrogen and various mineral constituents which make up the ash. These elements are combined in coal in some of the most complex substances known to organic chemistry. The determination of the percentages of carbon, hydrogen, oxygen and nitrogen is known as an **"Ultimate Analysis."** This form of analysis is seldom necessary in practical work and is mainly used in chemical investigations of the organic substance of coal.

For fuel and other practical purposes, coal is considered as being composed of **moisture, volatile matter, fixed carbon, ash and sulphur,** a statement of the percentage of each of these items being known as a **"Proximate Analysis."**

Moisture consists of the natural water combined in or held mechanically in the coal particles before mining, plus such atmospheric moisture and rain as may have been added while in transit or in storage. The amount of moisture in a given coal varies greatly after it is mined, increasing or decreasing in a short period of time according to atmospheric conditions.

Volatile Matter consists of many complex combinations of mainly carbon, hydrogen and oxygen, which form during the early stages of the combustion process and which are manifest in the gases and tarry or smoky vapors issuing from the surface of a fuel bed. Under suitable conditions volatile matter can be burned to prevent smoke, but not all of the volatile matter is combustible.

Fixed Carbon is virtually coke, or that portion of the coal (minus the ash) which remains after distillation of volatile matter. In laboratory work, 100 minus the sum of the amounts of moisture, volatile matter and ash is arbitrarily termed the amount of fixed carbon. Thus all possible errors are thrown into this item, which, however, is of the least importance in fuel calculations or comparisons.

Ash consists of the noncombustible residue remaining after combustion is complete. Its composition and amount vary greatly, according to the origin and nature of the coal from which it is derived. There are two sources of ash: from the coal substance itself, which is known as **inherent ash** and from the enclosing rock strata of the coal bed and so-called "partings" of clay or slate within the bed, which is known as **extraneous ash.** The amount of inherent ash is usually fairly constant in the same coal, if of good grade. The amount of extraneous ash varies according to the care exercised in mining and preparation for market. Ash is the most important constituent to observe in coal, both because of its varying amount and because of the effect of its fusibility upon the steaming quality of the coal.

Clinkers consist of fused ashes, and result from subjecting the ash to a higher temperature than its fusing point. Analyses of ash, as ordinarily made, are of little value in judging clinkering properties, but it is now possible to determine the actual fusing point and thus select coals of the highest fusing temperatures. **Ash with fusing temperature above 2500 deg. F. seldom forms troublesome clinkers.**

Sulphur most commonly occurs in the form of **iron pyrites,** either in large, bright, metallic masses or small, brassy scales. With few exceptions its effect upon the fuel value of coal is negligible, and, while it is customary to give the amount in any analysis, it is of more interest to the metallurgist than to the marine engineer.

Calorific Value. The heating value of coal is stated in terms of British thermal units in America and England, and Calories in most European countries.

Many tables and formulæ have been devised for the calculation of the calorific value from proximate analyses, but these have proven satisfactory only on samples of coals from which the tables or formulæ were derived.

DuLong's Formula, which follows, is accurate within 150 B.t.u. of actual calorimeter test when reliable ultimate analyses are used:

$$\text{B.t.u. per lb. coal} = 14{,}600\text{C} + 62{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 4000\text{S},$$

where **C**, **H**, **O** and **S** represent the percentages respectively of carbon, hydrogen, oxygen and sulphur found by ultimate analysis. From this formula, which is the only recognized standard method of calculation, it is seen that the heating value of coal is derived from the carbon, hydrogen and sulphur, notwithstanding the fact that in actual practice these elements give up their heat while in various combinations with each other. The **calometric determination** of the B.t.u. value of coal should be entrusted only to an expert, and if results are to prove reliable these tests must be made with properly taken samples, page 247.

Interpretation of Analyses. Coals are customarily compared in terms of their analyses, and as there are three general modes of statement, the analyses must be on the same basis to be comparable. These three modes of statement are **"As Received,"** which shows actual analysis of sample as received at the laboratory; **"Dry Coal,"** which is recalculated from the actual analysis, eliminating the uncertain moisture element, and **"Moisture and Ash Free"** (**M & AF**) erroneously called **"combustible basis."** In this last both moisture and ash are recalculated out of the "As Received" basis for the purpose of showing the true proportions of volatile matter and fixed carbon, and the heating value per pound thereof.

Table 1. Typical Analysis of Coal, Illustrating Effect of Mode of Statement on Same Sample

	As received	Dry coal	Moisture and ash free
Moisture	8.29		
Volatile Matter	31.19	34.01	38.56
Fixed Carbon	49.69	54.18	61.44
Ash	10.83	11.81	
Total	100.00	100.00	100.00
Sulphur	2.81	3.06	3.47
B.t.u.	11,837	12,906	14,636

To calculate Dry Coal Analysis from As Received Basis, subtract the percentage of moisture from 100.00 and divide the result separately into volatile matter, fixed carbon, ash, sulphur and B.t.u. **To calculate the Moisture and Ash Free analysis** from Dry Coal Basis, subtract Dry Coal ash from 100.00 and divide the result into each of the other Dry Coal items as in the case of calculating Dry Coal analysis. The Dry Coal basis of analysis is the one generally used in engineering. It eliminates all uncertainty concerning the accuracy of the constantly changing moisture percentage. In any analysis moisture and ash have the same effect on B.t.u. value when determined by calorimeter. These simply act as dilutants of the combustible part of the coal. Starting with the M. & AF B.t.u. value (sometimes called **Pure Coal Value**), the addition of each per cent of ash or moisture reduces the M & AF B.t.u. value by an equivalent percentage. The M & AF B.t.u. value for many coals varies so slightly in a large series of analyses that for practical purposes an average may be regarded as a standard from which to determine approximately the "Dry" and "As Received" B.t.u. value of other samples in which the ash and moisture only are known. The M & AF basis is frequently of value in identifying coals from analyses which show extraordinary percentages of moisture and ash.

Classification of Coals. Coals are classified according to both their physical and chemical characteristics. No classification based entirely on chemical analyses is satisfactory, owing to the overlapping of the various classes or the gradual merging of one class into the other. The **fuel ratio, fixed carbon ÷ volatile matter**, is the most common basis of classification.

Table 2. General Classification of Coal

	Volatile matter per cent	M & AF B.t.u.	Principal characteristic
Anthracite	3 to 8	14,500 to 15,000	Hardness and distinctive analysis
Semi-anthracite	8 to 12		
Semi-bituminous	15 to 28	15,400 to 15,900	Low volatile, high B.t.u., friability
Bituminous	30 to 40	13,500 to 15,500	Wide variety
Sub-bituminous	30 to 45	13,000 to 13,500	High moisture, 10–25%
Lignite	40 to 60	11,000 to 13,000	Brown color, excessive moisture, 30–40%

Marine Coals. Semi-bituminous and bituminous are the coals best adapted to marine boilers. Table 3 lists the principal American bunker coals.

Semi-bituminous, also known as "**low volatile**" or "**smokeless,**" is a coking coal of first quality for marine use. Pocahontas and New River brands in this country and **British Admiralty coals** of South Wales belong in this class. In addition to its low volatile and high B.t.u. characteristics,

it is usually rather friable or soft, resulting in considerable slack or fine coal if much handled. This slack, however, cakes in the furnace.

Bituminous. This class embraces a wide variety of coals, both in analysis and physical characteristics. Owing to the long use of "bituminous" as a general name for any coal that was not distinctly anthracite, many coals are still thus designated which are as distinct in type as the semi- and sub-bituminous coals which are now separated from the general class. The designation may be taken to mean simply **high volatile** coal.

Sizes and Grades. Coal as mined consists of a mixture of irregular pieces, ranging from large lumps to fine dust. This mixture is known as **"Run of Mine"** and in America is the grade generally used for **bunker purposes,** as it is always in greatest supply at coaling ports. Run of Mine is separated into various sizes by passing it over and through screens of various meshes. There are no standard screens, consequently sizes vary widely in different districts and even in the same district. The most common screened sizes, in the order of their relative size, are "Lump," "Egg," "Nut" and "Slack."

At some mines it is the practice to separate the entire output into these sizes for convenience in removing impurities, the sizes being recombined later into Run of Mine. This **recombined Run of Mine** appears, while still in cars, to be poorly mixed, but the process of unloading into bunkers completes thorough mixing.

Washed Coal, whether as Run of Mine or screened sizes, is coal from which the heavy impurities constituting the extraneous ash have been removed in water by taking advantage of the difference in specific gravity of coal (about 1.28) and of impurities (from 1.4 to 2.0). The dry coal is fed to tanks, called jigs, through which water flows. A strong upward pulsation being given to the water, the good coal floats over the side of the jig and the impurities gradually work their way against the upward flow of water to the bottom, where they are removed at intervals.

Washed Run of Mine and slack, as loaded at the mines, contain a high percentage of moisture depending upon the amount of fine coal. This not only holds large quantities of water but also prevents rapid draining of the car. The lower ash of washed coal more than compensates for its higher moisture as compared with the same unwashed coal in the dry state.

Impurities in coal comprise "slate," "bone," "sulphur band," pyrites, rock or any other ash forming material which becomes mixed with the coal in the process of mining. Of these the most common are slate and bone.

Slate is a term widely used to cover most of the foreign material and correctly designates pieces of shale or fire clay from the roof or floor of the mine or from layers within the coal bed itself, known as **partings.** Slate can be easily distinguished by its color and flat shape when black, and is unmistakable when handled.

Bone is either smooth, shaley or rough, sandy material containing sufficient carbonaceous substance to give it the appearance of coal. It ranges from 25 per cent upward in ash. In many coal beds bone has the same fracture as the associated coal, and is difficult to distinguish. It usually has a dull luster as compared with its coal, and can be recognised by this or its greater weight.

Preparation is the process of removing the extraneous impurities from the product as it comes from the mines and is being loaded into railroad cars. The cleaning and loading plant is called a **tipple,** which, at modern operations, is equipped with wide conveyors or **picking tables** over which the coal is

carried in a thin layer toward the **loading booms** or chutes to the railroad cars. The large pieces of slate, bone or other impurities are removed by hand at the picking tables. Fine impurities can only be removed by washing, in which case the small sizes carrying the impurities are screened out and passed through a **washery**, the larger sizes being cleaned by hand. At many of the old operations the impurities are removed in the railroad car, a small amount of coal being dumped and raked over at a time. There are no standards of preparation other than market demands and competition.

Selection and Inspection of Coal for Marine Use. The objects to be gained in selecting a marine coal are (1) Maximum heat value or B.t.u. per cubic foot of bunker space, thereby increasing steaming radius or allowing more cargo to be carried; (2) semi-bituminous or low volatile coal, if obtainable, to minimize soot and tube sweeping; (3) high fusing temperature of ash (2450 deg. F. or higher) to reduce clinker troubles and frequent cleaning of fires; (4) sufficient lump coal in run of mine to start fires quickly.

These qualifications must be learned from experience with different coals, although the general reputation, description and analysis of a given coal will provide some means of judgment. The space required per ton of coal and the analysis can usually be obtained from the supplier, in which case a comparison under "1." of coals available may be made by the following formula:

$$\text{B.t.u. per cu. ft. Bunker space} = \frac{2240 \times \text{B.t.u. per lb. of coal}}{\text{cu. ft. per ton of coal}}$$

Coals vary considerable in space per ton occupied, ranging from about 40 to over 55 cubic feet per ton of 2240 lbs., and it is important to take this into consideration when comparing B.t.u. values.

The inspection of a lot of coal before delivery is made to the ship is not to be wholly relied upon as a means of determining its quality, but is of value in comparing different lots of a known coal and in showing in an unknown coal whether it (1) contains large pieces of slate or other impurities, (2) is coarse (containing a good percentage of lumps) or is fine (containing much slack) (3) is dry or wet.

Experience has shown that attempted estimates of the percentages of slate, lump and water based on visual inspection have proven to be mere guesses, and that the natural tendency of inexperienced inspectors is to base their estimates on the size of the lumps and pieces of impurities observable rather than on the amounts. When inspecting coal in railroad cars, it should be borne in mind that the surface of the coal is the only part visible and that it may not represent the entire carload, either with respect to impurities, percentage of lump or moisture. In a car of run of mine, rain seldom penetrates to a greater depth than a few inches, and coal which is dripping wet on top may be perfectly dry a foot below the surface. At some mines which make recombined run of mine, the arrangement of chutes is such that the coarse coal goes to the bottom of the car and the slack on top, giving the appearance of slacky coal, while at other operations the reverse is true, while again a good mixture of both may be found.

The absence of slate or other impurities from the top of a car may not indicate good preparation nor is the presence of a few large pieces scattered over the top sufficient evidence for condemnation, as both conditions may be purely accidental. The presence of impurities all over the surface of a car, however, is sufficient evidence of careless preparation to warrant the conclusion that the surface indications are representative of the entire car.

Spontaneous Combustion and Storage. The spontaneous combustion of coal is caused by the absorption of oxygen from the atmosphere with resultant evolution of heat, which, in a large mass or pile of coal (itself a poor heat conductor) accumulates until the temperature rises to the ignition point, when, if sufficient air circulates through the mass, the pile takes fire. The oxidation is greatly accelerated as the temperature rises and it is believed

type. Under favorable conditions with slightly superheated steam the mean value $x = 0.923$ was found. The corresponding loss of energy is $y = 0.148$ or 14.8 per cent. With high superheat and a velocity of 2200 ft. per sec., the value $x = 0.95$ was obtained, but at about 800 ft. per sec. x dropped to about 0.85. Rosenhain's experiments give values of x for several types of nozzles. Table 45 gives data and results (see Peabody's "Thermodynamics," p. 442).

Table 45. Values of the Velocity Coefficient x for Various Types of Nozzles

(Rosenhain, *Proc. Inst. C. E.*, vol. 140, p. 199)

Ratio of greatest to least diameter	Taper	Initial pressure	Velocity coefficient, x	Loss of energy, y
1.56	1:20	150.0	0.946	0.105
1.96	1:12	275.0	.....	.....
1.36	1:12	97.5	0.972	0.045
1.28	1:12	80.0	0.903	0.185
1.39	1:30	105.0	0.913	0.166
1.32	1:30	90.0	0.929	0.137
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CONTENTS

COAL

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	Page
Composition	241
Analysis	241–243
Classification	243
Marine Coals	243
Selection and Inspection of Coal for Marine Use	245
Spontaneous Combustion and Storage	245
U. S. Navy Coal	246
Standard Steaming	247
Sampling	247

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Petroleum and Fuel Oil	250
Properties of Petroleum	251
Costs, Coal and Oil Fuel	252
Oil Fueling Stations	253
Advantages, Coal and Oil Fuel	254
Physical Properties	256
Heating Value	256
Viscosity	256
Viscosity Conversion Curves	260
Specific Gravity	262
Expansion	262
Specific Heat	267
Flash Point	268
Fire Point	269
Centrifuge	271
Conservation	273
Safety Precautions:	
Burning	274
Carrying	275
Loading	276

COMMERCIAL GASOLINE

By F. W. STERLING

Properties	277
Specifications	277
Tests	278

COMBUSTION

By F. W. STERLING

Air Required for	279
Excess Air	280
Volume Contraction	280
Combustion Products	280

OIL FUEL BURNING

By ALBERT M. PENN

Atomization and Atomizers	281
Air for Combustion	287
Air Registers	292
Boiler Furnace Design	297
Furnace Insulation	297
Relations Between Rate of Combustion, Heating Surface, Furnace Volume and Air Pressure	299
Fuel Oil Equipment	301
Boiler Room Instruments	315
Operation	322
Precautions	326

OIL FUEL*

BY

FRANK W. STERLING

Petroleum, which is a variety of combinations of hydrocarbons of the paraffin, napthene and olefin series, is, because of its abundance and cheapness, the commercial **source** of all marine **oil fuel.** Shale, tar and creosote oils are not considered in this field to-day because of their greater value for other purposes.

There are three kinds of petroleums in use, namely, the so-called paraffin base, the semi-paraffin base, and the asphaltic base. To the first group belong the oils of the Appalachian field, comprising Pennsylvania, southern New York, West Virginia and eastern Ohio. To the second group belong the oils found in the Middle West, such as the Illinois, Kansas, and Oklahoma fields. To the third group belong the oils found on the Gulf Coast in Texas and Louisiana, in California and in Mexico.

The first two groups obtain their name from the fact that they contain paraffin wax and amorphous waxes such as petrolatum. The third group obtains its name from the fact that on distillation a residue of asphalt is left in the still. Table 1 shows the world's total marketed production of crude petroleum. Composition and calorific values of various petroleums are given in Table 2.

Fuel oil may be classed as topped petroleum, that is the residue after the more valuable and volatile constituents have been removed by distillation leaving a heavier and safer oil for burning purposes. Upon distillation, petroleum gives off in the order named, gasoline, kerosene and lubricating oils. The residue is fuel oil

Fuel oil, from the standpoint of the petroleum trade, generally includes all oils which are burned under boilers. Special distillates sold as Diesel Oils come under this head. Until 1914 much of the California crude oil was sold as fuel oil. The rapidly increasing value of the lighter distillates has caused the abandonment of this practice.

* In preparing the data for this section the writer has drawn freely on a paper, *Oil Fuel*, presented by *Mr. Ernest H. Peabody* before the *International Engineering Congress*, San Francisco, Cal.

Table 1. Petroleum Production in Barrels of 42 Gallons

Country	Year 1912	Year 1913	Year 1914	Year 1915	Year 1916	Total Years 1857–1916
United States	222,935,044	248,446,230	265,762,535	281,104,104	300,767,158	3,917,328,402
Russia	68,019,206	62,834,556	67,020,522	68,546,062	*a* 72,801,110	1,763,583,017
Mexico	16,558,215	25,902,439	21,188,427	32,910,508	39,817,402	163,087,779
Roumania	12,976,232	13,554,768	12,826,579	12,029,913	*a* 10,296,206	140,310,595
Dutch E. Indies	10,845,624	11,966,857	12,705,206	12,386,800	13,174,399	162,174,312
India	7,116,672	7,990,149	8,000,000	8,202,674	8,491,137	90,083,522
Galicia	8,535,174	7,818,130	*b* 5,053,350	4,158,899	6,461,706	142,494,206
Japan and Formosa	1,671,405	1,942,009	2,738,376	3,118,464	2,997,178	33,166,800
Peru	16,558,215	25,902,439	21,188,427	2,487,251	2,590,645	19,344,868
Germany	1,031,050	*b* 995,764	*b* 995,764	*b* 995,764	*b* 995,764	14,957,097
Other countries	*a* 1,044,824	*a* 1,143,587	*a* 2,294,924	*a* 1,797,690	2,547,266	32,676,197
Total World Production, 1857–1916						

a Estimated in part

b Estimated

Table 2. Properties of Petroleum (Ennis)

	Name and Source	Density			Ultimate Analysis				Prox. H_2O	B.t.u. per Pound			
		Sp. Gr.	°F.	°B6.	C.	H_2	O_2+N_2	S		By Calorimeter	By S. & K. Form	High Value	Low Value
1	Ogaio, crude	0.985	32	12.13	87.1	10.4	2.5			18,146	18,735	18,983	18,065
2	California, Whittier	0.9637	60	15.28				0.845	8.71	18,518	18,861		
3	California, Whittier	0.9629	60	15.39				0.84	8.82	18,596	18,866		
4	California and Bakersfield fuel	0.962	60	15.53	84.43	10.99	3.99	0.59			18,871	18,976	18,005
5	California crude	0.9572	60	16.24	86.3	16.7		0.8		18,646	18,900	21,723	21,254
6	Hanover	0.955	32	16.5	86.2	11.4	2.4				18,910	19,488	18,493
7	California crude	0.9533	60	16.85	85.75	11.3		0.67		18,797	18,924	19,356	18,363
8	California, Whittier and Los Angeles	0.953	60	16.9				0.98	4.93	18,714	18,926		
9	California, Whittier and Los Angeles	0.9529	60	16.91				0.955	4.62	18,754	18,926		
10	California	0.943	60	18.47				0.735	1.06	18,677	18,989		
11	California, Whittier	0.9417	60	18.67				0.975	1.06	18,626	18,997		
12	California	0.9410	60	18.78				1.010	0.74	18,705	19,001		
13	California	0.9407	60	18.83				0.96	0.42	18,657	19,003		
14	Baku Russia heavy	0.938	60	19.26	86.6	12.3	1.1			19,440	19,021	20,052	18,978
15	Borneo	0.936		19.58						18,831	19,033		
16	Petroleum, residue, Baku	0.928	32	20.95	87.1	11.7	1.2			19,832	19,088	19,761	18,739
17	Texas, Beaumont crude	0.924	60	21.56	84.60	10.90	2.87	1.63			19,112	18,977	18,025
18	Java	0.923	60	21.71	87.1	12.0	0.9			19,496	19,119	19,943	18,095
19	Texas, Beaumont	0.920	60	22.17	86.8	13.0	0.0				19,136	20,505	19,370
20	California	0.920	60	22.17	84.0	12.7	1.2				19,136	19,917	18,807
21	Pennsylvania	0.914	70	23.18	86.1	13.9		0.06			19,176	20,949	19,735
22	Texas, Beaumont, Richardson and Wallace	0.912	70	23.51	85.03	12.3	0.92	1.75			19,191	19,894	18,810
23	Pechelbronn	0.912	32	23.51	86.9	11.8	1.3			17,474	19,191	19,792	18,761
24	Hanover crude	0.892	32	26.95	80.4	12.7	6.9				19,328	19,393	18,283
25	Pechelbronn	0.892		26.95	85.7	12.0	2.3			18,036	19,328	19,739	18,691
26	Ohio distillate	0.887	60	27.84	84.2	13.1	2.7			18,718	19,364	20,188	19,044
27	Pennsylvania crude, heavy	0.886	32	28.01	84.9	13.7	1.4			19,210	19,370	20,654	19,457
28	Pennsylvania, heavy	0.886	60	28.01	84.9	13.7	1.04			19,224	19,370	20,654	19,457
29	West Galicia	0.885	32	28.09	85.3	12.6	2.1			18,416	19,374	20,045	18,944
30	Russian, crude, light	0.884	32	28.38	86.3	13.6	0.1			22,628	19,385	20,796	19,608

31	Balachany	0.882	32	28.73	87.4	12.5	0.1			21,060	19,400	20,289	19,198
32	West Virginia, heavy	0.873	60	30.37	83.5	13.3	3.2				19,464	20,207	19,046
33	Galicia, Austria	0.870	60	30.92	82.2	12.1	5.7			18,416	19,487	19,291	18,235
34	East Galicia	0.870	32	30.92	82.2	12.7	5.7			18,153	19,487	19,675	18,545
35	Kansas crude	0.866	60	31.67	85.4	13.07					19,517	20,345	19,203
36	Schwabweiler	0.861	32	32.60	86.2	13.3	0.5			18,644	19,555	21,200	19,439
37	Baku, light	0.884		35.87	86.3	13.6				20,628	19,685	20,796	19,608
38	West Virginia, light	0.8412		36.43	84.3	14.1	1.6			18,502	19,705	20,809	19,578
39	West Virginia, crude	0.841	60	36.47	84.3	14.1	1.6			18,400	19,709	20,809	19,578
40	West Virginia, light	0.841	60	36.47	84.3	14.1	1.6			21,240	19,709	20,809	19,578
41	Ohio, Mabery Noble	0.829	70	38.89	85.0	13.8	0.6	0.6			19,806	20,752	19,547
42	Pennsylvania, crude	0.826	60	39.50	82.0	14.8	3.2			17,930	19,830	20,699	19,606
43	Balakhany	0.822		40.32	87.4	12.5	0.1			21,600	19,863	19,269	19,198
44	American petroleum	0.82		40.73	83.4	14.7	1.9			17,388	19,879	21,042	19,758
45	Pennsylvania, light	0.816		41.57	82.0	14.8	3.2			17,533	19,913	19,899	19,606
46	Parma, Italy	0.786	60	46.13	84.0	13.4	1.8			18,215	20,175	20,341	19,171

The least valuable products of distillation, together with some residuum, constitute the fuel oil of to-day. The distillates of such low value as to be sold for fuel oil are usually the products distilling off after kerosene, and those too heavy for burning in lamps and too thin to be used as lubricating oil.

About 2/5 of the world's production of petroleum may be considered fuel oil.

Storage and Distributing Stations. The stations on the following pages, in the United States and Canada, are equipped with ample pumping capacities and pipe-line facilities for unloading tankers, loading tanks, cars, and fueling vessels, Table 3.

Relative Costs, Coal and Oil Fuels. The fuel cost of producing steam with oil and coal are the same when cost of coal in dollars per long ton = 1.907 × cost of oil in cents per gallon; or, as *Mr. Walter M. McFarland* first pointed out, **when the cost of coal in dollars per long ton is double the cost of oil in cents per gal., the fuel cost of producing steam will be approximately equal.** This assumes equal boiler capacity with oil and coal and that boiler efficiencies of oil and coal are as 75 to 68, and further assumes coal of 14,400 B.t.u. per lb. and oil of 18,600 B.t.u. per lb. with a weight of 7.88 lbs. per gal. (18° Baumé). In round numbers, as steam producers **1 lb. oil = 1½ lbs. coal, or 4½ bbl. oil = 1 ton coal.**

Fig. 1, charts *A* to *D*, from data by C. C. Moore & Co., furnish **a ready method of comparing fuel costs** for oil and coal. Charts *B* an *D* are constructed for 13,800 B.t.u. coal and 18,500 B.t.u. oil. Charts *A* and *C* are used for coals and oils of other heating values.

Assuming a coal costing $4.50 a ton, with a value of 13,800 B.t.u. and giving a boiler efficiency of 65 per cent, if the efficiency obtainable with an oil of 18,500 B.t.u. heat value is not known, it can safely be assumed that the net efficiency will be 4 per cent greater, or 69 per cent. On chart *B* follow the horisontal for 65 per cent efficiency with coal to the diagonal of 69 per cent net efficiency with oil (using the upper row of figures). 4.3 barrels of this oil is found to be equivalent to a ton of coal. On chart *D* follow the vertical for 4.3 barrels till it meets the $4.50 line for coal. The equivalent price for oil will then be $1.04 per barrel. If oil could be purchased for anything less than this, the fuel expense would be decreased by changing over.

Table 3. Partial List of Oil Distributing Stations in the United States (Peabody)

Place	Oil Company	Storage Capacity Bbl. of 42 Gal.
Aberdeen, Wash.	Standard Oil Co. of Cal.	34,000
Alameda Pt., Cal.	Associated Oil Co	37,500
Aransas Pass, Tex.	The Texas Company	55,000
Astoria, Ore.	Standard Oil Co. of Cal.	10,000
Avon, Cal.	Associated Oil Co	1,000,000
Baltimore, Md.	Interocean Oil Co	220,000
Baltimore, Md.	Standard Oil Co. of N. J.	50,000
Baltimore, Md.	The Texas Company	55,000
Baton Rouge, La.	Standard Oil Co. of La.	
Bayonne, N. J. (N. Y. Harbor)	Standard Oil Co. of N. J.	300,000
Bayonne, N. J.	The Texas Company	276,000
Bellingham, Wash.	Standard Oil Co. of Cal.	8,500
Boston, Mass.	Mexican Petroleum Co.	200,000
Boston, Mass.	Standard Oil Co. of N. Y.	50,000
Boston, Mass.	United States Navy	50,000
Carteret, N. J. (Near N. Y.)	Interocean Oil Co	100,000
Chester, Pa.	Interocean Oil Co	120,000
Charleston, S. C.	Standard Oil Co. of N. J.	15,000
Charleston, S. C.	The Texas Company	104,000
Charleston, S. C.	United States Navy	35,000
Cristobal, Pan.	The Texas Company	110,000
El Segundo, Cal.	Standard Oil Co. of Cal.	100,000
Eureka, Cal.	Standard Oil Co. of Cal.	35,000
Galveston, Texas	The Texas Company	110,000
Gaviota, Cal.	Associated Oil Co.	120,000
Guantanamo Bay, Cuba	United States Navy	212,000
Honolulu, H. I.	Associated Oil Co.	102,000
Honolulu, H. I.	Standard Oil Co. of Cal.	100,000
Jacksonville, Fla.	Standard Oil Co. of Ky.	50,000
Jacksonville, Fla.	The Texas Company	55,000
Ketchikan, Alaska	Standard Oil Co. of Cal.	38,000
Key West, Fla.	Standard Oil Co. of Ky.	50,000
Key West, Fla.	United States Navy.	35,000
Linnton, Ore. (Near Portland)	Associated Oil Co	173,000
Marcus Hook, Del.	Sun Oil Co.	500,000
Mare Island, Cal.	United States Navy	100,000
Melville Station, R. I.	United States Navy	85,000
Mobile, Ala.	The Texas Company	25,000
Monterey, Cal.	Associated Oil Co	340,000
Morgan City, La.	The Texas Company	55,000
New Orleans, La.	Mexican Petroleum Co.	400,000
New Orleans, La.	The Texas Company	124,000
New York, N. Y.	Mexican Petroleum Co.	500,000
New York, N. Y.	Standard Oil Co. of N. Y.	100,000
Norfolk, Va.	The Texas Company	138,000
Norfolk, Va.	United States Navy	185,000
Oakland, Cal.	Standard Oil Co. of Cal.	10,000
Panama	Mexican Petroleum Co.	100,000
Pearl Harbor, Hawaii	United States Navy	235,000
Philadelphia, Pa.	The Texas Company.	334,000
Point Breese, Pa.	Atlantic Refining Co.	500,000
Point Wells, Wash. (Near Seattle)	Standard Oil Co. of Cal.	200,000

Table 3. Partial List of Oil Distributing Stations in the United States (Peabody)—*Continued*

Place	Oil Company	Storage Capacity Bbl. of 42 Gal.
Port Arthur, Texas	The Texas Company	Practically unlimited
Port Chalmette, La.	Standard Oil Co. of La.	
Port Costa, Cal. (Carquines Strait)	Associated Oil Co	480,000
Portland, Me.	Mexican Petroleum Co. . . .	200,000
Portland, Me.	Standard Oil Co. of N. Y. . .	30,000
Portland, Ore.	Standard Oil Co. of Cal. . . .	85,000
Port Townsend, Wash.	Standard Oil Co. of Cal. . . .	27,000
Providence, R. I.	Mexican Petroleum Co. . . .	100,000
Providence, R. I.	Standard Oil Co. of N. Y. . .	10,000
Providence, R. I.	The Texas Company	253,000
Puget Sound, Wash.	United States Navy	100,000
Richmond, Cal.	Standard Oil Co. of Cal. . . .	200,000
Sacramento, Cal.	Associated Oil Co..	5,100
Sabine Pass, Texas	Sun Oil Company	250,000
San Diego, Cal.	Standard Oil Co. of Cal. . . .	34,000
San Diego, Cal.	United States Navy	100,000
San Francisco, Cal.	Associated Oil Co..	97,800
San Francisco, Cal.	Standard Oil Co. of Cal. . . .	15,000
San Pedro, Cal.	Standard Oil Co. of Cal. . . .	35,000
Seattle, Wash.	Standard Oil Co. of Cal. . . .	90,000
Stockton, Cal.	Associated Oil Co	5,200
Tacoma, Wash.	Standard Oil Co. of Cal. . . .	34,000
Tampa, Fla.	Standard Oil Co. of Ky. . . .	70,000

Fuel Oil Distribution Stations of the Imperial Oil Company, Limited

Fort William, Ontario	Prince Rupert, B. C.	Toronto, Ontario.
Halifax, N. S.	Quebec, Quebec.	Vancouver, B. C.
Montreal, Quebec.	Sarnia, Ontario.	Victoria, B. C.

Assume a coal of 14,400 B.t.u. giving an efficiency of 70 per cent, and an oil having a heat value of 6,323,000 B.t.u. per barrel. On chart *A* follow the vertical corresponding to 70 per cent efficiency for coal till it intersects the diagonal of 14,400 B.t.u. for coal. The intersection comes on the horisontal for 73 per cent, which is the equivalent efficiency of coal of 13,800 B.t.u. Use this on chart *B* as before, assuming 4 per cent, greater net efficiency with oil. 4.33 barrels of oil of 18,500 B.t.u. per lb., or 6,216,000 B.t.u. per barrel are found to be equivalent to one ton of coal. On chart *C* follow the vertical for 4.33 barrels to the diagonal for oil of 6,323,000 B.t.u. per barrel, obtaining 4.25 barrels of oil equivalent to one ton of the given coal. From chart *D*, using the vertical for 4.25 barrels, as the coal costs $4.50 per ton, $1.06 per barrel would be equivalent for the given oil.

Relative Advantages of Oil Fuel and Coal. The advantages of oil fuel are (1) reduced handling costs and ash handling eliminated, (2) reduced labor in the firerooms, (3) increased life of boiler and reduced maintenance costs in all departments, (4) increased steaming radius for same bunker capacity, or, for same cruising radius, saving in dead weight or cargo space, (5) increased efficiency and capacity, due to more constant furnace temperature and cleaner tubes, (6) time saved on voyage due to steadier steaming and shorter fueling time, (7) fueling at sea simplified (military), (8) smokeless

steaming possible (military), and (9) maximum rate of combustion can be attained instantly (military). Against these advantages must be charged the higher first cost of installation for oil burning.

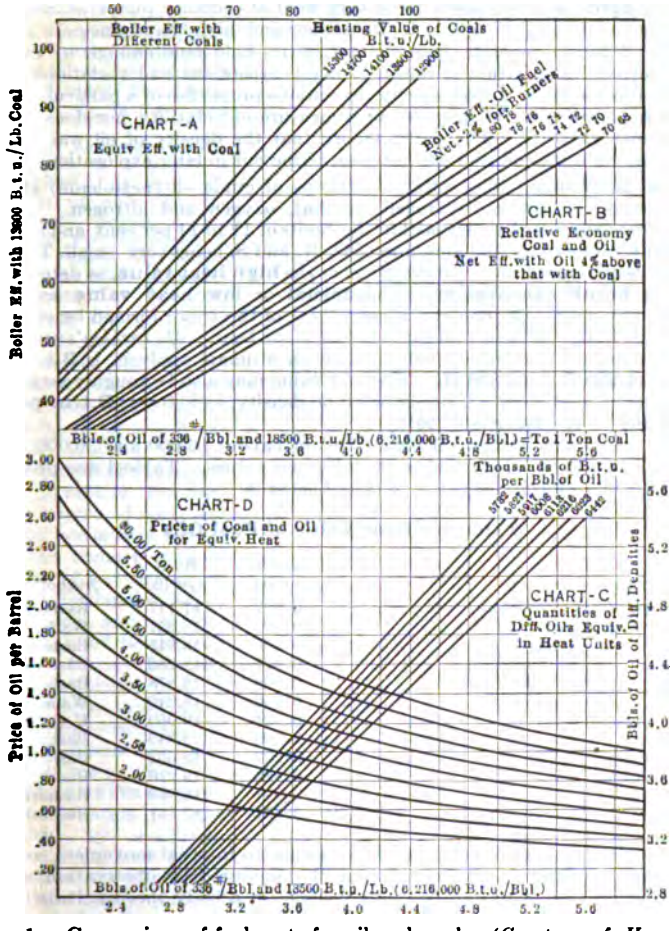

Boiler Eff. with Different Coals
Heating Value of Coals B.t.u./Lb.
CHART-A
Equiv Eff. with Coal
Boiler Eff. with 13500 B.t.u./Lb. Coal
Boiler Eff.-Oil Fuel Net-% for Burners
CHART-B
Relative Economy Coal and Oil
Net Eff. with Oil 4% above that with Coal
Bbls. of Oil of 336 #/Bbl. and 18500 B.t.u./Lb. (6,216,000 B.t.u./Bbl.) = To 1 Ton Coal
Thousands of B.t.u. per Bbl. of Oil
CHART-D
Prices of Coal and Oil for Equiv. Heat
Price of Oil per Barrel
CHART-C
Quantities of Diff. Oils Equiv. in Heat Units
Bbls. of Oil of Diff. Densities
Bbls. of Oil of 336 #/Bbl. and 13500 B.t.u./Lb. (6,216,000 B.t.u./Bbl.)

FIG. 1.—Comparison of fuel costs for oil and coal. (*Courtesy of Harrison Safety Boiler Works.*)

It has been conservatively estimated that the saving per annum of oil fuel burning over coal burning in a 3800 I. H. P. vessel is equal to over 25 per cent of the difference of first costs of installations.

Physical Properties. The classifications of oils according to their density is very commonly used to denote other characteristics. Thus a **heavy** oil is usually expected to be viscous and sluggish with a high percentage of asphalt and comparatively low heat value on a weight basis, while a **light** oil is supposed to be very fluid at ordinary temperatures, very volatile and rich in the lighter hydrocarbons and high heat value on a weight basis. While in general, these characteristics hold true enough to explain the prevalent association of ideas, so many exceptions and variations occur that is it essential to clearly specify the various properties of a particular oil in order to identify it. Density is not a measure of volatility, nor does weight determine viscosity. The early theory that the density of oil was related to the depth of the oil well has not been borne out in later exploration.

Heating Value. The ultimate analysis of crude oil (petroleum) gives a composition of carbon, hydrogen, sulphur, oxygen and nitrogen. The H and C contents usually vary within the limits of 11 to 14 per cent and 87 to 84 per cent respectively the amounts of O, N, and S, being very small, Table 2.

The heat value of oil is usually given as the **high heat value**, as determined by the **bomb calorimeter.** The **actual** or **low heat value** available for furnace use is less than this because heat of the vapor formed by combustion of H in the oil passes up the stack as waste heat. *M. Inchley's* empirical formula for the higher heat value, based on ultimate analysis, is **B.t.u. per lb. = 13,500 C + 60,890 H.** The heat value may also be roughly estimated by *Sherman and Kroff's* formula based on density, as follows: **B.t.u. per lb. = 18,560 × 40 (Baumé − 10).**

The heat value of fuel oil varies between about 16,000 and 20,000 B.t.u. per lb., 18,000 to 19,000 being the common values. Table 4 was prepared from the files of *The Babcock & Wilcox Company:*

Table 4. Heat Value of Fuel Oils

Source	Specific gravity	B.t.u per lb.	Authority
California, Coalinga Field	0.927	17,117	Bashore
" Bakersfield Field	0.992	18,257	Wade
" Kern River Field	0.950	18,845	Bashore
" Los Angeles Field	0.977	18,280	Bashore
" Monte Cristo Field	0.966	18,878	Bashore
" Whittin Field	0.936	18,240	Wade
Texas, Beaumont	0.924	19,060	U. S. Navy
" "	0.903	19,349	Bashore
" Sabine	0.937	18,662	Bashore
Pennsylvania	0.886	19,210	Booth
Mexico	0.921	18,840	Bashore
"	0.981	17,551	Bashore

Viscosity may be defined as the resistance to internal movement, or the **internal friction** of a liquid. It may be measured by observation of the ability of the liquid to oppose the movement of a body through it, or more commonly by noting the time required for a definite quantity of the liquid to pass through an orifice or short pipe under known conditions of temperature and head. In **stating viscosity** the **name of the instrument** used must be given, also the **temperature** at which it was run.

Absolute Viscosity may be defined as the force in dynes required to move a layer of the liquid of 1 sq. cm. area over another layer of the liquid with a velocity of 1 cm. per sec. It has not been found practical to actually measure viscosity by sliding plane surfaces

over each other. The most accurate method developed is that of observing slow flow of the liquid through **capillary tubes, Poisseuille's formula**, in which the viscosity is expressed as a function of the applied pressure and the quantity discharged, is based on investigations of capillary flow; it is,

$$\text{A. V.} = \frac{\pi p r^4}{8vl} t,$$ where A. V. = absolute viscosity, p = pressure in grams per sq. cm., r = radius of capillary tube in cm., l = length of tube in cm., v = volume of liquid passed through the tube in c.c., and t = time in sec.

The absolute viscosity of water at 0 deg. C. is 0.018086. If this were taken as unit and the absolute viscosity of other liquids were expressed in these units, the result might be called "**specific viscosity.**"

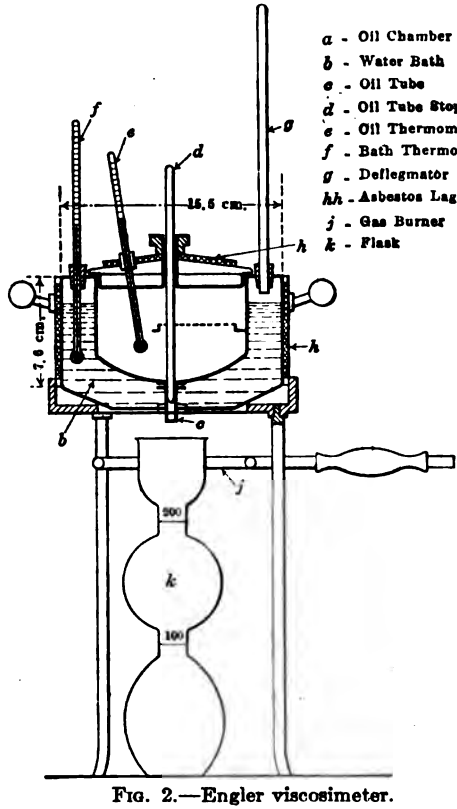


FIG. 2.—Engler viscosimeter.

With the advent of the viscous crude oils of Mexico and the increased use of heavy distillates from other fields, coupled with the wide adoption of the **mechanical atomizer** or **pressure burner**, the degree of viscosity of oil becomes a matter of the utmost importance. Most viscous oil must be heated before pumping, and the viscosity must be reduced before it can be used in the burner; in fact, the whole problem of handling viscous oils really hinges on viscosity, see p. 286.

Viscosimeters. Determination of viscosity by capillary tubes is the most accurate method as yet evolved. However, it is essentially a laboratory method, the apparatus involved is too delicate and complicated and the time required too long for commercial application.

In the **commercial viscosimeters**, such as the *Engler*, *Redwood*, and *Saybolt*, a small quantity of the liquid is permitted to flow through a cylindrical or conical opening that is short and relatively large in diameter (½ to ¾ in. long, approximately 0.01 in. diam.). The time of efflux of a given quantity is taken as indicating the viscosity. The accuracy of the capillary tube is sacrificed to obtain ruggedness and commercial rapidity. Flow through

short discharges is so rapid that turbulence and eddies with the lighter liquids cause nearly as much resistance as the actual viscosity of the fluid.

For identifying a liquid by comparison with a standard liquid the commercial viscosimeters are satisfactory, even though they may not indicate the true viscosity. For accurate study of the flow of liquids they cannot be relied upon.

The Engler Viscosimeter, Fig. 2, is specified for U. S. Navy fuel oil tests. It consists of an oil chamber *a*, with platinum tube outlet *c*, surrounded by water bath *b*. For high temperatures an oil bath is used. The platinum outlet is 20 mm. long, with a bore of 2.9 mm. diam. at top and 2.8 mm. at bottom. A volume of 200 c.c. of the oil to be tested is allowed to flow out and the time is noted in seconds. The number of seconds required for 200 c.c. water to flow out at the temperature of 20 deg. C. (68 deg. F.) is then determined (50 to 52 sec. in the standard instrument). Viscosity is found by dividing the time required for the outflow of oil by the time of outflow of water.

The Saybolt Universal Viscosimeter, Fig. 3, is specified for U. S. Navy fuel oil tests where the Engler instrument is not available. It is the most popular of the viscosimeters for lubricating oil testing. It is made entirely of metal. The cylinder *C* which holds 83 c.c. of oil, is surrounded by a large bath *B*. The upper edge of the cylinder is surrounded by a gallery *G*, into which any surplus oil flows. At the bottom of the cylinder is a jet *J*, through which the oil flows into the receiving flask *F*, which holds 60 c. c. to a mark on its neck. The jet is enclosed by the tube *T*, which extends below the orifice of the jet, and into the bottom of which a cork *L* with a string attached is lightly inserted, so as to close the tube. The bath is provided with stirrers, *S*, and fitted with a thermometer, for keeping the temperature uniform. The bath is heated by a gas ring burner *R*, which extends around in a circle under the bath, or by a steam U-tube or by an electric attachment.

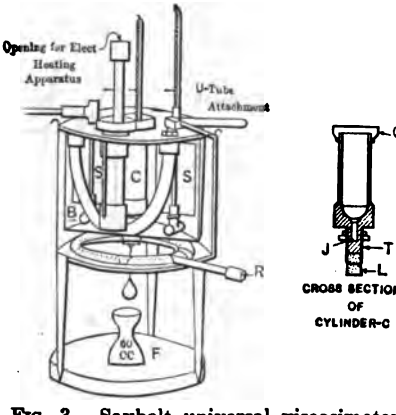


FIG. 3.—Saybolt universal viscosimeter.

To make the test, fill oil cylinder with oil to be tested, and fill bath with oil or water of the required temperature. The oil temperature is regulated by the bath temperature, the oil thermometer being used to stir the oil. When the oil is at the correct temperature, the bath temperature being a trifle higher, the thermometer is withdrawn and the gallery is emptied of all surplus oil by means of a pipette. The oil now exactly fills the cylinder. Flow of oil is started by quickly pulling string which withdraws cork *L*. Time of outflow of 60 c.c. of oil indicates the viscosity. Saybolt viscosimeters are so standardised that viscosity of water is 30 with this instrument.

The Redwood Viscosimeter, Fig. 4, consists of a silvered copper oil cylinder *A*, 1¼ in. diam. and 3½ in. long, fitted with an agate outlet jet. The cylinder is fixed in a copper water or oil bath *C* which is fitted with a heating tube *E* projecting down at an angle of 45 deg. *H* is an agitator. *T* and T_1 are thermometers for bath and oil cylinders respectively. The oil cylinder stopper consists of a small brass sphere which rests in a hemispherical cavity in the agate jet. *R* is an adjustable gauge to indicate the filling height of cylinder.

To make the test, fill bath with suitable liquid (water for temperatures to 200 deg. F. and mineral oil above that range) to a height corresponding to that of the gauge. When bath is at the required temperature the oil to be tested is poured into the cylinder until its level just reaches the gauge point. A gas flame may be applied to the heating tube to bring oil to the test temperature. When at this temperature the ball valve is

raised by the wire *V* and the time of outflow of 50 c.c. of oil is noted. The viscosity = $(F \times 100 \times G) \div (535 \times 0.915)$, where F = the number of sec. of outflow of 50 c.c. of oil and G is the specific gravity of the oil under test. It takes 535 sec. for 50 c.c. of rapeseed

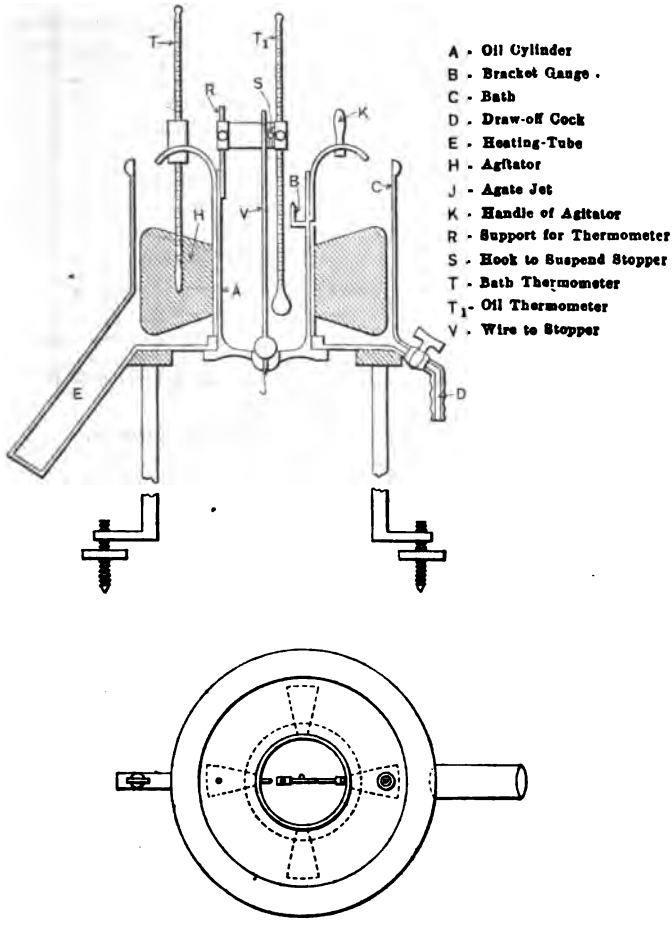


FIG. 4.—Redwood viscosimeter.

oil at 60 deg. F. to flow out at standard conditions and 0.915 is the sp. gr. of rape oil at 60 deg. F. The time of outflow of 50 c.c. of water at 60 deg. F. is 25.5 sec.

In the original standardising of the instrument, *slight* departures from the standard size of the orifice may be corrected by placing the gauge-point a little higher or lower. If, however, there be any considerable deviation from the model, especially in respect

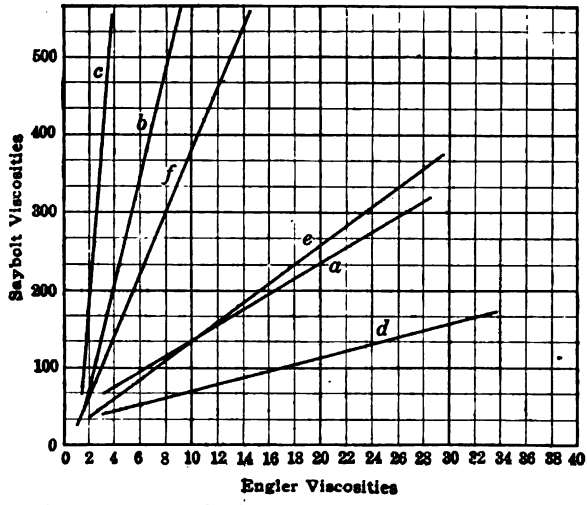


FIG. 5.—Engler-Saybolt viscosity conversion curves.

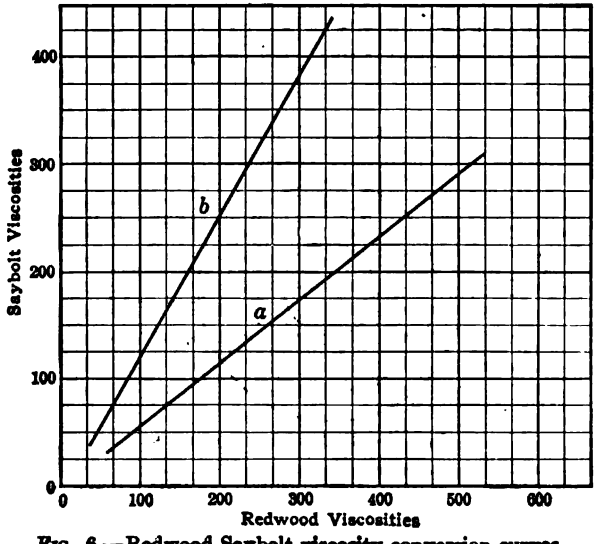


FIG. 6.—Redwood-Saybolt viscosity conversion curves.

of the height of the column of oil, the instrument may be standardised to give the same results as the pattern with a certain oil, but will not necessarily furnish concordant results with another oil of different viscosity.

The Admiralty Type Redwood Viscosimeter, a modification of the above, is the English standard for oil fuel. It has been specially designed for fuel oil work and is intended for use at 32 deg. F. The oil-vessel is of the same dimensions as that of the original instrument, but the agate jet is longer, and of larger bore, this viscometer being designed to give an outflow in one-tenth of the time occupied in the outflow of a similar volume from the original pattern. The jet is so mounted as to be completely surrounded by the water in the bath. The water-bath, which is of comparatively large size, and is jacketed, has no lateral heating-tube, but is provided with the usual rotating stirrer. Oil to be tested should be subjected to prolonged cooling at 32 deg. F., by being kept in a refrigerator through the night, or for at least six hours, immediately before being placed in the oil-cup, and should be thoroughly stirred before being transferred. When accurate results are desired, the viscometer should be placed in a refrigerating chamber, the temperature of which is maintained at 32 deg. F.

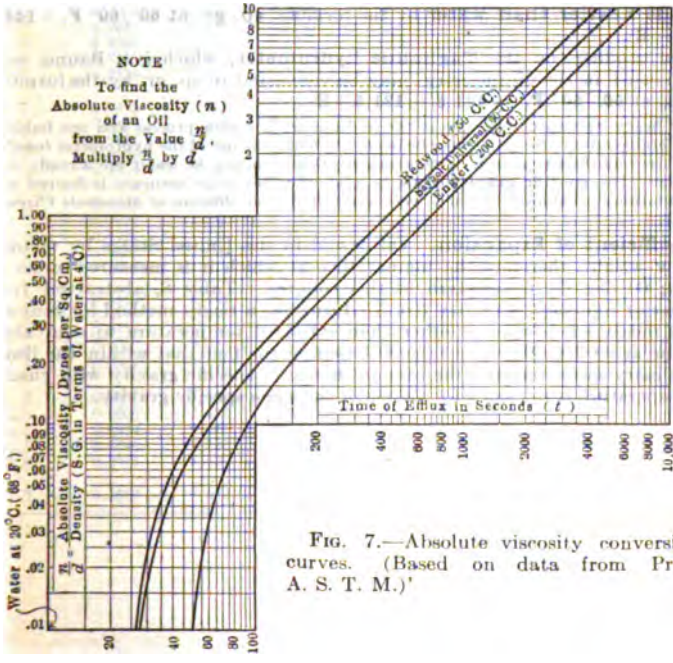

NOTE
To find the Absolute Viscosity (n) of an Oil from the Value $\frac{n}{d}$, Multiply $\frac{n}{d}$ by d
Redwood (50 c.c.)
Saybolt Universal (60 c.c.)
Engler (200 c.c.)
Time of Efflux in Seconds (t)
$\frac{n}{d}$ = Absolute Viscosity (Dynes per Sq. Cm.) / Density (S.G. in Terms of Water at 4°C)
Water at 20°C. (68°F.)

FIG. 7.—Absolute viscosity conversion curves. (Based on data from Proc. A. S. T. M.)

Conversion of Viscosity Scales. An exact relation between the different viscosity scales does not exist; it varies with different liquids, especially so with different oils, for the reasons above mentioned. Under the majority of working conditions, the error is only a few per cent. For practical purposes the curves given in Figs. 5, 6 and 7 are sufficiently accurate.

To Convert to Absolute Viscosity by Fig. 7, which curves were published in Standard Oil Bulletin by Standard Oil Co. of California: Assume an oil which at 70 deg. F. has a viscosity of 2200 Saybolt seconds and a density of 0.95. Follow dotted line verti-

cally from 2200 up to curve marked, "Saybolt Universal, thence horizontally to the left. "n/d" for this oil is 4.2; since "d" the density of the oil at 70 deg. F. is 0.95, its absolute viscosity "n" is $0.95 \times 4.2 = 4.0$.

Specific Gravity. The ease with which the density of oils lighter than water is determined by reading the scale of a Baumé hydrometer immersed in the liquid has resulted in the general adoption of this method. In tables of sp. gr. it is usual to show the temperatures, thus sp. gr. at 60°/60° F. indicates the sp. gr. of the oil at 60 deg. F. is referred to water at 60 deg. F. as unity. In the evolution of marking the Baumé scale, various makers deviated from time to time until the scale became somewhat confused. All Baumé scales in use to-day are based on an assumed relation between the Baumé scale and the sp. gr. scale at the same temperature.

The **U. S. Bureau of Standards** has determined that, for **liquids lighter than water,** the true relation between the Baumé gravity and the sp. gr. is expressed by the formula: **sp. gr. at 60°/60° F. = 140 ÷ (130 + B)**; and for **liquids heavier than water** by the formula, **sp. gr. at 60°/60° F. = 145 ÷ (145 − B)**.

The readings on the **Tagliabue hydrometer,** which is a Baumé scale much used in the oil industry, may be converted to sp. gr. by the formula: **sp. gr. at 60°/60° F. = 141.5 ÷ (131.5 + B)**.

For heavy viscous oils the use of the hydrometer is a slow process and one liable to considerable error. Such oils may be heated to make the use of the hydrometer feasible, due correction being made for temperature. Oils as heavy as water are already used as fuels. The **specific gravity scale** is used where extreme accuracy is desired as in computations. Tables 5, 6 and 7 were abstracted from *Bureau of Standards* Circular No. 57.

Coefficient of Expansion. Oil is sold in the United States by volume, not by weight, therefore the temperature at which it is measured must be given; 60 deg. F. is standard in this country. Table 8, abstracted from *Bureau of Standards* Circular No. 57, furnishes a ready method for converting a measured volume at other than standard temperature to equivalent volume at 60 deg. F. This table is based on the fact that within the limits of ordinary measurements the rate of change of specific gravity with change of temperature is the same for all oils of the same specific gravity.

Table 5. Degrees Baumé and Corresponding Sp. Gr. of Oil, Lbs. per Gal., and Gal. per Lb.

Degrees Baumé	Specific gravity at 60°/60° F.	Pounds per gallon	Gallons per pound	Degrees Baumé	Specific gravity at 60°/60° F.	Pounds per gallon	Gallons per pound
10.0	1.0000	8.328	0.1201	28.0	0.8861	7.378	0.1355
10.5	0.9964	8.299	0.1205	28.5	0.8833	7.355	0.1360
11.0	0.9929	8.269	0.1209	29.0	0.8805	7.332	0.1364
11.5	0.9894	8.240	0.1214	29.5	0.8777	7.309	0.1368
12.0	0.9859	8.211	0.1218				
				30.0	0.8750	7.286	0.1373
12.5	0.9825	8.182	0.1222	30.5	0.8723	7.264	0.1377
13.0	0.9790	8.153	0.1227	31.0	0.8696	7.241	0.1381
13.5	0.9756	8.125	0.1231	31.5	0.8669	7.218	0.1385
14.0	0.9722	8.096	0.1235	32.0	0.8642	7.196	0.1390
14.5	0.9688	8.069	0.1239				
				32.5	0.8615	7.173	0.1394
15.0	0.9655	8.041	0.1244	33.0	0.8589	7.152	0.1398
15.5	0.9622	8.013	0.1248	33.5	0.8563	7.130	0.1403
16.0	0.9589	7.986	0.1252	34.0	0.8537	7.108	0.1407
16.5	0.9556	7.959	0.1256	34.5	0.8511	7.087	0.1411
17.0	0.9524	7.931	0.1261				
				35.0	0.8485	7.065	0.1415
17.5	0.9492	7.904	0.1265	35.5	0.8459	7.044	0.1420
18.0	0.9459	7.877	0.1270	36.0	0.8434	7.022	0.1424
18.5	0.9428	7.851	0.1274	36.5	0.8408	7.001	0.1428
19.0	0.9396	7.825	0.1278	37.0	0.8383	6.980	0.1433
19.5	0.9365	7.799	0.1282				
				37.5	0.8358	6.960	0.1437
20.0	0.9333	7.772	0.1287	38.0	0.8333	6.939	0.1441
20.5	0.9302	7.747	0.1291	38.5	0.8309	6.918	0.1446
21.0	0.9272	7.721	0.1295	39.0	0.8284	6.898	0.1450
21.5	0.9241	7.696	0.1299	39.5	0.8260	6.877	0.1454
22.0	0.9211	7.670	0.1304				
				40.0	0.8235	6.857	0.1459
22.5	0.9180	7.645	0.1308	40.5	0.8211	6.837	0.1463
23.0	0.9150	7.620	0.1313	41.0	0.8187	6.817	0.1467
23.5	0.9121	7.595	0.1317	41.5	0.8163	6.797	0.1471
24.0	0.9091	7.570	0.1321	42.0	0.8140	6.777	0.1476
24.5	0.9061	7.546	0.1325				
				42.5	0.8116	6.758	0.1480
25.0	0.9032	7.522	0.1330	43.0	0.8092	6.738	0.1484
25.5	0.9003	7.497	0.1334	43.5	0.8069	6.718	0.1489
26.0	0.8974	7.473	0.1338	44.0	0.8046	6.699	0.1493
26.5	0.8946	7.449	0.1342	44.5	0.8023	6.680	0.1497
27.0	0.8917	7.425	0.1347				
				50.0	0.7778	6.476	0.1544
27.5	0.8889	7.402	0.1351	50.5	0.7756	6.458	0.1548

Table 6. Temperature Corrections to Apparent Degrees Baumé of Petroleum Oils

This table gives the corrections to be subtracted from the apparent degrees Baumé of heavy petroleum oils (fuel oils, lubricating oils, etc.) at temperatures from 60 deg. to 210 deg. F. to give the true degrees Baumé at 60 deg. F. (modulus, 140). It is assumed that the hydrometer is of glass having a coefficient of cubical expansion of 0.000023 per deg. C., and is correct at 60 deg. F.

(For more complete oil tables see Circular 57, Bureau of Standards)

Observed temperature, deg. F.	Observed degrees Baumé											
	14	16	18	20	22	24	26	28	30	32	34	36
	Subtract from observed degrees Baumé to obtain true degrees Baumé at 60 deg. F.											
60	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
62	.1	.1	.1	.1	.1	.1	.1	.1	.1	.1	.1	.1
64	.2	.2	.2	.2	.2	.2	.3	.3	.3	.3	.3	.3
66	.3	.3	.3	.3	.3	.3	.4	.4	.4	.4	.4	.4
68	.4	.4	.4	.5	.5	.5	.5	.5	.6	.6	.6	.6
70	.5	.5	.5	.6	.6	.6	.6	.6	.7	.7	.8	.8
72	.6	.6	.6	.7	.7	.7	.7	.7	.8	.8	.9	.9
74	.7	.7	.7	.8	.8	.8	.9	.9	.9	.9	1.0	1.1
76	.8	.8	.8	.9	.9	.9	1.0	1.0	1.1	1.1	1.2	1.2
78	.9	.9	.9	1.0	1.1	1.1	1.2	1.2	1.2	1.2	1.3	1.4
80	1.0	1.0	1.1	1.1	1.2	1.2	1.2	1.3	1.3	1.3	1.4	1.5
82	.....	1.1	1.2	1.2	1.3	1.3	1.3	1.4	1.4	1.5	1.5	1.6
84	.....	1.3	1.3	1.3	1.4	1.4	1.5	1.5	1.5	1.6	1.7	1.8
86	.....	1.4	1.4	1.4	1.5	1.5	1.6	1.6	1.7	1.8	1.8	1.9
88	.....	1.5	1.5	1.6	1.6	1.7	1.7	1.8	1.8	1.9	2.0	2.0
90	.....	1.6	1.6	1.7	1.7	1.8	1.8	1.9	2.0	2.0	2.1	2.1
92	.....	1.7	1.7	1.8	1.8	1.9	1.9	2.0	2.1	2.1	2.2	2.3
94	.....	1.8	1.8	1.9	1.9	2.0	2.0	2.1	2.2	2.2	2.3	2.4
96	.....	1.9	1.9	2.0	2.0	2.1	2.2	2.3	2.3	2.4	2.5	2.5
98	.....	2.0	2.0	2.1	2.2	2.2	2.3	2.4	2.4	2.5	2.6	2.7
100	.....	2.1	2.2	2.2	2.3	2.3	2.4	2.5	2.6	2.7	2.7	2.8
105	.....	2.4	2.4	2.5	2.6	2.6	2.7	2.8	2.9	3.0	3.1	3.2
110	.....	2.6	2.7	2.8	2.8	2.9	3.0	3.1	3.2	3.3	3.4	3.5
115	.....	2.9	2.9	3.0	3.1	3.2	3.3	3.4	3.5	3.6	3.8	3.9
120	.....	3.1	3.2	3.3	3.4	3.5	3.6	3.7	3.8	3.9	4.0	4.2
125	.....	.....	3.4	3.5	3.6	3.7	3.8	4.0	4.1	4.2	4.3	4.5
130	.....	.....	3.7	3.8	3.9	4.0	4.1	4.3	4.4	4.5	4.7	4.8
135	.....	.....	3.9	4.1	4.2	4.3	4.4	4.6	4.7	4.8	5.0	5.2
140	.....	.....	4.2	4.3	4.4	4.6	4.7	4.8	5.0	5.1	5.3	5.5
145	.....	.....	4.4	4.6	4.7	4.8	5.0	5.1	5.3	5.4	5.6	5.8
150	.....	.....	4.7	4.8	5.0	5.1	5.2	5.4	5.6	5.7	5.9	6.1
155	.....	.....	4.9	5.1	5.2	5.4	5.5	5.7	5.9	6.0	6.2	6.4
160	.....	.....	.....	5.3	5.5	5.6	5.8	6.0	6.2	6.3	6.5	6.7
165	.....	.....	.....	5.6	5.7	6.1	6.3	6.3	6.5	6.6	6.8	7.0
170	.....	.....	.....	5.8	6.0	6.2	6.3	6.5	6.7	6.9	7.1	7.3
175	.....	.....	.....	6.0	6.2	6.4	6.6	6.8	7.0	7.2	7.4	7.6
180	.....	.....	.....	6.3	6.5	6.6	6.8	7.1	7.3	7.5	7.7	8.0
185	.....	.....	.....	6.5	6.7	6.9	7.1	7.3	7.6	7.8	8.0	8.3
190	.....	.....	.....	6.8	7.0	7.2	7.4	7.6	7.8	8.1	8.3	8.6
195	.....	.....	.....	7.0	7.2	7.4	7.6	7.9	8.1	8.4	8.6	8.9
200	.....	.....	.....	.....	7.5	7.7	7.9	8.1	8.4	8.6	8.9	9.2
205	.....	.....	.....	.....	7.7	7.9	8.2	8.4	8.7	8.9	9.2	9.5
210	.....	.....	.....	.....	8.0	8.2	8.4	8.7	8.9	9.2	9.5	9.8

Table 7. Temperature Corrections to Apparent Sp. Gr. of Petroleum Oils

This table gives the correction to be added to apparent specific gravities of heavy petroleum oils (fuel oils, lubricating oils, etc.,) at temperatures of 60 deg. to 210 deg. F. to give the true specific gravity of the oil at 60°/60° F. It is assumed that the hydrometer or pienometer used is of glass having a coefficient of cubical expansion of 0.000023 per deg. C., and is correct at 60 deg. F. (For more complete oil tables, see Circular 57, Bureau of Standards).

Observed temperature, deg. F.	Observed specific gravity											
	0.850	0.860	0.870	0.880	0.890	0.900	0.910	0.920	0.930	0.940	0.950	0.960
	Add to observed specific gravity to give true specific gravity at 60°/60° F.											
60	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
62	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001	0.001
64	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002
66	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002
68	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003
70	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
72	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004	0.004
74	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005	0.005
76	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006
78	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006	0.006
80	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007	0.007
82	0.008	0.008	0.008	0.008	0.008	0.007	0.007	0.007	0.007	0.007	0.007	0.007
84	0.009	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008
86	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009	0.009
88	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
90	0.011	0.011	0.011	0.011	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
92	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011	0.011
94	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012	0.012
96	0.013	0.013	0.013	0.013	0.013	0.013	0.012	0.012	0.012	0.012	0.012	0.012
98	0.014	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013	0.013
100	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014	0.014
105	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016	0.016
110	0.018	0.018	0.018	0.018	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.017
115	0.020	0.020	0.020	0.020	0.019	0.019	0.019	0.019	0.019	0.019	0.019	0.019
120	0.022	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021	0.021
125	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.023	0.022	0.022	
130	0.025	0.025	0.025	0.025	0.025	0.024	0.024	0.024	0.024	0.024	0.024	
135	0.027	0.027	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	
140	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.028	0.027	0.027	
145	0.030	0.030	0.030	0.030	0.030	0.030	0.029	0.029	0.029	0.029	0.029	
150	0.032	0.032	0.032	0.031	0.031	0.031	0.031	0.031	0.031	0.031	0.031	
155	0.034	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.033	0.033		
160	0.035	0.035	0.035	0.035	0.035	0.035	0.034	0.034	0.034	0.034		
165	0.037	0.037	0.037	0.037	0.036	0.036	0.036	0.036	0.036	0.036		
170	0.039	0.039	0.038	0.038	0.038	0.038	0.038	0.038	0.038	0.037		
175	0.040	0.040	0.040	0.040	0.040	0.040	0.039	0.039	0.039	0.039		
180	0.042	0.042	0.042	0.041	0.041	0.041	0.041	0.041	0.041	0.041		
185	0.044	0.044	0.043	0.043	0.043	0.043	0.043	0.043	0.042			
190	0.045	0.045	0.045	0.045	0.045	0.044	0.044	0.044	0.044			
195	0.047	0.047	0.047	0.047	0.046	0.046	0.046	0.046	0.046			
200	0.049	0.049	0.048	0.048	0.048	0.048	0.048	0.048	0.047			
205	0.051	0.050	0.050	0.050	0.050	0.050	0.049	0.049	0.049			
210	0.052	0.052	0.052	0.051	0.051	0.051	0.051	0.051	0.051			

Table 8. Expansion Table for Oil Fuel

This table, from Bureau of Standards Circular No. 57 shows volume that would be occupied at 60 deg. F. by a quantity of oil, of various specific gravities, occupying unit volume at the designated temperatures. Example: 1 gal. oil measured at 98 deg. F. having sp. gr. 0.85 at this temperature, will occupy volume of 0.988 gal. at 60 deg. F. "Observed temperature," and "observed sp. gr." refer to true indications of instruments corrected for instrumental errors.

Observed temperature in deg. F.	Observed specific gravity								
	0.800	0.810	0.820	0.830	0.840	0.850	0.860	0.870	0.880
	Volume at 60 deg. F. occupied by unit volume at various temperatures								
30	1.016	1.015	1.015	1.014	1.014	1.014	1.013	1.013	1.013
32	1.014	1.014	1.014	1.013	1.013	1.013	1.012	1.012	1.012
34	1.013	1.013	1.013	1.012	1.012	1.012	1.011	1.011	1.011
36	1.012	1.012	1.011	1.011	1.011	1.011	1.010	1.010	1.010
38	1.011	1.011	1.010	1.010	1.010	1.010	1.009	1.009	1.009
40	1.0105	1.0100	1.0095	1.0095	1.0095	1.0090	1.0090	1.0090	1.0085
42	1.0095	1.0090	1.9090	1.0085	1.0085	1.0080	1.0080	1.0080	1.0075
44	1.0085	1.0080	1.0080	1.0075	1.0075	1.0075	1.0070	1.0070	1.0070
46	1.0075	1.0070	1.0070	1.0065	1.0065	1.0065	1.0065	1.0060	1.0060
48	1.0060	1.0060	1.0060	1.0060	1.0055	1.0055	1.0055	1.0050	1.0050
50	1.0050	1.0050	1.0050	1.0050	1.0045	1.0045	1.0045	1.0045	1.0045
52	1.0040	1.0040	1.0040	1.0040	1.0035	1.0035	1.0035	1.0035	1.0035
54	1.0030	1.0030	1.0030	1.0030	1.0025	1.0025	1.0025	1.0025	1.0025
56	1.0020	1.0020	1.0020	1.0020	1.0020	1.0020	1.0015	1.0015	1.0015
58	1.0010	1.0010	1.0010	1.0010	1.0010	1.0010	1.0010	1.0010	1.0010
60	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
62	.9990	.9990	.9990	.9990	.9990	.9990	.9990	.9990	.9990
64	.9980	.9980	.9980	.9985	.9985	.9985	.9985	.9985	.9985
66	.9970	.9970	.9970	.9975	.9975	.9975	.9975	.9975	.9975
68	.9960	.9960	.9960	.9965	.9965	.9965	.9965	.9965	.9965
70	.9950	.9950	.9950	.9955	.9955	.9955	.9955	.9960	.9960
72	.9940	.9945	.9945	.9945	.9945	.9945	.9945	.9950	.9950
74	.9930	.9935	.9935	.9935	.9940	.9940	.9940	.9940	.9940
76	.9920	.9925	.9925	.9925	.9930	.9930	.9930	.9935	.9935
78	.9910	.9915	.9915	.9915	.9920	.9920	.9920	.9925	.9925
80	.990	.990	.990	.991	.991	.991	.991	.991	.992
82	.989	.989	.989	.990	.990	.990	.990	.991	.991
84	.988	.988	.989	.989	.989	.989	.989	.990	.990
86	.987	.987	.988	.988	.988	.988	.989	.989	.989
88	.986	.987	.987	.987	.987	.987	.988	.988	.988
90	.985	.986	.986	.986	.987	.987	.987	.987	.987
92	.984	.985	.985	.985	.986	.986	.986	.986	.987
94	.983	.984	.984	.985	.985	.985	.985	.985	.986
96	.982	.983	.983	.984	.984	.984	.984	.985	.985
98	.981	.982	.982	.983	.983	.983	.984	.984	.984
100	.980	.981	.981	.982	.982	.982	.983	.983	.983
102	.979	.980	.980	.981	.981	.982	.982	.982	.983
104	.979	.979	.980	.980	.981	.981	.981	.981	.982
105	.978	.978	.979	.979	.980	.980	.980	.981	.981
108	.977	.977	.978	.978	.979	.979	.980	.980	.980
110	.976	.976	.977	.977	.978	.978	.979	.979	.979
112	.975	.976	.976	.977	.977	.978	.978	.978	.979
114	.974	.975	.975	.976	.976	.977	.977	.977	.978
116	.973	.974	.974	.975	.975	.976	.976	.977	.977
118	.972	.973	.973	.974	.974	.975	.975	.976	.976
120	.971	.972	.973	.973	.974	.974	.975	.975	.976

Table 8. Expansion Table for Oil Fuel.—*Continued*

Observed temperature in ° F.	Observed specific gravity						
	0.890	0.900	0.910	0.920	0.930	0.940	0.950
	Volume at 60° F. occupied by unit volume at various temperatures						
30	1.013	1.012	1.012	1.012	1.012	1.012	1.011
32	1.012	1.011	1.011	1.011	1.011	1.011	1.011
34	1.011	1.010	1.010	1.010	1.010	1.010	1.010
36	1.010	1.010	1.009	1.009	1.009	1.009	1.009
38	1.009	1.009	1.009	1.008	1.008	1.008	1.008
40	1.0085	1.0080	1.0080	1.0080	1.0080	1.0080	1.0080
42	1.0075	1.0075	1.0075	1.0070	1.0070	1.0070	1.0070
44	1.0070	1.0065	1.0065	1.0065	1.0065	1.0060	1.0060
46	1.0060	1.0060	1.0060	1.0055	1.0055	1.0055	1.0055
48	1.0050	1.0050	1.0050	1.0050	1.0050	1.0045	1.0045
50	1.0040	1.0040	1.0040	1.0040	1.0040	1.0040	1.0040
52	1.0035	1.0035	1.0035	1.0030	1.0030	1.0030	1.0030
54	1.0025	1.0025	1.0025	1.0025	1.0025	1.0025	1.0025
56	1.0015	1.0015	1.0015	1.0015	1.0015	1.0015	1.0015
58	1.0010	1.0010	1.0010	1.0010	1.0010	1.0010	1.0005
60	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
62	.9990	.9995	.9995	.9995	.9995	.9995	.9995
64	.9985	.9985	.9985	.9985	.9985	.9985	.9985
66	.9975	.9980	.9980	.9980	.9980	.9980	.9980
68	.9970	.9970	.9970	.9970	.9970	.9970	.9970
70	.9960	.9960	.9960	.9960	.9960	.9960	.9965
72	.9950	.9955	.9955	.9955	.9955	.9955	.9955
74	.9945	.9945	.9945	.9945	.9945	.9945	.9945
76	.9935	.9935	.9935	.9935	.9935	.9940	.9940
78	.9925	.9930	.9930	.9930	.9930	.9930	.9930
80	.992	.992	.992	.992	.992	.992	.992
82	.991	.991	.991	.991	.991	.991	.991
84	.990	.990	.990	.990	.990	.990	.991
86	.989	.989	.990	.990	.990	.990	.990
88	.988	.988	.989	.989	.989	.989	.989
90	.988	.988	.988	.988	.988	.988	.988
92	.987	.987	.987	.987	.987	.987	.988
94	.986	.986	.986	.986	.987	.987	.987
96	.985	.985	.985	.986	.986	.986	.986
98	.985	.985	.985	.985	.985	.985	.985
100	.984	.984	.984	.984	.984	.984	.985
102	.983	.983	.983	.983	.984	.984	.984
104	.982	.982	.982	.983	.983	.983	.983
106	.981	.981	.982	.982	.982	.982	.983
108	.981	.981	.981	.981	.981	.982	.982
110	.980	.980	.980	.980	.981	.981	.981
112	.979	.979	.979	.980	.980	.980	.981
114	.978	.978	.978	.979	.979	.979	.980
116	.977	.978	.978	.978	.978	.979	.979
118	.976	.977	.977	.977	.978	.978	.978
120	.976	.976	.976	.976	.977	.977	.978

Specific Heat of Oil varies with its composition. It increases with an increase of hydrogen content and decreases with increase of carbon content. The following figures, table 9, are reproduced from *Holde's* work on *Examination of Hydrocarbon Oils.*

Table 9. Specific Gravity and Specific Heat of Crudes

Crude Oils From—	S. G.	S. H.
Japan	.862	.453
Pennsylvania	.810	.500
Russia	.908	.435
California	.960	.398
Bustenari	.842	.462

Attention is drawn to the fact that these specific heats are on a weight basis. According to this table California oil would heat up much quicker

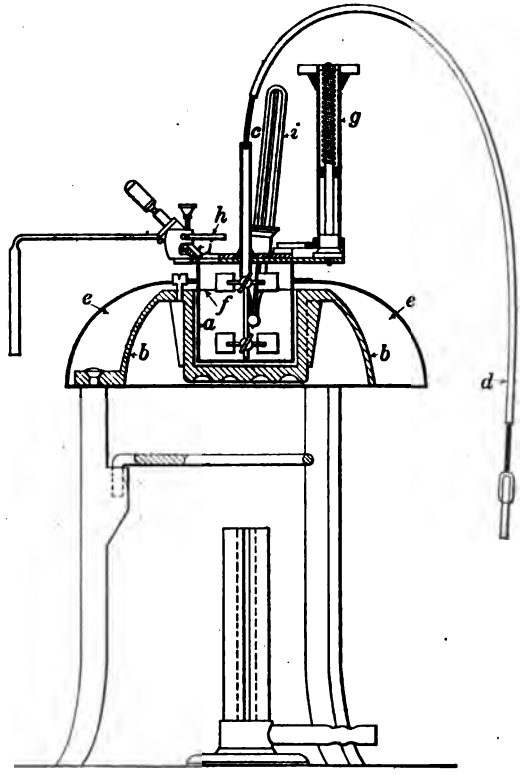


FIG. 8.—Pensky-Martens closed-cup tester. (*a*) oil cup; (*b*) air jacket; (*c*) stirrer shaft; (*d*) stirrer; (*r*) brass mantle; (*f*) graduation mark; (*g*) flame adjuster; (*h*) safety light; (*i*) thermometer; (*j*) pilot light.

than Pennsylvania oil. This difference, however, would be corrected largely by difference in specific gravity.

The Flash Point of an oil is the temperature at which the oil, upon being heated slowly, gives off sufficient vapor to form over its surface,

with the air present, an explosive mixture. It is determined by heating the oil and, as the temperature rises, testing it with a spark or flame until the flash occurs. Oil may be stirred (as in the *Pensky-Marten* tester) in order to get a uniform temperature and this temperature must be accurately measured. Conditions of the test may introduce wide differences in results. Thus, merely by closing in the top of the test vessel, the flash will be detected sooner and at a lower temperature than if the vessel is open, as in

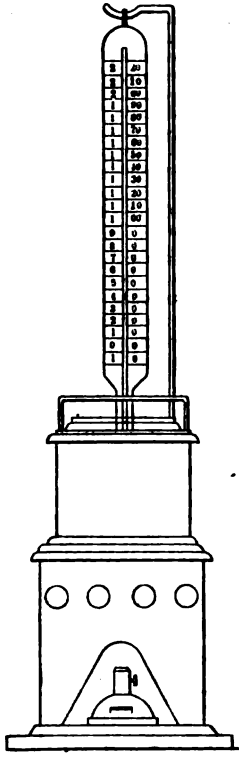


FIG. 9.—Tagliabue open tester.

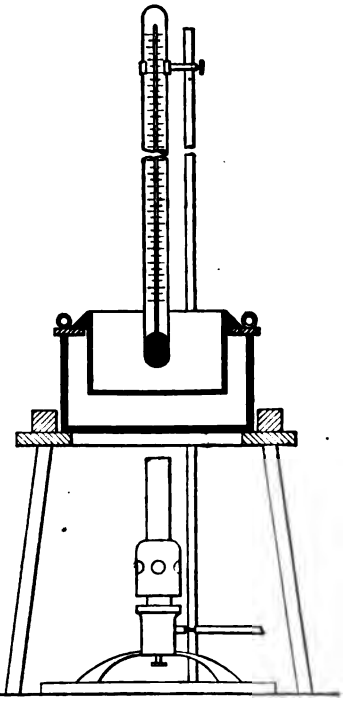


FIG. 10.—Cleveland open tester.

the Tagliabue method. The former is the **closed cup method** and the latter is the **open cup**. The closed cup is the more accurate method and instruments devised by *Pensky-Marten* and by *Abel* are recognized as standards for fuel oil.

The Fire Point, or burning point, is the temperature at which sufficient vapor is given off to remain ignited after flashing. As a free supply of oxygen is required, this test is made with the open cup.

Testing for Flash Point. In the **Pensky-Marten Apparatus**, Fig. 8, *a* is the oil container which can be closed by a tightly fitting lid. It is placed in a heating air

jacket *b* which is provided with a brass mantle *e* to reduce radiation losses. The stirring paddles are operated by the stirring shaft *c*, which passes through the lid.

By turning *g*, an opening of the slide can be made to coincide with one of several orifices in the cover, and simultaneously a jet of flame *J* from the test jet, is tilted on the surface of the oil.

Operation. All water must be removed from the oil before testing by filtering through one of the small felt filters of the outfit. When the sample is ready, the oil cup is filled to mark *f*, cover is closed and the oil is rapidly heated until its temperature is about 50 deg. F. below the expected flash point. Wire gauze screen, Fig. 8, is then put in place and the rate of rise in temperature is retarded to about 5 deg. per min. Handle *d* is turned continuously and slowly. At intervals of 5 deg. rise in temperature, reduced to 2 deg. when near the expected flash point, *g* is turned, thus opening the shutter and tilting flame into cup. The flash point is the temperature at which a slight explosion occurs. A sample can be used for but one test, since at the first test some of the more volatile products are driven off thus raising the flash point of the remaining oil.

FIG. 11.—Hand centrifuge.

Testing for Fire Point. The Tagliabue, Fig. 9, and Cleveland, Fig. 10, Open Testers are best adapted for this work.

Water and Sediment. Though water is slightly soluble in oils, for practical purposes the engineer is only interested in the water present in **mechanical suspension.** Water and sediment are generally reported combined in percentage by volume but it is much better to record them separately.

To determine the percentage of water and sediment in any oil, mix 50 c.c. of the oil under test with 50 c.c. of benzol in a **centrifuge graduate,** Figs. 11 and 12. Then centrifuge until a clear line of demarkation is visible between the oil, water and sediment.

Showing interchangeable heads

FIG. 12.—Electric centrifuge.

The oil may be thinned with **gasoline,** but this is objectionable as many of the asphaltic hydrocarbons are insoluble in gasoline and are therefore precipitated as sediment, but its use will give a true indication of the presence of water.

Carbon disulphide is recommended by some authorities as preferable to benzol for thinning the oil, as the latter will not dissolve the free carbon in suspension, a proportion of which is therefore deposited as sediment.

The most accurate method of determining percentage of water and sediment is by means of distillation in the presence of an excess of hydrocarbons, but this is essentially a laboratory method.

If no centrifuge is available a 50 c.c. sample of oil thinned with 50 c.c. of benzol may be allowed to stand for several hours in a slender graduate. Water and sediment will settle to the bottom *to some extent*, but a true reading is not obtained in this way. This method will only indicate when large quantities of water or sediment are present. The following tests, 50 c.c. of oil known to contain 2.0 per cent of water and sediment being used for test, were conducted at the Navy Fuel Oil Testing Plant to show the relative efficiency of settling test with and without centrifuging:

	Stood for 15 hours, per cent of water and sediment	Heated and allowed to stand 6 hours, per cent of water and sediment
With benzol	0.2	0.2
With gasoline	0.0	0.0
With kerosene	0.0	0.0
With benzol and centrifuged	2.0	

The Centrifuge, as employed for centrifugal separation of water and sediment from oil, can be obtained as a hand or power operated machine. In either case it consists of a vertical spindle, capable of rotation, which carries a head for test tubes. Oil to be tested is placed in the tubes. As the head is revolved, the test tubes which are held in gimbal-like collars assume horizontal positions and the water and sediment are thrown to the bottom of the tube by the centrifugal action.

The Eimer and Amend hand operated instrument, Fig. 11, has a gear ratio between handles and revolving head of 15 to 1. It can be operated at 1500 R.P.M. with two 15 c.c. graduated tubes charged. This type is supplied to ships. The laboratory instrument, Fig. 12, is an electrically driven machine made by the *International Instrument Co.* It has a maximum speed of 3000 R.P.M. and can be supplied with a variety of heads as illustrated. Tubes of 15 c.c. and 50 c.c. can be used with this machine.

Specifications. The following are specifications as adopted by the Committee on Standardization of Petroleum Specifications, U. S. Fuel Administration, October 2, 1918:

Methods of Test. (*a*) Flash point will be taken as indicated in the specifications.

(*b*) Viscosity will be taken by the Engler Viscosimeter. (See note under "Specifications.")

(*c*) Water and sediment will be taken by the distillation method. When oil in small lots is consigned to naval vessels or to navy yards the centrifuge test will be used in order to obviate delay. In this test 30 c.c. of oil and an equal quantity of best commercial benzol, 50 per cent white, will be used, and the mixture heated to 100 deg. F.

Navy Standard Fuel Oil. (*a*) Fuel oil shall be a hydrocarbon oil free from grit, acid, and fibrous or other foreign matter likely to clog or injure the burners or valves. If required by the Navy Department, it shall be strained by being drawn through filters of wire gauze having 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe and strainers shall be in duplicate.

(*b*) The unit of quantity to be the bbl. of 42 gal. of 231 cu. in. at a standard temperature of 60 deg. F. For every decrease or increase of temperature of 10 deg. F. (or proportion thereof) from the standard, 0.4 of 1 per cent (or prorated percentage) shall be added or deducted from the measured or gauged quantity for correction.

(c) The flash point shall not be lower than 150 deg F. as a minimum (Abel or Pensky-Marten's closed cup) or 175 deg. F. (Tagliabue open cup). In case of oils having a viscosity greater than 8 Engler at 150 deg. F. the flash point (closed cup) shall not be below the temperature at which the oil has a viscosity of 8 Engler.
(d) Viscosity shall not be greater than 40 Engler at 70 deg. F.
(e) Water and sediment not over 1 per cent. If in excess of 1 per cent, the excess to be subtracted from the volume; or the oil may be rejected.
(f) Sulphur not over 1.5 per cent.

NOTE. If the Engler viscosimeter is not available, the Saybolt Standard Universal viscosimeter may be used. Equivalent viscosities:
8 Engler = 300 seconds Saybolt.
40 Engler = 1500 seconds Saybolt.

Special Specifications for Gas Oil for Diesel Engines. 1. Flash point not lower than 150 deg. F. (Abel or Pensky-Marten's closed cup).
2. Water and sediment, trace only.
3. Asphaltum, none.

Bunker Oil "A." To comply strictly with the provisions for Navy specifications fuel oil, except that there shall be no limit on sulphur.

Bunker Oil "B." Specifications to be the same as for Navy Standard fuel oil except:
(c) Omit and substitute "The Flash point shall not be lower than 150 deg. F. as a minimum Abel or Pensky-Marten's closed cup) or 175 deg. F. (Tagliabue open cup).
(d) Omit and substitute "To have a minimum gravity of 18 Baumé.
(f) Omit.

Bunker Oil "C." Specifications to be the same as for bunker oil "B" except it is to have a gravity of approximately 16 Baumé.

Uses of Various Grades. Navy standard fuel oil only will be supplied to battleships, destroyers and other vessels subject to heavy forced draft conditions or required to run smokeless. It will also be supplied for cargo oil for all shipments abroad or to Navy storage.

Bunker Oil "A" will be used by other types of vessels requiring a light oil and by shore stations fitted with separate storage for yard use. It will not not be used where bunker oil "B" or "C" can be satisfactorily used.

Bunker Oil "B" will be used by all transports and cargo vessels which can satisfactorily burn an oil not heavier than 18 Baumé gravity. It will not be used where bunker oil "C" can be satisfactorily used.

Bunker Oil "C" will be used by all transports and cargo vessels which can satisfactorily burn an oil of approximately 15 Baumé gravity.

Oil Measurement. For contract purposes and bunker and tank measurement the standard unit is the U. S. gallon (231 cu. in.). The litre (1 gal. = 3.785 litres) and the imperial gallon (4.54 litres) are used abroad. U. S. bbl. = 42 U. S. gallons. Imperial bbl. = 50 U. S.

Volumetric measurement of oil should always be based on a standard temperature (in U. S. A., 60 deg. F.), See Coef. Exp., p. 262. The Navy Department specifies 60 deg. F., and a correction of 0.4 of 1 per cent is made for each 10 deg. variation from this standard. Tank Soundings and oil meters, page 318, are the practical measuring agents used on shipboard.

For the calculation of evaporative results, fuel used for power, heat value, etc., units of weights are employed, *i.e.*, lb., kilo, etc.

Corrosion Due to Oil. The much prophesied deleterious effects of sulphur in oil on boiler tubes has not developed in practice. In cases of neglect, pitting may occur, under certain conditions, but will not exceed in magnitude the same effects with coal of equal sulphur contents under the same conditions. However, high sulphur oils will attack copper and for

this reason fuel oil pipes and heating coils are made of steel. Brass and bronze fittings in pumps and pipes lines are quite safe.

Compartments and tanks used for the storage of fuel oil should not be painted on the inside, but should be protected from corrosion either by the oil itself or by some special coating.

The inside of compartments and tanks used for carrying fuel oil shall be inspected every twelve months and the plating or bulkheads separating fuel oil compartments from others shall be carefully examined for leaks during the quarterly inspection and also each time oil is taken on board.

Conservation. Generally speaking, conservation is best promoted by using oil of the highest viscosity and the highest apparent sulphur content that will give the desired results. This is apparent from the specifications on page 271, which specifications were primarily drawn up in the interest of conservation.

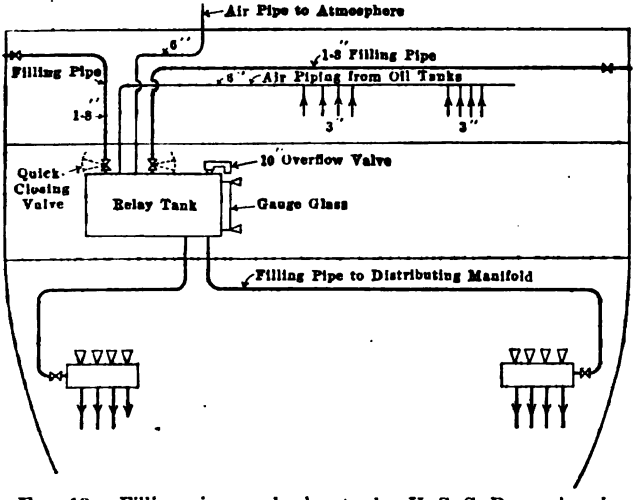


FIG. 13.—Filling pipes and relay tanks, U. S. S. Pennsylvania.

Storage on Board Ship. Whereas coal must be stored adjacent to firerooms, fuel oil is adaptable to storage in almost any part of the vessel, in bunker spaces, tanks remote from firerooms, and double bottoms. While easily handled by pumps, the feasibility or its use on shipboard depends primarily in the ability to keep it where stored until pumped to boilers. Fitting oil tight tanks to steamers increases the structural cost over coal burners.

The tanks must be fitted with swash plates, if deep, expansion trunks to allow for increase in volume due to heating, vent pipes carried above the main deck and fitted with goosenecks and gauze screens to liberate oil vapors, and sounding pipes for measuring depth of oil in the tanks. These should be of ample size, not less than 2½ in., and small holes should be drilled in these

pipes to give free access to the oil at all depths, as it is possible for some difference in density to exist at various levels, if the oil has been standing for some time. To prevent undue pressure being put on closed tanks, due to too rapid filling, the combined area of the vents and sounding pipes must be sufficient to provide adequate overflow. For this reason, when tanks in the lower part of the vessel, such as the double bottoms, are used for oil, a system of relay tanks, Fig. 13, is fitted to eliminate danger of a large head of oil exerting heavy pressure on the storage tanks.

A filling pipe on each side of the ship runs to the relay tank. The latter is fitted with a removable cover and a large overflow closed by a relief valve with a very light spring. Filling pipes lead from the relay tank to the storage tanks, and the greatest pressure that can be put on the latter is that due to the head from the relay tank, which can be reduced to a small amount by suitably locating such tank. The vent pipes from the storage tanks lead into the relay tank, which is fitted with a common vent pipe leading to the atmosphere and covered at the end with wire gauze. The supply pipes to the relay tanks are fitted with quick closing valves and the relay tank is equipped with a gage glass to mark the level of oil. Also an annunciator at the relay tank, operating from pneumercators fitted in the storage tanks, gives warning when the latter are 95 per cent full.

The use of fittings of any kind on the outside of tanks below the oil level is very bad practice. The storage tank has a high and low suction, *i.e.*, two pipes either separate or connected through a manifold, one taking the oil from a level 12 or 18 inches from the bottom of the tank and the other from a point within a few inches of the bottom. The upper suction is used for regular service, and at all times except in emergency or when the supply is very low or when the low suction is employed to pump overboard water or very dirty oil which has accumulated at the bottom of the tank.

SAFETY PRECAUTIONS FOR OIL-BURNING VESSELS

Fuel oil is inert, non-explosive, very difficult to ignite in bulk, and not capable of spontaneous combustion. However, the **vapor from this oil is explosive** when mixed with air. This vapor is heavier than air and tends to accumulate in low levels, such as bilges and bottoms of tanks; it is always present in a partly filled oil tank, or in one that has contained fuel oil and from which the vapor has not been removed by artificial means. It is expelled through the vents from fuel oil tanks while they are being filled. **Ignition** may be caused by an open light, electric spark, or spark made by striking metal, heat of the filament of a broken electric lamp, smoking, sparks from smoke pipe or galley, or fire under boilers. An oil fire may be extinguished by dry sand, steam, or chemical extinguishers, but not by water.

While oil is being received on board no naked light, smoking, or electrical apparatus liable to spark should be permitted within 50 feet of an oil hose, tank, compartment containing a tank, or the vent from a tank. Storage tanks should be closely watched for leaks.

Precautions against Danger on Shipboard. Oil in bulk is perfectly safe so long as its flash point is more than 60 deg. F. above the temperature to which it is exposed. Not being subject to spontaneous combustion, oil bunker fires are practically unknown at sea.

The U. S. Navy, in coöperation with the Bureau of Mines, investigated the matter of possible explosion of gases in storage tanks, and it was found that no inflammable gases were formed in any amount in the storage tanks or bunkers until the oil was heated to the flash point. It was also found that any oil in the bunker tank had to be heated to

within 60 deg. F. of the flash point before even a faint glow or partial burning was obtained on introducing a naked flame.

No person should be allowed to **enter a fuel oil** tank until it has been freed of gas by the use of water or steam, and then anyone entering such tank should have a life line around his body, properly tended, in order that he may be hauled out if overcome by gas.

All **bunkers must be ventilated** by vents, the openings of which must be lead well away from possible exposure of flame. No smoking should be allowed in the vicinity of the vent. **Wire gauze protectors** in these vents must be kept intact. Bunker bulkheads exposed to heat must be insulated. Bunkers should never be more than 95 per cent. full, to allow for expansion of the oil. Bunkers and fire rooms should be fitted with tank type steam or chemical **smothering apparatus**; see p. 327.

No naked light, smoking, or electrical fuses, switches (unless enclosed type), or other apparatus liable to spark should be permitted in a compartment containing a fuel oil tank or fuel oil pumps or piping, except that smoking may be permitted in the engine rooms and on fireroom floor in front of boilers which are in operation. Electric lamps, see p. 1372, used in such compartments should have steam-tight globes, or be of a type that will insure breaking the circuit through the lamp in case the bulb breaks.

The **valves on glass gauges** fitted to the storage or settling tanks should be kept habitually shut. Valves may be opened to read the gauge but they should be shut again immediately thereafter.

Whenever the **fuel oil tanks** are to **remain empty** for any length of time, or whenever they are to be entered, or whenever any work is done in them requiring heated rivets, hammering, etc., or any lights other than portable electric (which lights should be tested before use), or whenever such work is done in the vicinity of open tanks or pipes, all such tanks and all pipes leading thereto should be cleared of vapor, after the fuel oil has been removed, by blowing through live steam, or air by means of portable blower, for at least 12 hours.

When vessels carrying fuel oil are **in dry dock**, no oil should be allowed to drain into the dock. Should it be necessary to remove oil from tanks or receptacles on vessels in dry dock, such precautions should be taken as will prevent any of the oil reaching the floor of the dock or escaping so as to permit the accumulation of explosive vapors in the dock.

SAFETY PRECAUTIONS FOR VESSELS CARRYING OIL CARGOES

Vessels carrying oil cargoes should observe all rules and safety precautions prescribed for oil-burning vessels, see p. 274, and in addition the following special rules should be followed:

While oil is on board no **smoking**, except in officers' rooms and crew's spaces, or naked lights should be allowed on board. Waste, oilskins, paints, or other **combustible material** should not be stored in pump rooms. Oily waste should be destroyed by burning under the boilers. The use of other than **safety matches** should be prohibited.

All access hatches and manholes to fuel oil tanks should be painted red and fitted with locks, and **lable plates** should be secured thereto, stating that the hatch affords access to an oil tank.

Electrical apparatus should be frequently inspected and any condition liable to lead to sparking remedied at once. Fuses, switches, etc., between

decks or in other than officers' rooms and crew's spaces, should be protected by flame proof wire gauze.

Protable electric lights, if used, should be fitted with steam-tight globes and heavy wire guards or other suitable devices to prevent the ignition of vapor from the heat of the filament in the event of breaking the bulb.

When loading or discharging cargo or when freeing the tanks of gas, doors to boiler rooms and living quarters and all air ports should be closed; galley fires should be extinguished, if practical, otherwise the doors and openings to galley should remain closed; there should be no fire in donkey boiler; tugs or other steam vessels should not be allowed alongside when hatches are open. When loading or discharging at night, no lights should be carried around on deck. Water-tight clusters should be suspended in the rigging at a safe distance from deck, and the wires should not be dragged across the deck. During the period of loading no naked lights or smoking should be allowed on board ship or within 50 feet of the oil hose or vent discharges from the tanks.

When filling tanks they should be carefully watched to prevent leaks, and should such leaks develop, the loading must at once cease and steps should be taken to stop the leaks without generating excessive heat or causing sparks.

COMMERCIAL GASOLINE

BY

FRANK W. STERLING

All figures relative to boiling point, specific gravity, composition, etc., must be comparative, for naturally the product varies with the original crude oil. Approximately the range of **distillation temperatures** for commercial gasoline is 115 to 350 deg. F. At the lower temperature gasoline is distilled off, then, as the temperature is increased, follow benzine, naphtha and light kerosene in the order named. Commercial gasoline may contain any or all of these fractions.

Its **specific gravity** varies from 0.65 to 0.75, depending upon the proportions of C and H in its composition, and it weighs about 6.2 lbs. per gal. The **analysis** of an ordinary sample shows approximately C 85 per cent, H 14.8 per cent, impurities (principally O) 0.2 per cent. Its **net thermal value** varies with the analysis around 18,000 B.t.u. per lb.

The ultimate **value of a gasoline as fuel** depends upon its volatility. For this reason the popular specific gravity test is not a true criterion. For instance a high speed engine needs a light fuel, easily volatilized, while a heavy-duty, slow-speed motor can use a much heavier fuel. Were the entire supply of gasoline derived from one field, fractions obtained at the same temperatures would always have the same composition and hence the same specific gravity. But, as the world's supply is obtained from many fields in which the compositions vary, it is possible to obtain two gasolines of widely differing specific gravities, which will distil at the same temperature and which might be of equal value as fuels.

The volatility of two gasolines being equal, the heavier is more efficient due to the presence of a higher percentage of carbon. To prepare gasoline for combustion it must be vaporized by passing air over or through the liquid, of by spraying the liquid into the air by force or suction; this process is called carburetion.

Specifications for Motor Gasoline were adopted by the Committee on the standardization of Petroleum Specifications, U. S. Fuel Administration October 2, 1918, as follows:

Quality. Gasoline to be high grade, refined, and free from water and all impurities, and shall have a **vapor tension** not greater than 10 lbs. per sq. in. at 100 deg. F., same to be determined in accordance with the current "Rules and regulations for the transportation of explosives and other dangerous articles by freight"-par. 1824 (k) as issued by the Interstate Commerce Commission.

Inspection and Tests. Inspection. Before acceptance the gasoline will be inspected. Samples of each lot will be taken at random. These samples immediately after drawing will be retained in a clean absolutely tight closed vessel and a sample for test taken from the mixture in this vessel directly into the test vessel.

Test. One hundred cubic centimeters will be taken as a test sample. The apparatus and method of conducting the distillation test shall be that described in Bureau of Mines Technical Paper No. 166, Motor Gasoline.

(*a*) Boiling point must not be higher than 60 deg. C. (140 deg. F.)

(*b*) 20 per cent of the sample must distil below 105 deg. C. (221 deg. F.)

(*c*) 45 per cent must distil below 135 deg. C. (275 deg. F.).

(*d*) 90 per cent must distil below 180 deg. C. (356 deg. F.).

(*e*) The end or dry point of distillation must not be higher than 220 deg. C. (428 deg. F.).

(*f*) Not less than 95 per cent of the liquid will be recovered from the distillation.

COMBUSTION

BY

FRANK W. STERLING

Combustion is a rapid exothermic or heat liberating chemical combination of oxygen and fuel. That portion of the fuel so combined, *i.e.*, C, H, and S, is the combustible. In marine engineering practice, **combustible** includes all constituents of fuel except moisture and ash, thus including the O and N fuel contents.

Perfect Combustion occurs when the maximum possible amount of oxygen unites with the combustible as in the formation of CO_2. Partial combustion takes place when incomplete oxidation occurs, as in the formation of CO. C and H are the only chemical elements of heat value, the single exception being S, which exists in fuel as an impurity, and which has a small heat value.

The Products of Complete Combustion are correspondingly few in number, being CO_2, H_2O, N of the air, and a small amount of sulphur compounds that are often ignored.

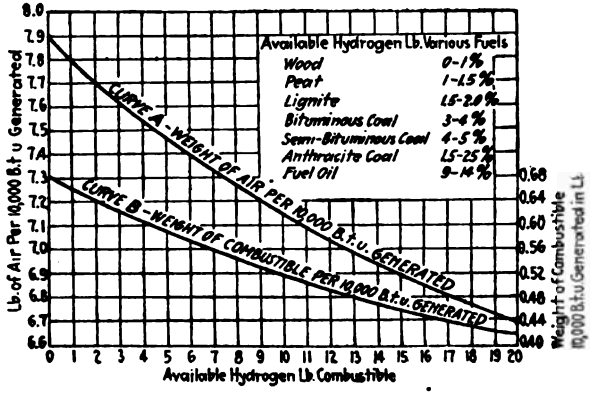

Available Hydrogen Lb. Various Fuels
Wood 0-1%
Peat 1-1.5%
Lignite 1.5-2.0%
Bituminous Coal 3-4%
Semi-Bituminous Coal 4-5%
Anthracite Coal 1.5-2.5%
Fuel Oil 9-14%
CURVE A - WEIGHT OF AIR PER 10,000 B.T.U. GENERATED
CURVE B - WEIGHT OF COMBUSTIBLE PER 10,000 B.T.U. GENERATED
Lb. of Air Per 10,000 B.t.u Generated
Weight of Combustible 10,000 B.t.u Generated in Lb.
Available Hydrogen Lb. Combustible

FIG. 1.

Air Required for Combustion. Let c, h, and o, denote respectively the parts by weight of carbon, hydrogen and oxygen in one lb. of the fuel. Then the minimum weight of oxygen required for complete combustion is $11.6[c+3\ (h-o+8)]$ lbs.

With air at 62 deg. F. and at atmospheric pressure, the minimum volume of air required is $v_m = 147\ [c+3(h-o+8)]$ cu. ft.

Fig. 1, Curve A, gives the weight of air required to generate 10,000 B.t.u. with fuels of different H content. The abscissæ are the per cent of available H per lb. of combustible. The available H = per cent H −

per cent o + S. Per cent of available H = (per cent H − per cent o + 8) ÷ per cent H. This in commercial coals varies from 2.5 to 5.

Curve B gives the weight of combustible to generate 10,000 B.t.u. To obtain weight of air theoretically required per lb. of fuel, multiply the heating value of the fuel by the weight of air theoretically required to generate 10,000 B.t.u., as obtained from the curve, and divide by 10,000. The heating value of contained S is disregarded in these curves, its percentage being small.

Excess Air. In practice an excess of air over that required for perfect combustion is admitted to the furnace. An excess of air is required (1) because it is impossible to bring into intimate contact every particle of air and combustible due to dilution of the O in the air by N, and because of uneven thickness of fires, etc., and (2) because an appreciable time is required for combustion according to the law of *mass action*. With only the theoretical amount of air present, as combustion proceeds in a furnace, the mass of free oxygen and combustible gas will become less and less compared with the total mass of gas present and their combustion will become slower until it reaches zero. With the present arrangements the reactions will take so long that the gases will have passed through the furnace before combustion is complete, see p. 481.

Losses Due to Excess Air are (1) products of combustion are diluted and carry off an excessive amount of heat in the smoke pipe, (2) the furnace temperature may be so lowered as to retard combustion or cause formation of CO (3) torching, that is burning at the smoke pipe top, may be caused due to (2).

Volume Contraction. The water vapor formed by combustion of hydrogen condenses at the temperature at which a flue gas analysis is made. As the nitrogen that accompanies the oxygen used in the hydrogen combustion retains its gaseous state, the result of this condensation is to increase the apparent percentage of nitrogen in the flue gas.

Combustion Products. If v_m is the minimum volume of air required for complete combustion and xv_m the actual volume supplied, then the products will contain per lb. of fuel $O_2 = 0.21\ v_m\ (x-1)$ cu. ft., $N_2 = 0.79\ xv_m$ cu. ft.

From the reaction equation $C + O_2 = CO_2$, the volume of CO_2 formed is equal to the volume of oxygen required for the carbon constituent alone; hence, volume of $CO_2 = 0.21 v_m c + [c + 3\ (h - 0.125o)]$.

Of the *dry* gaseous products (*i.e.*, without water) the CO_2 content by volume is therefore given by the expression,

$$CO_2 = 0.21c + [xc + (x - 0.21)3(h - 0.125o)].$$

The combined CO_2 and O_2 content is

$$CO_2 + O_2 = 0.21\{1 - 0.79 \div [x + cx + 3(h - 0.125o) - 0.21]\}.$$

If the fuel is all carbon the combined CO_2 and O_2 is by volume 21 per cent of the gaseous products. The more hydrogen contained in the fuel the smaller is the $CO_2 + O_2$ content. The CO_2 content depends in the first instance on the excess of air. Thus, for pure carbon it is $CO_2 = 0.21 \div x$.

The excess of air may be calculated from the composition of the gases and that of the fuel. Thus $x = 0.21\left[\frac{c}{CO_2} + 3(h - 0.125o)\right] + [c + 3\ (h - 0.125o)]$, in which CO_2 denotes the per cent by volume of the CO_2 in the dry gas.

OIL FUEL BURNING

BY

ALBERT M. PENN

The three fundamental factors which control the proper combustion of fuel oil are: (1) the atomization or the breaking up of the oil into minute particles; (2) the introduction of air in just sufficient quantity and in such a manner that it will surround every particle of oil and cause complete combustion; and (3) a furnace so constructed that the gases will be kept above ignition temperatures until combustion is completed.

ATOMIZATION

Prior to the development of the atomizers in general use at present, attempts were made, with but limited success, to burn the oil in shallow **open pans** and modifications thereof. This system was only practicable on shore and in very small units. Its inefficiency lies primarily in the inability to introduce sufficient air for complete combustion to the surface of the oil.

Extensive experimentation in **pre-gasifying** the oil fuel in auxiliary apparatus before its introduction into the furnace has also met with very limited success, though excellent results are obtained with the lighter petroleum distillates—naphthas, gasolines and kerosenes. In gasification, the application of heat is essential; the very high temperature to which it must be heated causes a breaking down of fuel oils with the resultant depositing of the residuum in the form of sludge and carbon, which soon clogs up the apparatus and makes it inoperative.

Present Methods of Atomization are a compromise between the two systems referred to above. The oil is injected into the furnace in the form of a spray of finely divided, fog-like particles. The atomization is accomplished in three ways. These are named after the atomizing agent used in each case: (1) Steam Atomizers; (2) Air Atomizers; (3) Pressure or Mechanical Atomizers.

For Marine Purposes, the last named is the only one worthy of serious consideration, but the engineer should be acquainted with the principles, advantages and disadvantages of all three.

Steam Atomization is the simplest and is efficient for stationary plants where the water supply is plentiful. The first cost of equipment is low. Oil at low pressure is delivered to the atomizer, where it comes in contact with a jet of dry steam which breaks it up into fine particles. The vapor-like mixture enters the furnace where it is consumed. The air for combustion is generally supplied by natural draft. Some claim that the expansion of the steam, due to its rapid superheating to the high furnace temperature, tends to further atomize the oil and thereby aids combustion.

From ½ to ¼ of a lb. of steam is required for every pound of oil fuel. In stationary practice this loss of fresh water in the form of steam may be of minor consequence and is offset by the increase in furnace efficiency, but for marine purposes the use of steam as an atomizing agent means a serious drain, whether the water used is attained by distillation or whether a dead weight of water be carried in the double bottoms.

Compressed Air Atomizers operate on the same principle as those using steam. A jet of air at high pressure breaks up the oil. Some types use air at only a few pounds pressure, but their efficient use is limited to the lighter grades of oil. The necessary air compressors mean a high first cost, and, on board ship, encroach upon the weight and space given to machinery. Upkeep and power used are also serious considerations. That the compressed air supplied for atomizing purposes is far from sufficient for complete combustion must not be overlooked.

Air atomizers occupy an extensive field of usefulness. They are well adapted to heat treating furnaces, for annealing, case hardening, tempering, tool dressing, etc.

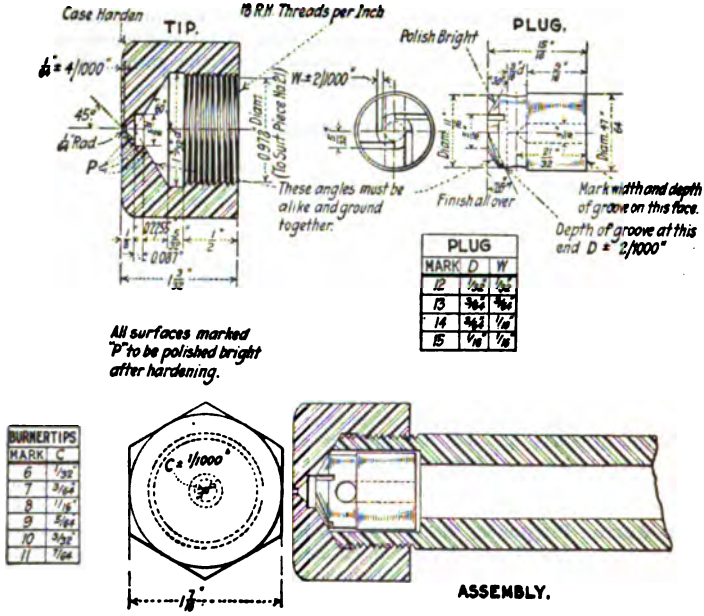


FIG. 1.—Bureau S. E. type 1 standard pressure atomizer.

The pressure or mechanical atomizer has practically been **universally adopted for marine boiler use.** There are many types of pressure atomizers on the market. All incorporate the principle of forcing oil under high pressure through passages in the atomizer so arranged as to give the oil a high velocity of rotation and thus break it up under the action of centrifugal force.

The rotation is accomplished in two general ways: in one, the oil is forced at high pressure through spiral grooves that discharge into a small cylindrical chamber. The end of this chamber opposite the entering spiral grooves is coned out. At the apex of this cone is the tip orifice through which the oil flows in a direction at right angles to the rotation given it by the spiral

grooves. In the second method, the rotation is accomplished by means of tangential grooves or holes discharging into a small cylindrical chamber. The tip end of this chamber is coned out and, as with the spiral type, at the apex of this cone is the orifice through which the oil is discharged.

As the oil leaves this orifice, in both general types, it is acted on by three forces: gravity; centrifugal action, due to rotation; and translation along the axis of the atomizer. As a result, on leaving the orifice the small streams of oil break up into very fine fog-like particles. Each particle of oil tends to fly off in a straight line at an acute angle to the axis of the orifice, and also at an acute angle to the tangent to this orifice at the point left by the particle of oil. The path of each individual particle of oil would be a straight line as above, but for the downward curve, due to the action of gravity. This gives a cone of finely atomised oil.

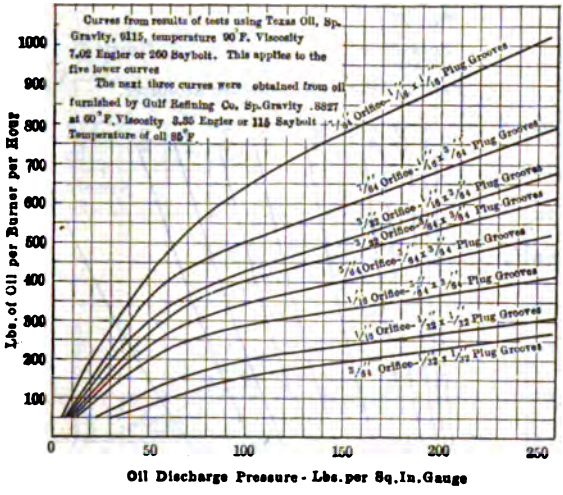

Curves from results of tests using Texas Oil, Sp. Gravity, 9115, temperature 90°F. Viscosity 7.02 Engler or 260 Saybolt. This applies to the five lower curves
The next three curves were obtained from oil furnished by Gulf Refining Co. Sp.Gravity .8827 at 60°F. Viscosity 3.35 Engler or 115 Saybolt
Temperature of oil 85°F.
Lbs. of Oil per Burner per Hour
1000
900
800
700
600
500
400
300
200
100
0
50
100
150
200
250
Oil Discharge Pressure - Lbs. per Sq. In. Gauge
1/8" Orifice - 1/8" x 1/8" Plug Grooves
1/64 Orifice - 1/16" x 3/64" Plug Grooves
5/32 Orifice - 1/16" x 3/64" Plug Grooves
3/32 Orifice - 3/64" x 3/64" Plug Grooves
5/64 Orifice - 3/64" x 3/64" Plug Grooves
1/16 Orifice - 3/64" x 3/64" Plug Grooves
1/16 Orifice - 1/32" x 1/32" Plug Grooves
3/64 Orifice - 1/32" x 1/32" Plug Grooves

FIG. 2.—Pressure-capacity curves Bureau S. E. type 1 standard pressure atomizer.

The **oil cone** should be hollow and the wall so thin and uniform that air admitted to it can readily mix with the particles of oil. Atomizers of different designs, but of the same capacity and making sprays of the same angle, can be quite different in the quality of the sprays produced, depending upon how hollow the cone is, the fineness of the particles of oil, and their degree of freedom from streaks.

There are practical considerations which make it desirable to keep the **angle of spray** at about 75 deg., such as, the size of the furnace openings, the interference of one spray with another, the formation of carbon on the boiler tubes and furnace walls. The angle of spray and the fineness of atomization are affected by: (1) the depth and shape of the orifice walls; (2) the size and shape of the whirling chamber; (3) the number of tangential slots that direct the oil to the whirling chamber; (4) the angle at which the slots are inclined to the axis of the burner; and (5) the ratio of the combined cross-sectional areas of the tangential slots to the area of the orifice.

The **proportions of the various oil passages** is determined experimentally, and **slight variations** from a set standard have a noticeable effect on results. Where a number of atomizers are used on a boiler front,

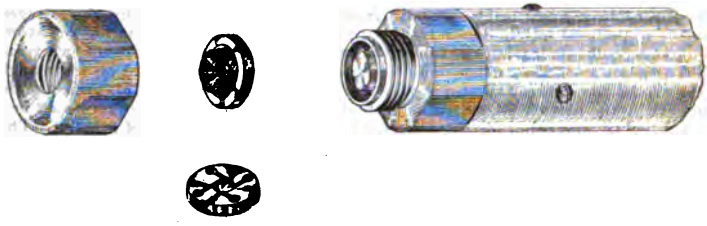


FIG. 3.—Peabody pressure atomizer.

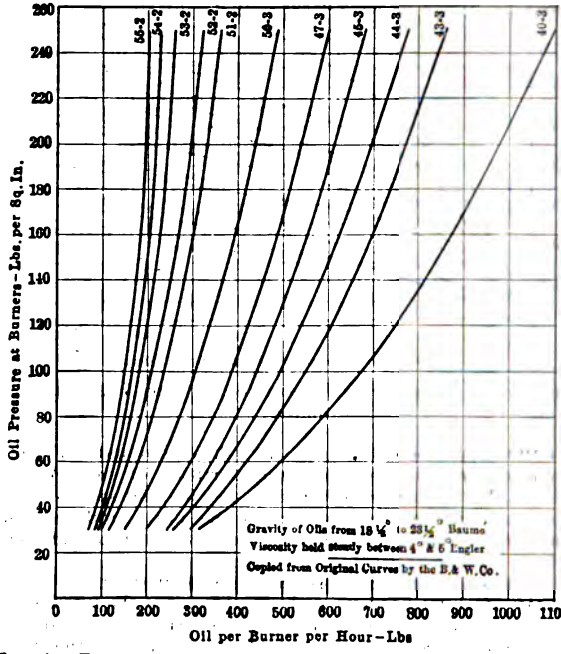
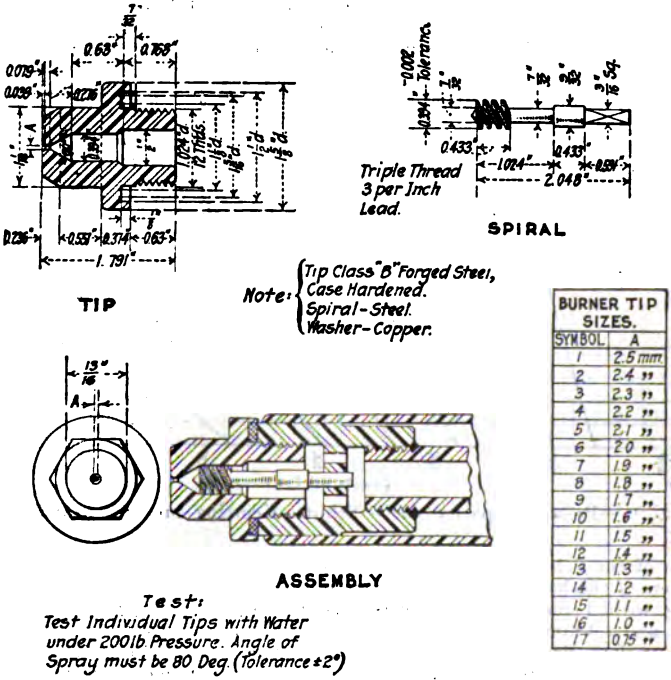


FIG. 4.—Pressure-capacity curve Peabody pressure atomizer.

it is essential, for efficient manipulation, that all produce uniform sprays. This requires in manufacture that the atomizers be as nearly uniform with each other as it is practically possible to make them.

It is not only necessary that the various oil passages be made to size but they must also be polished and free from blemishes. In use, **care must be taken not to scratch or mar the polished surfaces.** Nails or drills should never be used **to clean orifices.** Copper wire, match sticks or tooth picks will clean an orifice without marking it. In **removing carbon,** soaking with kerosene will loosen most of it and steel wool applied with a hardwood stick, properly shaped and chucked in a high speed sensitive drill press, is effective. **Abrasives** that will wear whirling chamber or orifice **must not be used. Atomizers of different capacities should never be used in a boiler at the same time.**

Before being used for the first time all atomizers should be tested for defects with water or oil at a pressure of about 200 lbs.


TIP
SPIRAL
Triple Thread 3 per Inch Lead.
Note: Tip Class "B" Forged Steel, Case Hardened. Spiral - Steel. Washer - Copper.
ASSEMBLY
Test: Test Individual Tips with Water under 200 lb Pressure. Angle of Spray must be 80 Deg. (Tolerance ±2°)

BURNER TIP SIZES.	
SYMBOL	A
1	2.5 mm
2	2.4 "
3	2.3 "
4	2.2 "
5	2.1 "
6	2.0 "
7	1.9 "
8	1.8 "
9	1.7 "
10	1.6 "
11	1.5 "
12	1.4 "
13	1.3 "
14	1.2 "
15	1.1 "
16	1.0 "
17	0.75 "

FIG. 5.—Schutte-Koerting pressure atomizer.

Types of Pressure Atomizers. Figs. 1, 3 and 5 are sectional views of pressure atomizers used extensively in the Naval Service.

Fig. 1 is the **Bureau S. E. Type 1 Standard Atomizer.** It is composed of two parts, **the tip** and **the plug.** The **tip** contains the orifice and whirling chamber, and **the plug** the tangential **grooves** which impart the whirling motion to the oil. The plug does not revolve but is held in place against the tip by the pressure of the oil behind it. The proper combination of, tips and plugs and the curves of **capacity** in each case are illustrated in **Fig. 2.**

Fig. 3 is a sectional view of the **Peabody Atomizer.** In this design the orifice, whirling chamber, and the oil channels are all contained in one piece known as the **sprayer plate.** Suitably drilled holes direct the oil to the channels tangent to the whirling chamber.

In designating Peabody Atomizers the first number indicates the size of the orifice by drill numbers, and the figure following the dash is the ratio between the combined areas of the slots and the area of the orifice. The most common sizes of this tip and their **capacity** curves are shown in Fig. 4.

Fig. 5 is a sectional view of the **Schutte-Koerting Atomizer.** This is one of the very few atomizers that uses a screw thread to cause rotation of the oil just prior to its exit through the orifice. This **screw** has a triple thread which fits in the **tip** with a very little clearance, and is kept from rotating by a square shank which fits in a socket in the base of the atomizer. The

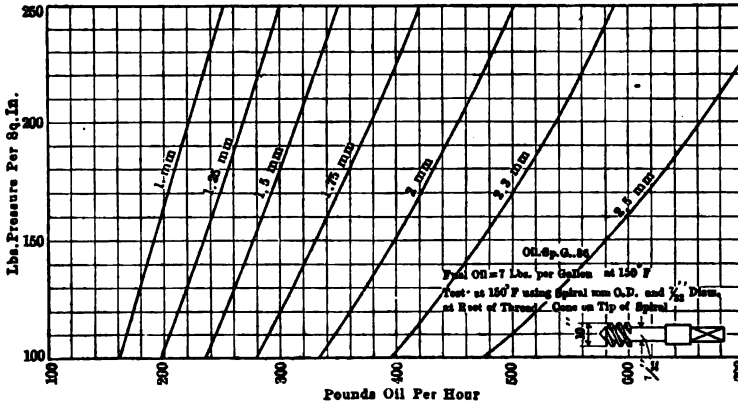


FIG. 6.—Pressure-capacity curve Schutte-Koerting pressure atomizer.

sizes of the Schutte-Koerting tips are designated by the diameter in millimeters of the hole in the tip. They are made in a number of sizes beginning with 1 mm. On destroyers a 1-mm. tip is the size usually used in port and 2.3 mm. at sea. Battleships generally use the 1.5 to 1.7 mm. sizes. The various sizes and their rated **capacity** are shown in Fig. 6.

Effect of Heating Oil. With the pressure or mechanical systems, **fuel oil is heated for the sole purpose of reducing the viscosity,** or, in other words, increasing the fluidity. This is essential, primarily as an aid to fineness of atomization, but, occasionally, with extremely viscous oils at low temperatures, some heating is necessary to reduce the viscosity so that the pumps may handle the oil at full capacity. This viscosity necessarily depends on the type of pumps used.

Of primary importance to the operating engineer is the **requisite viscosity to obtain the best atomization.** The extent to which an oil may be atomized is not alone dependent upon the design of the atomizer but upon the viscosity of the oil as well. The thinner the oil the finer will be the atomization produced. The greater the viscosity, the more difficult it

becomes for the atomizer to break up the oil into particles fine enough for its intimate mixture with the necessary air for combustion. Where viscous oils are used ,this is a difficulty which the pressure atomizer of itself is unable to overcome.

The curves in Figs. 7 and 7*a* show graphically the effect of heating on the viscosity of fuel oils. It will be observed that, for those oils having a gravity Baumé of 20 (sp. gr. 0.9340) and above, very little reduction in viscosity is gained by heating above 130 deg. F. Inasmuch as pressure atomizers will produce efficient atomization with oils at a viscosity between 2 deg. and 4 deg. Engler (75 and 150 Saybolt sec.) there is no practical gain in heating the oil above the temperature that will give a viscosity between these limits.

In the case of the **more viscous** oils, it is neither desirable nor practicable to reduce the viscosity to so low a point, as to do so would require heating the oil to a temperature high above the flash point; a dangerous expedient unless extreme care is taken to guard against leaks in the oil lines. With some very viscous oils, having a low flash point, it will be necessary to ignore this precaution in order that the viscosity may be reduced to that required for moderately good atomization. With these oils, that temperature which will give a viscosity of about 8 deg. Engler (300 Saybolt sec.), or slightly lower, is sufficient to give satisfactory combustion in a boiler furnace. A lower viscosity would afford finer atomization, and consequently more rapid and efficient combustion, which, incidentally, would permit of a greater rate of combustion per cu. ft. of furnace volume per hour. In addition to the objections cited to the heating of a very viscous oil to excessive temperatures, may be added the large extra weight and space required for oil heaters of sufficient capacity, and also the danger of breaking down the oil, causing it to clog up the heaters and oil lines and to carbonize the atomizing tip.

AIR FOR COMBUSTION

Amount of air theoretically required for complete combustion of fuel oil varies with its chemical composition. Under "Boiler Tests," page 480 are the necessary formulæ for calculating exactly the amount required.

For average fuel oils, the amount of air theoretically required varies between **13.8 and 14.4 lbs. per lb. of fuel, or,** expressed in cu. ft. from 180 to 200 per lb. of fuel.

In practice, the efficiency of combustion may be considered very good if **the excess air is kept** below 30 per cent, and excellent if below 10 per cent. On boiler evaporative tests conducted under the most favorable conditions the excess air has been reduced to as low as 4 and 5 per cent, but this is exceptional. Percentages of excess air as high as 200 per cent are of common occurrence. It is the most **serious and most common cause of low furnace efficiency.**

The aid of a gas **analysis apparatus,** described on p. 483, is essential to reduce excess air to a minimum. A study of the appearance of the furnace is also of valuable assistance. If the flame is an incandescent white and the furnace walls are clearly discernible through it, there is considerable excess air present.

As the percentage of excess air is reduced, the **color of the flame** at the rear of the furnace becomes a pale yellow, then yellowish orange and orange red. In general, with an efficient installation operating under a minimum of excess air, the ends of the flame farthest from the atomizers are a yellowish orange or golden shade. The gases of combustion are colorless and the seams in the brick are just discernible.

At very high rates of combustion, when the flame completely fills the furnace, the very high furnace temperatures preclude the presence of these temperature colors. An incandescent and dazzling white still indicates excess air but a reduction in this excess is only indicated in the furnace by a softening in the intensity of the white flame.

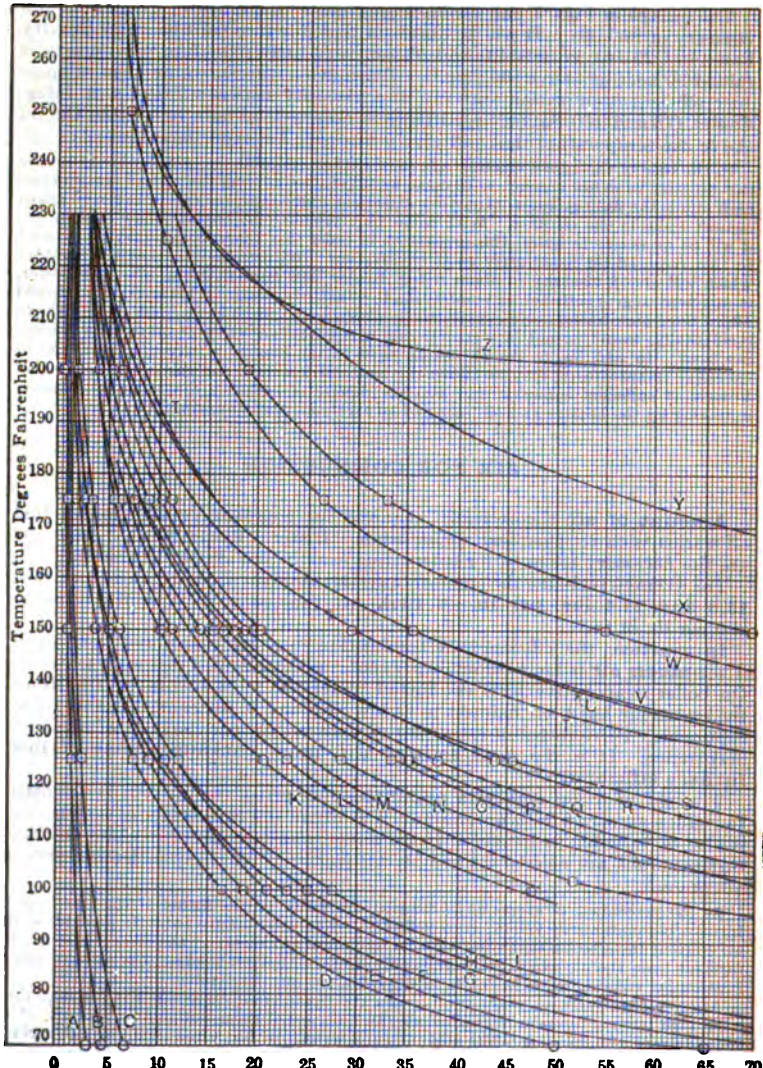
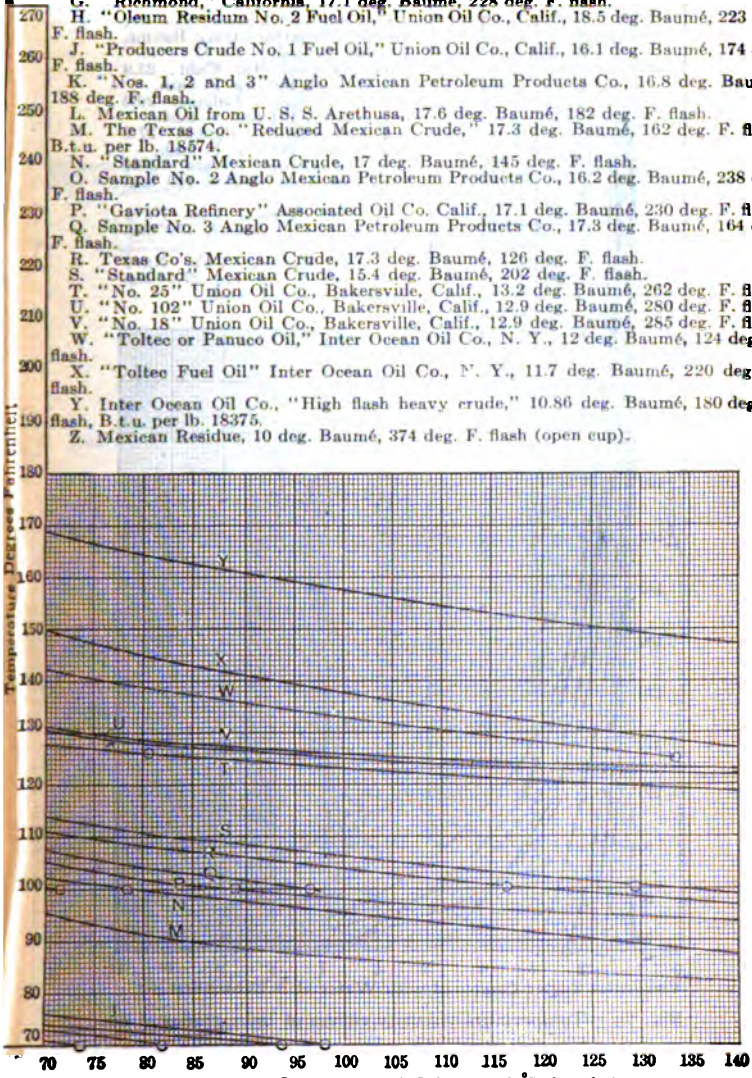


FIG. 7.—Temperature viscosity

A. Star California, 23.9 deg. Baumé, 180 deg. F. flash.
B. "Standard" Indiana, 29.6 deg. Baumé, 144 deg. F. flash.
C. "Standard" Illinois, 27.2 deg. Baumé, 146 deg. F. flash.
D. Sun Company's Louisiana, 19.8 Baumé, 275 deg. F. flash.
E. "Avon Refinery" Associated Oil Co., Calif., 17.1 deg. Baumé, 168 deg. F. flash.
F. "Coalinga Field" Associated Oil Co., Monterey, Calif., 16.5 deg. Baumé, 186 deg. F. flash.
G. "Richmond," California, 17.1 deg. Baumé, 228 deg. F. flash.
H. "Oleum Residum No. 2 Fuel Oil," Union Oil Co., Calif., 18.5 deg. Baumé, 223 deg. F. flash.
J. "Producers Crude No. 1 Fuel Oil," Union Oil Co., Calif., 16.1 deg. Baumé, 174 deg. F. flash.
K. "Nos. 1, 2 and 3" Anglo Mexican Petroleum Products Co., 16.8 deg. Baumé, 188 deg. F. flash.
L. Mexican Oil from U. S. S. Arethusa, 17.6 deg. Baumé, 182 deg. F. flash.
M. The Texas Co. "Reduced Mexican Crude," 17.3 deg. Baumé, 162 deg. F. flash, B.t.u. per lb. 18574.
N. "Standard" Mexican Crude, 17 deg. Baumé, 145 deg. F. flash.
O. Sample No. 2 Anglo Mexican Petroleum Products Co., 16.2 deg. Baumé, 238 deg. F. flash.
P. "Gaviota Refinery" Associated Oil Co. Calif., 17.1 deg. Baumé, 230 deg. F. flash.
Q. Sample No. 3 Anglo Mexican Petroleum Products Co., 17.3 deg. Baumé, 164 deg. F. flash.
R. Texas Co's. Mexican Crude, 17.3 deg. Baumé, 126 deg. F. flash.
S. "Standard" Mexican Crude, 15.4 deg. Baumé, 202 deg. F. flash.
T. "No. 25" Union Oil Co., Bakersvule, Calif., 13.2 deg. Baumé, 262 deg. F. flash.
U. "No. 102" Union Oil Co., Bakersville, Calif., 12.9 deg. Baumé, 280 deg. F. flash.
V. "No. 18" Union Oil Co., Bakersville, Calif., 12.9 deg. Baumé, 285 deg. F. flash.
W. "Toltec or Panuco Oil," Inter Ocean Oil Co., N. Y., 12 deg. Baumé, 124 deg. F. flash.
X. "Toltec Fuel Oil" Inter Ocean Oil Co., N. Y., 11.7 deg. Baumé, 220 deg. F. flash.
Y. Inter Ocean Oil Co., "High flash heavy crude," 10.86 deg. Baumé, 180 deg. F. flash, B.t.u. per lb. 18375.
Z. Mexican Residue, 10 deg. Baumé, 374 deg. F. flash (open cup).



rves of heavy grades of oils.

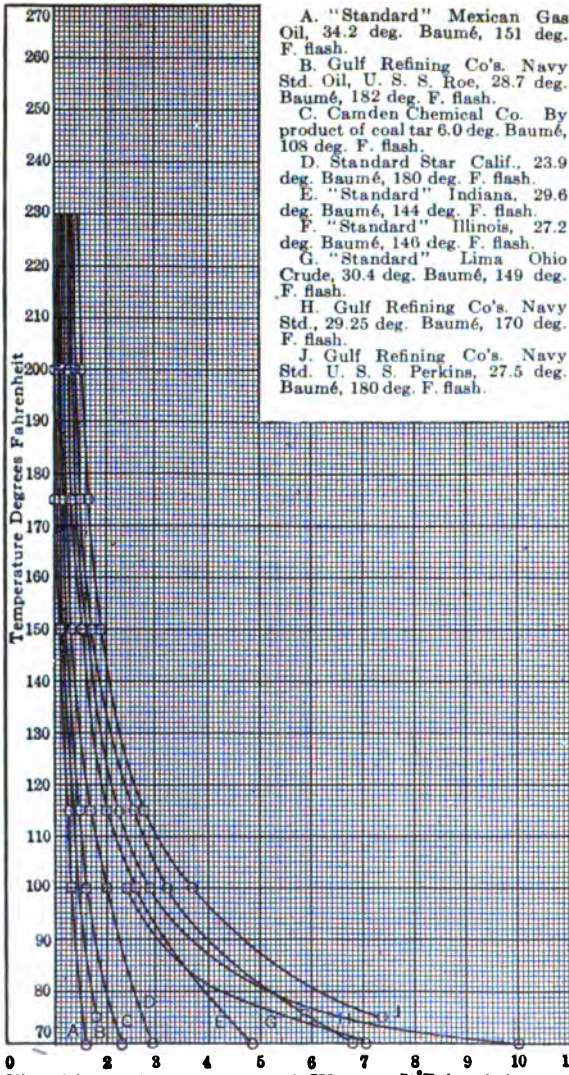


FIG. 7a.—Temperature viscosity curves of light grades of oils.

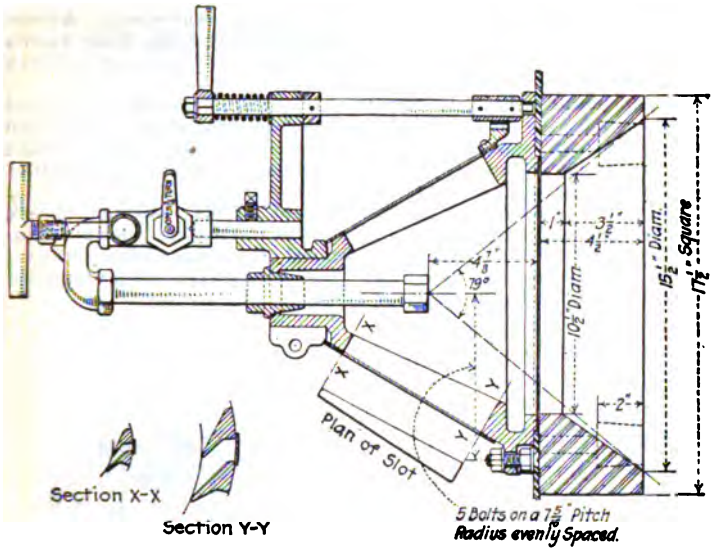


FIG. 8.—Bureau S. E. type 2 standard forced draft air control register.

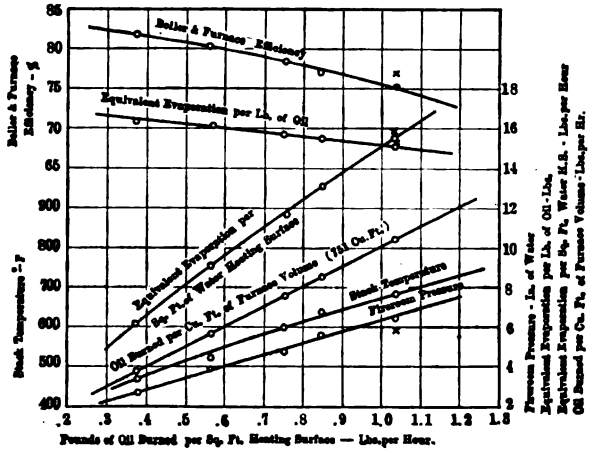

Boiler & Furnace Efficiency
Equivalent Evaporation per Lb. of Oil
Equivalent Evaporation per Sq. Ft. of Water Heating Surface
Oil Burned per Cu. Ft. of Furnace Volume (Tail Cu. Ft.)
Stack Temperature
Firebox Pressure
Pounds of Oil Burned per Sq. Ft. Heating Surface — Lbs. per Hour.

FIG. 9.

A perfectly **clear smoke pipe** is deceiving; it may mean only a small amount of excess air, or it may mean as much as 300 per cent excess. **A clear smoke pipe is ideal when the gas analysis at the same time shows high CO_2, very low O_2 and no CO; for example,** 14.7 per cent CO_2, 0.8 per cent O_2, 0.0 per cent CO.

Under the **usual operating conditions** in marine service, it is good practice to require the fire room personnel to reduce the supply of air until a **very light brown haze** is emitted from the smoke pipe. This assumes that dirty or streaky atomizers have been eliminated, and that the installation when properly operated is efficient.

It is well to remember that the **presence of smoke** does not always mean insufficient air or high CO. An inefficient installation, poor atomization, poor mixture of air and oil, unconsumed oil striking cooling surfaces, and poorly designed furnace, frequently cause smoke when the air supplied for combustion may be far in excess of that required.

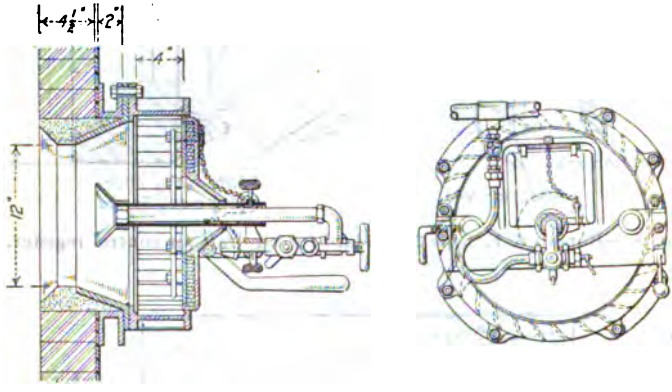


FIG. 10.—Bureau S. E. type 1 standard natural-forced draft air control register.

Leaks in boiler casings are a serious source of excess air. This air does not become intimately mixed with the fuel and every effort should be made to keep boiler casings tight. Air leaking through casings not only does not aid in further combustion of unburned gases, but also has a chilling effect on both the gases of combustion and the heating surfaces. Loss of boiler efficiency due to this cause may amount to 10 per cent or more and increases as the draft increases.

Air Control Registers. With the pressure or mechanical system, the most important part of the installation is the air register or tuyère. **The purpose** of the air register is to so introduce the air supplied for combustion that it penetrates the oil spray and becomes thoroughly mixed therewith in the proper proportion. This must be accomplished with a minimum of excess air.

In practically all marine installations the air for combustion is supplied to the air controlling registers by one of the following four methods: (1) **The**

Closed Fire Room, in which the air is forced into an air-tight fire room by forced draft fan blowers; (2) the **Howden System,** in which the air for combustion is forced by blowers through air ducts to an air-tight box casing on the

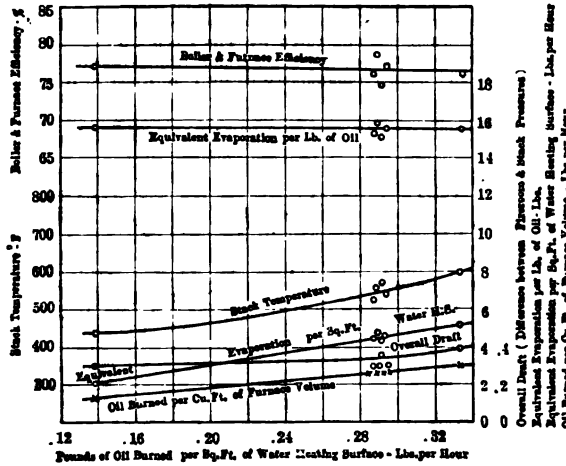

Boiler & Furnace Efficiency
Equivalent Evaporation per Lb. of Oil
Stack Temperature
Water H.S.
Evaporation per Sq.Ft.
Overall Draft
Equivalent
Oil Burned per Cu.Ft. of Furnace Volume
Pounds of Oil Burned per Sq.Ft. of Water Heating Surface - Lbs.per Hour

FIG. 11.

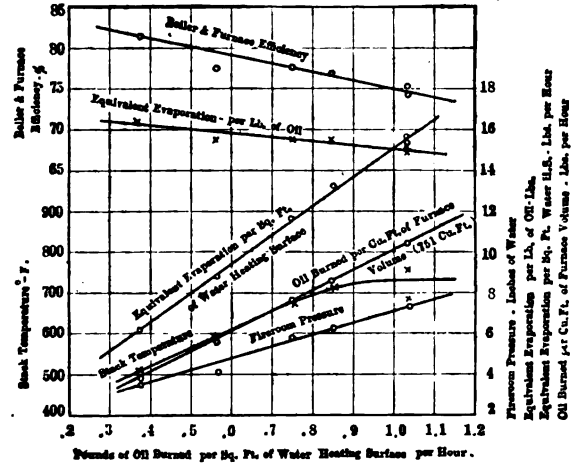

Boiler & Furnace Efficiency
Equivalent Evaporation - per Lb. of Oil
Equivalent Evaporation per Sq. Ft. of Water Heating Surface
Oil Burned per Cu.Ft. of Furnace Volume (75) Cu.Ft.
Stack Temperature
Firerom Pressure
Pounds of Oil Burned per Sq. Ft. of Water Heating Surface per Hour.

FIG. 12.

furnace front; (3) **Natural Draft,** in which the height of smoke pipe is relied upon to induce a sufficient flow of air into the furnace; and (4) **Induced Draft,** in which blowers are installed in the uptake or smoke pipe to produce a

suction from the furnace and to drive the gases of combustion out of the smoke pipe.

The following is a partial list of air registers that will be met with in Marine Service:

Bailey Register and Burner—manufactured by Newport News S. B. & D. D. Co
Coen Register and Burner—manufactured by The Coen Co.
Dahl Register and Burner—manufactured by Union Iron Works.
Fore River Register and Burner—manufactured by Fore River S. B. Corp.

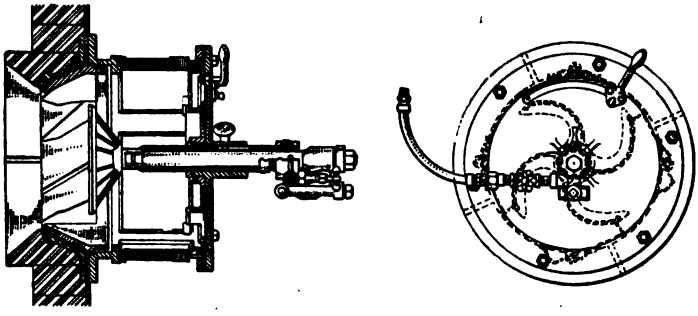


FIG. 13.—Peabody air control register.

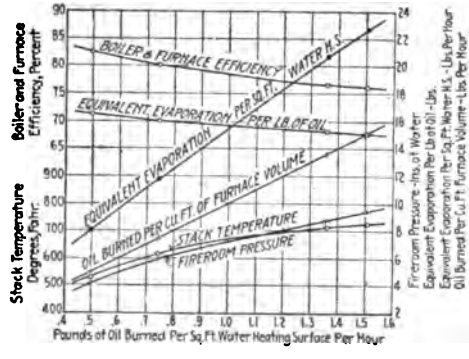

Boiler and Furnace Efficiency, Percent
Stack Temperature Degrees, Fahr.
BOILER & FURNACE EFFICIENCY
EQUIVALENT EVAPORATION PER SQ. FT. WATER H.S.
EQUIVALENT EVAPORATION PER LB. OF OIL
EQUIVALENT EVAPORATION PER CU. FT. OF FURNACE VOLUME
OIL BURNED PER CU. FT. OF FURNACE VOLUME
STACK TEMPERATURE
FIREROOM PRESSURE
Pounds of Oil Burned Per Sq. Ft. Water Heating Surface Per Hour
Fireroom Pressure - Ins. of Water
Equivalent Evaporation Per Lb. of Oil - Lbs.
Equivalent Evaporation Per Sq. Ft. Water H.S. - Lbs. Per Hour
Oil Burned Per Cu. Ft. Furnace Volume - Lbs. Per Hour

FIG. 14.

Moore and Scott Register and Burner—manufactured by Moore & Scott Iron Works.
Normand Register and Burner—manufactured by Bath Iron Works.
Peabody Register and Burner—manufactured by Babcock & Wilcox Co.
Schutte-Koerting Register and Burner—manufactured by Schutte & Koerting Co.
The White System—manufactured by The Washington Engine Works.
The Tate-Jones System—manufactured by The Tate-Jones Co.
Bureau of Steam Engineering's Standard Installations.

Types of Air Registers. The five air-controlling registers illustrated hereafter bring out the general characteristics of all registers used with pressure atomisers on shipboard.

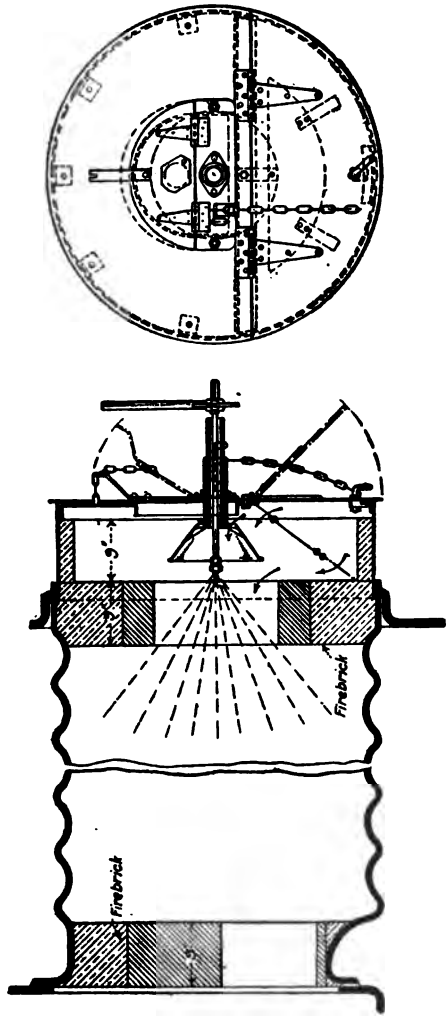


FIG. 15.—Dahl atomiser and air control for Scotch boiler furnaces.

Fig. 8 shows the **Bureau of S. E. Type 2 Standard Forced Draft Register,** with the Bureau Standard Atomiser properly located. After a series of tests conducted on the Experimental White-Forster boiler at the

Naval Fuel Oil Testing Plant, Philadelphia, this **register** and atomiser were adopted by the Bureau of Steam Engineering as the standard high-power destroyer installation. Fig. 9 is a set of curves showing the results of these tests.

Fig. 10 shows the **Bureau of S. E. Type 1 Standard Natural-Forced Draft Register**, as developed at the *Naval Fuel Oil Testing Plant*. It was especially designed for efficient operation under natural draft and at low powers, but it can readily be forced to high rates of combustion. This register has been adopted by the Bureau of S. E. for battleships and auxiliary ships of moderate power. It has been installed on a few destroyers where it has given very satisfactory results at high powers. Fig. 11 is a set of curves showing the results of **evaporative tests under natural draft** using this register on a Babcock & Wilcox oil-burning boiler of the type installed on the Battleships *Oklahoma* and *Pennsylvania*. Fig. 12 is a set of curves showing the results of **tests under high power at forced draft** with this register installed on the experimental White-Forster boiler at the *Fuel Oil Testing Plant*.

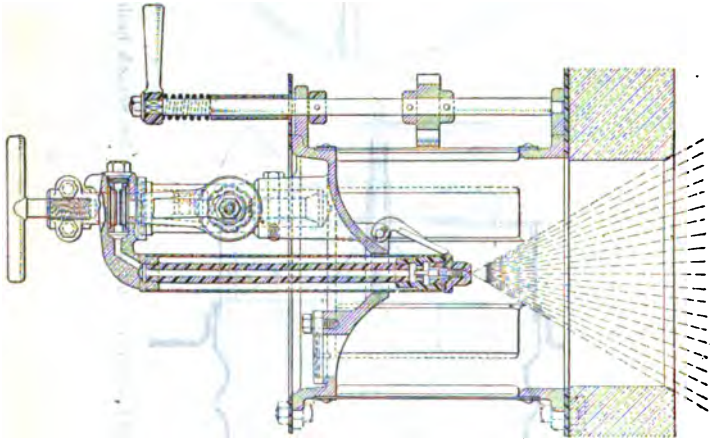


FIG. 16.—Schutte-Koerting air control register fitted with Schutte-Koerting atomizer.

Fig. 13 shows the latest type **Peabody Register** fitted with the Peabody Atomizer as manufactured by the *Babcock & Wilcox Company*. It is installed on the Battleships *New Merico* and *Mississippi* and on numerous vessels of other types, both in the Navy and in the Merchant Marine. This register was designed to operate under natural draft and at moderate powers, but may be readily and efficiently forced to high rates of combustion. Fig. 14 is a set of curves showing the results of tests conducted on the experimental White-Forster boiler installed at the *Fuel Oil Testing Plant* with this installation.

Fig 15 illustrates the **Dahl Air Control Register**, manufactured by the *Union Iron Works*, installed on the furnace of a Scotch boiler. This system is a low power, low draft type extensively used in the merchant marine on the Pacific Coast.

The **Schutte-Koerting Air Register and Burner** as manufactured by the Schutte & Koerting Company is illustrated in Fig. 16. This installation is essentially a high-forced draft system. It was extensively used in the Naval Service on destroyers and other vessels some six years ago.

BOILER FURNACE DESIGN

The **shape, size and materials of construction** of the marine boiler furnace have a very decided influence on both the **furnace efficiency** and the **rate of combustion** when oil is used as the fuel. The ideal furnace would have the following characteristics: (1) Sufficient furnace volume for total oil burned at the designed rate of combustion; (2) the maximum of refractory surface to assist in maintaining a high furnace temperature; (3) increase of cross-sectional area of the furnace with the expansion of the gases; (4) minimum area of cooling surfaces close to the flame, tending to reduce the furnace temperature; (5) walls and tubes just sufficiently remote from burners to permit of complete combustion of oil before coming in contact with them; (6) a minimum of infiltration of air except through the air controlling registers provided for the purpose; (7) exit from furnace of incandescent gases of completed combustion at a maximum distance from the burner.

The design of the marine boiler, and the various structural limitations, such as economy in weight and space, preclude complying fully with all of the above requirements.

The **different types of oil-burning furnaces** may be classified for marine purposes as follows: (1) Flue furnaces; (2) Tube-roofed furnaces; and (3) Triangular section, inclined tube-sided furnaces.

In the **Scotch Boiler** the length of the furnace flue and the combustion chamber provided are favorable features of this type, but perhaps the **main advantage** that can be claimed is the elimination of flame interference between adjacent burners by the use of only one burner per furnace. The close proximity of cooling surfaces and the difficulty experienced with circular refractory linings are the **most serious objections.**

The Babcock & Wilcox marine boiler is an excellent example of the **large tube-roofed furnace.** The general shape of the furnace, the increasing volume for the expanding gases, the large incandescent refractory surface which tends to maintain a high furnace temperature, and the location of the exit to the gas pass through the tubes; all tend to make this furnace ideal for the purpose.

The furnaces of all boilers having furnaces of class 3 type are very similar in shape and size and are such as exist in all boilers of the well known **"Express Type."** The furnace exit to the gas passes of the Normand boiler is different from the others of this type in that the gases of combustion can enter the tube banks at one end. Insofar as this arrangement affects furnace efficiency, it is an excellent design where entry occurs at the rear end.

FURNACE INSULATION

Radiation of heat from the furnace walls is taken care of in two ways in the marine installation; either by circulating air over the radiating surface, and in this way carrying off the heat, or by backing the refractory lining up with a good insulating material and thereby keeping the heat in the furnace. The method of **circulating air** is used in destroyer installations where weight and space do not permit the use of thick refractory walls backed by several inches of insulation. A light metal outer casing open at the bottom and top forms an air space. Air flowing up through this space carries off the radiant

heat into the fire room where it finds its way into the furnace through the air registers or is dissipated through the ship's side to the water.

Such an arrangement is not permissible on large ships with double bottoms as the heat banks up in the fire rooms and makes them untenable for the personnel; to avoid this the heat is kept in the furnace by the use of insulation.

In low-powered installations, common to auxiliary ships of the Navy and the Merchant Marine, the question of **insulation** involves no serious difficulties. Nine inch brick walls backed by about 4 in. of an improved insulation is satisfactory. But in the case of high-powered light-weight installations, such as battle cruisers and scout cruisers the problem becomes more serious. A 4 or 4½-in. brick wall is the maximum limit for refractory

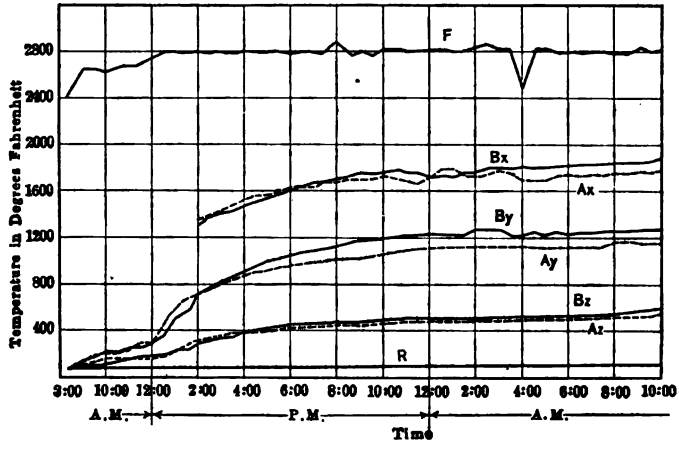


FIG. 17.

lining. This must be backed by a superior high-heat insulation. The insulation problem is constantly under investigation and the set of curves in Fig. 17 provide some interesting data on this subject.

The furnace used for testing insulation was designed to simulate, as closely as possible, service conditions in the furnaces of high-powered, marine oil-burning boilers. A 4½-in. refractory lining is backed by 3 in. of insulation in two layers of 1½-in. thickness. This is in turn backed by the usual metal furnace casing.

The test is essentially a comparative service test between two grades of insulation. Curve F is the furnace temperature. Curves A_z and B_z represent the temperatures through the 4½-in. refractory brick wall on each side of the furnace, taken between the brick and insulation A on one side and between the brick and insulation B on the other side. They are the temperatures to which the inner surfaces of the two insulations were exposed.

Curves A_y and B_y are the temperatures measured through the inner 1½-in. layers of the two insulations.

Curves A_z and B_z are the temperatures measured through the 3-in. thickness of the two insulations.

Curve R is the room temperature measured 6 inches away from the metal side casings.

A study of these curves combined with other characteristics such as shrinkage, loss in weight, etc., determines the superiority of one or the other.

The more **common insulating materials** are not satisfactory as a furnace refractory backing, especially in the high powered light weight installations.

Two to four inches of **block magnesia** protected by ¼ in. asbestos millboard was common practice until recent years. Though both are excellent insulations at usual temperatures, they are unable to withstand the extreme furnace conditions peculiar to oil burning marine high powered boilers. The **asbestos board** soon **vitrifies** and loses its insulating value at the high temperatures to which it is subjected. The block magnesia, in time, completely **disintegrates**, becoming a fine powder and losing all its insulating characteristics.

The most satisfactory insulating materials on the market for furnace linings are made from **diatomaceous earth** (otherwise known as "infusorial," "Kielsilguhr," "fossil meal," and "celite"), a mineral of minute cellular structure and of almost pure silicious composition. In the raw state as mined, it is an excellent insulation at moderately high temperatures, but is subject to excessive shrinkage at temperatures above 1600 deg. F. (871 deg. C.); this soon destroys the insulating qualities.

By **calcining**, the shrinkage is reduced and the material is made available for use as a high temperature insulation, and though this process increases the weight and tends to reduce the insulating value, nevertheless, it reduces the tendency to spall and disintegrate at higher temperatures. The use of carbonates, such as calcium carbonate, as a bonding material is common practice. This is objectionable as the carbonates cause spalling. Their presence can be detected by testing with hydrochloric acid.

The raw diatomaceous earth may be procured as mined in powdered or brick form, also combined with bonding material in plastic or block form. Calcined, it may be obtained pure as a powder and in lump or, combined with bonds, in plastic and block form.

Calcined to about 2100 deg. F. (1149 deg. C.) then bonded together with silicates and long fibre asbestos into a compact block, diatomaceous earth is probably the best material for use as an insulation backing to the refractory lining of high powered oil burning boiler furnaces.

In its present state of development, the calcined diatomaceous earth in block form is heavier than desirable but there is every assurance that this objection will soon be overcome and a much improved high heat insulation developed.

For **insulating materials** on the market, made from diatomaceous earth, both raw as mined and calcined, see p. 439.

RELATIONS BETWEEN RATE OF COMBUSTION, HEATING SURFACE, FURNACE VOLUME AND AIR PRESSURE

The **rate of combustion** of an oil burning boiler is expressed in the amount of **oil burned per each sq. ft. of heating surface per hour** when the capacity of the boiler as a generating unit is considered. Closely related to this and bearing directly on furnace efficiency is the rate of combustion as expressed in **lbs. of oil burned per cu. ft. of Furnace Volume per hr.**

Experience has demonstrated that for design purposes, when the **small tube express type** marine boiler is to be used as in destroyers, scout and battle cruisers, 1 lb. of oil may be burned for each sq. ft. of heating surface provided though this is frequently exceeded. On a recent test conducted at the Naval Fuel Oil Testing Plant, 1.5 lbs. of oil per sq. ft. of H. S. was exceeded with a resultant evaporation of slightly better than 24 lbs. of water per sq. ft. of H. S.

For the **large tube boiler** extensively used on **battleships**, auxiliaries and merchant ships, this maximum ratio is 0.75 lb. per sq. ft. In the case of merchant ships this is generally reduced to about 0.6.

In Naval practice experience has shown that the size of the furnace should be sufficient to allow at least 1 cu. ft. of F.V. for every 10.8 lbs. of oil burned per hour. In merchant practice this would generally be increased to 1 cu. ft. for every 5 lbs. of oil.

With **Scotch Boilers**, since the rate of combustion is generally relatively low, 0.5 lb. per sq. ft. of heating surface may be considered a maximum. The furnace volume may be given little consideration as it is generally much in excess of that required for the usual rate of combustion.

The figures given above for F.V. are frequently exceeded, there being records of better than 13 lbs. of oil per hour per cu. ft. **The rate of combus-**

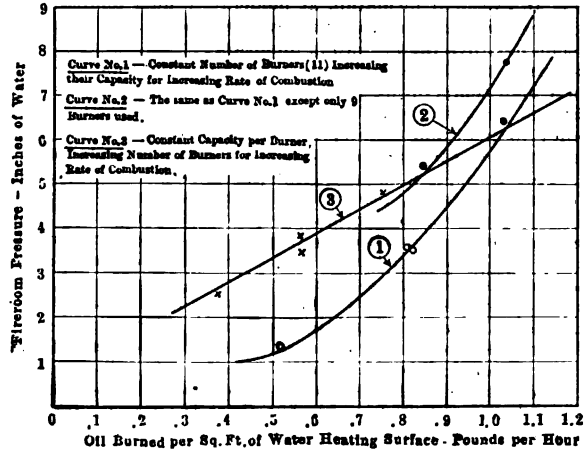


FIG. 18.

tion per cu. ft. of F.V. for any given boiler installation is influenced by the heat value of the oil, the air pressure, the design of air controlling register and the capacity, number and efficiency of the atomizers or so-called burners.

The **air pressure available** has quite a **decided influence** on the rate of combustion as will be observed by reference to the fire-room air-pressure curves illustrated in Figs. 9, 12 and 14. In each series of tests the curve is practically a straight line. Comparing these curves to those showing the oil burned per cu. ft. of F.V., which are also straight lines, it will be observed that the rate of combustion in pounds increases more rapidly than the corresponding increase of air pressure in inches. From this it is possible to predict the **air pressure required for increased rate of combustion** obtained by increasing the number of burners, the capacity of the individual burner remaining the same: The above does not apply where the number of burners remains the same and their capacity individually is increased.

Fig. 18, curves 1 and 2, give the approximate slope to the curves of the fire-room air pressure plotted against rate of combustion in the case where the number of atomizers remains the same and the rate of combustion is increased by **increasing individual burner capacity.**

Curve 3, Fig. 18, shows the slope to the curve of increasing fire-room air pressure as the rate of combustion is increased by **increasing the number of burners** in operation, the capacity of the individual burners remaining constant.

It will be observed that the increase in air pressure as the rate of combustion is increased is more rapid in the case where the number of burners remains the same and their capacity is increased. If other conditions, such as extent of furnace front, permit, higher efficiency and lower air pressures will be attained when the rate of combustion is increased by increasing the number of atomizers in operation instead of increasing the capacity of the individual atomizers.

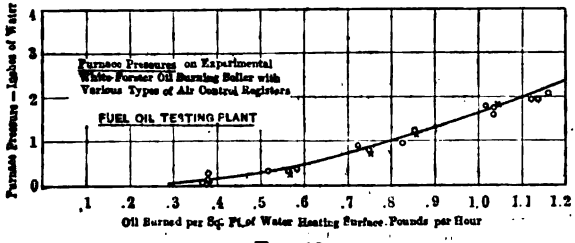


FIG. 19.

Fig. 19 illustrates a curve of **furnace pressures plotted against rates of combustion,** for the White-Forster Boiler. This curve is applicable to this boiler regardless of the type of oil-burning installation, provided the furnace efficiency in each case is the same. A curve of this type may be plotted for any given boiler by measuring the furnace pressure at different rates of combustion when the gas analysis is practically the same in each case. Once this curve is obtained, the operating personnel can be required to regulate the fire-room air pressure to maintain the furnace pressure required for any particular rate of combustion. The slight variations in the furnace pressure due to atmospheric conditions are negligible.

This method of air control is a refinement that may never be attained on ship board, but its discussion lends an understanding to the study of efficient furnace combustion.

FUEL OIL EQUIPMENT

On Naval vessels the **main oil storage tanks** are usually placed both forward of the fire rooms and aft of the engine rooms. The distribution of weights is generally such that the greatest number of tanks will be found forward. In some types of ships additional oil tanks are located in the double bottoms under engine-room compartments.

From the bottom of each forward storage tank a pipe is led to the **suction and filling manifold** in the forward fire-room. A similar manifold is pro-

vided in the engine room and connected to pipes from the after group of storage tanks. The ends of all suction pipes are located about 4 in. from bottom of tanks and are fitted with heating **coils**. Drains from these coils lead to an inspection tank.

From each of the above-mentioned manifolds a combined suction and discharge pipe runs to the combined suction and discharge manifold at each **booster pump**. This pump, of which there is one or more in each fire room, takes it suction from this manifold through a **duplex strainer** and discharges through an additional strainer into the oil service pump suction or, as in case of some of the larger ships, discharges into a main forming a continuous loop common to all fire rooms and from which all **oil service pumps** take their suction.

These service pumps, of which there are at least two in each fire room, supply the oil through another strainer to the **oil fuel heater**, thence by way of the burner line to the **burners**. In most cases all the oil service pumps can also take suction direct, without the use of the booster pumps. Both fuel oil heaters and strainers can be by-passed.

For raising steam with no source of power available, a **hand pump** is provided in each fire room. This pump is of suitable size to give a pressure of at least 200 lb. on two or more burners. Frequently the discharge from the hand pump passes through a **charcoal heater**.

Air banks are placed on the discharge manifold between the two oil service pumps, and either between the cut-out valve and the first burner or on the extreme end of the burner line. Small connections are installed for charging air banks with compressed air.

The **cut-out valves** to the burner lines across boiler fronts are fitted with extension stems for emergency operation from the deck above. These valves are usually of the plug valve type.

Suitable **deck connections** and **filling lines** are installed for filling all oil tanks through the combined fuel oil suction and discharge piping. In addition, the centre tanks may be filled through the top manhole plates and the oil sluiced to the side tanks. Oil may be equalized in the different storage tanks through the combined suction and discharge manifolds in forward engine room and in after fire rooms.

The Booster Pumps frequently have a suction connection from the filling pipes and can discharge through manifolds to any fuel oil storage tank. They are fitted with an overboard discharge connection for the purpose of cleaning tanks and for pumping water and sediment from tanks. A section in this line is generally made removable to avoid any possibility of accidentally pumping oil overboard.

In general, the installation on oil-burning merchant ships conforms to Naval Practice. On these vessels the suction and filling piping is also used for oil ballast and trimming. Instead of air banks being placed on the burner line, a large air tank is frequently installed between the service-pump discharge strainers and the oil fuel heater. The main oil manifolds have steam connections for steaming out oil lines and tanks. A return connection to the oil service pump suction is provided for the purpose of circulating the oil through the burner supply line when first lighting up.

Pumps. The most common type met with in service is the **reciprocating piston pump**, both single and duplex, but there is now a decided tendency toward the use of the **rotating plunger and the screw types**. The main advantage of the latter two types is that they deliver the oil to the atomizers at a steady pressure, while the reciprocating pump requires the fitting of

air banks to cushion and absorb the pulsations imparted to the oil on every stroke.

Piston Type Oil Pumps. The following comments on the piston type oil pump were written by *Mr. W. A. Ebsen*, of the *International Steam Pump Co.*, and published in "*Oil Fuel*" by *Mr. E. H. Peabody:*

"The pumps designed, and usually preferred, for handling crude or fuel oils are of the duplex **piston pattern**, except for large capacities accompanied by heavy pressure, where an **outside-packed plunger pump** is to be recommended.

"With the piston pump there is only one small stuffing box for the piston rod, so that the opportunity for leakage, with its resulting danger of fire, is reduced to a minimum. With an outside-packed plunger pump, there is more or less drip or leakage from the large stuffing boxes.

"The **handling of high-gravity fuel oils**, running from 30° Baumé up, and quite liquid in consistency, is usually best accomplished by the use of an ordinary duplex pump fitted with brass ring packing in the pump pistons and brass valves and special oil-proof gaskets in the pump cylinder joints to overcome the **solvent action of the oil.** For **heavy viscous oil**, like Mexican crude, the ordinary duplex piston pump is suitable, provided a size sufficiently large is selected to keep down the **oil velocities** through the ports and passages and valve seats in the pump cylinders to a minimum. For ideal conditions, the velocity through the valve seats should not exceed a speed of 100 ft. per min. for pumps of large capacity, and about half of this for small pumps. For **light oils** around 40° Baumé the above velocities can be doubled.

"The type of pump ordinarily used for pumping fuel oil to burners, where the oil pressure will not exceed 150 lbs. per sq. in., is a pump similar to the Snow Duplex piston-pattern pump. In selecting this type for low gravity oils, it is advisable to use nothing smaller than a 4½-in. × 2¾-in. × 4-in. size, as the ports and valve passages in small pumps are too restricted to operate successfully with oils of this character, and it might be well to remember that the 4-in., 5-in., and 6-in. stroke duplex fuel oil pumps have valve areas equivalent to 35 to 40 per cent of the piston area and should operate at speeds, say, not to exceed 30 to 40 single strokes per min. each piston, and 10-in. and 12-in stroke pumps 25 to 30 single strokes. For example, take the 6-in. × 4-in. × 6-in. size operating at 30 strokes—this would represent a piston speed of 15 ft. per min., and the resulting speed of the oil through the valves will be about 40 ft. per min., which is well within the limit.

"These pumps are very often set in duplicate on a cast-iron stand with an oil heater, strainers, by-pass valves, thermometers, gages, etc., as a Fuel Oil Pumping System.

"With this arrangement one pump is operative and the other is a reserve. A **pressure governor** or regulator is set for the oil pressure and controls the pump automatically. These systems are usually used in connection with low pressure burners of the steam or air atomizing type. In rare instances, the single cylinder pump is used, although in this country the duplex pump is favored.

"For **high pressure fuel oil** systems it is necessary to use pumps designed for operating against oil pressures up to 200 lbs. per sq. in. For horisontal work, the duplex-pattern pump of the valve-plate style makes a very efficient and satisfactory pump for the service. For Naval installations or for marine work where floor space is limited, the **vertical duplex** Admiralty pump attached to the bulkhead is usually employed. This type of pump is good for 300 lbs. maximum oil pressure.

"The same remarks made above with reference to the selection of the size and capacity of the pump, apply to the high pressure pumps.

"The duplex fuel oil pump has been universally regarded as the standard in the United States, while in England, for Admiralty purposes, they occasionally use vertical simplex pumps, which, if properly fitted up with suction air chamber and discharge air chamber, will give almost as steady and constant a flow of oil as the duplex pump.

"The **simplex pump**, both of the horisontal and vertical type, has better suction qualities than the duplex. The former are usually made longer stroke and permit the use of half the number of valves, which necessarily have to be of larger diameter than in the duplex pumps, and consequently are more favorable for the flow of heavy, viscous oils. On board ship where the bottom of the fuel oil tanks is placed below the location of the pump, the simplex pump, with its better suction qualities, will drain the tank in a more satisfactory manner than a duplex pump. A suction air chamber should

be so arranged as to permit the direction of flow through the suction pipe to cushion against the air or gas contained in the suction chamber as in Fig. 20, rather than have the flow pass directly into the suction opening of the pump.

"The **pump valve service** ordinarily used in fuel oil pumps is the plain **bronze disc** valve spring loaded or with a **wing-guided bronze** valve; the latter is used in the Snow heavy pressure pattern and in the Blake vertical duplex Admiralty pump.

"For large-capacity crude oil pumps we have adopted **double- and quadruple-beat valves**, which pumps have only one valve per section, the valve being of large diameter, with a very low lift. A valve of this type will give full passageway through the valve discs, as would be ordinarily the case with a disc valve, thus necessitating high lift; the oil will pass out through annular openings in the seat and have practically four outlets through the valve. This type of valve reduces slip to a minimum, reduces the wear and is practically noiseless.

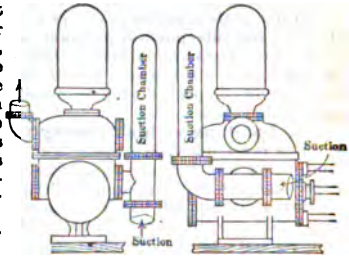


FIG. 20.

"With reference to the **manner of fitting fuel oil pumps**, there is some diversity of opinion, but the ordinary construction for **stationary service** is to install pumps fitted with bronze-lined pump cylinders, iron pump pistons, fitted with spring ring packing and steel piston rods. For **marine service** it is usual to install brass-fitted pumps, that is, arranged with bronze pump pistons and bronze piston rods. For handling **crude oils containing considerable sulphur**, steel piston rods and hard cast-iron pump cylinder linings give better service, especially if the oils contain considerable grit. A **fibrous-packed pump piston is not to be recommended**, as there will be danger of clogging the burners with shreds of packing passing out with the oil."

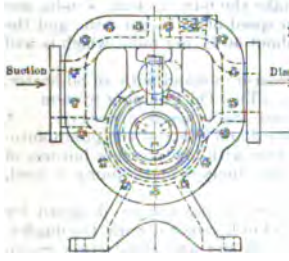


FIG. 21.

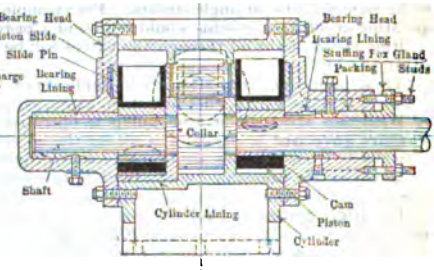


FIG. 22.

Kinney rotating plunger type H.P. pump for fuel oil service.

Kinney Rotating Plunger Pumps, now in use in the Naval Service for supplying fuel oil to the burners at a steady pressure are shown in Figs. 21 and 22, and described by Mr. E. B. Neal as follows:

"The Kinney Rotating Plunger Pump consists of a cylinder casting, having suction and discharge openings to the pump chamber The chamber is divided by a centre plate into two sections for the dual assembly of the pump working parts. The cylinder casting is accurately machined at both ends and the heads are attached to it by studs and nuts. The heads and cylinder casting are tongued and grooved, requiring simply a light coat of shellac to prevent leakage. **No gaskets are required.**

"The cylinder casting is provided with openings for flanged or screwed fittings for suction and discharge connections, according to the size of the pump. The removable cylinder heads are provided with properly packed bearings for the pump drive shaft. The bearings are provided with adjusting nuts. The drive shaft may be extended or adapted to receive any desired type of power transmission.

"The drive shaft passes through and actuates the piston cams to which they are keyed. The cams actuate the rotating pistons. The pistons rotate eccentrically, passing through the movable slide pins which open and close the discharge port of the rotating piston slides. All parts of the pump, except the drive shaft, are arranged in duplex opposed at an angle of 180 deg. giving perfect balance to the pump and no pulsation to the fluid.

"In operation, the rotation of the piston produces a high vacuum causing the liquid to flow through the suction port and follow the piston until the cylinder is completely filled. The port in the piston slide is then mechanically closed, remaining so until the cam has completed its full revolution. When the piston passes the suction port, it mechanically opens the discharge port and immediately begins forcing out the liquid in the cylinder chamber, at the same time drawing in a new supply. The ports are not in any way congested by valves or springs.

"Lubricating holes are drilled from the bottom of the piston slides through the pistons carrying the liquid which is being pumped to the cams under pressure, the parts thereby receiving perfect lubrication. The absence of mechanical contact between the pistons and the cylinder insures a minimum amount of wear on the cylinder and pistons.

"The **prominent features** of the Kinney Pump are its positive action, extreme simplicity, accurate and substantial construction resulting in high mechanical efficiency and long service. The pump requires little expenditure for repairs or maintenance.

The Kinney Pump may be connected to any type of power transmission; belt, chain or gear. It may be directly coupled to engine or motor of correct speed for the pump. The pump may be driven by portable or stationary steam engine, steam turbine, all types of electric motors, also oil or gasoline engines.

The **capacities** of the standard sizes in which this pump is Manufactured are shown in Table 2.

Table 2. Kinney Rotating Plunger Pumps for 250 Lbs. Pressure

Size of Pump	Theo. gals. per 100 R.P.M.	Maximum R.P.M.	Approximate Actual Gals. Delivery at 250 lbs. at Max. R.P.M.
	High-speed pumps		
4×1⅜×¼	1.8	1000	9
4×1¾×¼	2.25	1000	12
4×2×¼	2.6	1000	18
5×2×¼	3.2	1000	25
6×2½×¼	5.00	1000	40
6×2½×⅜	7.00	1000	60
8×3×⅜	11.25	1000	90
8×3×½	15.00	1000	120
	Low-speed pumps		
5×3×½	9.3	500	30
6×4×½	15.0	500	60
8×4×¾	30.0	450	105
9×4×⅞	40.0	450	140
10×5×1	60.0	400	190
12×6×1½	130.0	350	360
14×7×1¾	198.0	350	560
16×8×2	325.0	300	800
20×10×2½	630.0	275	1400
25×13×3	1225.0	250	2500

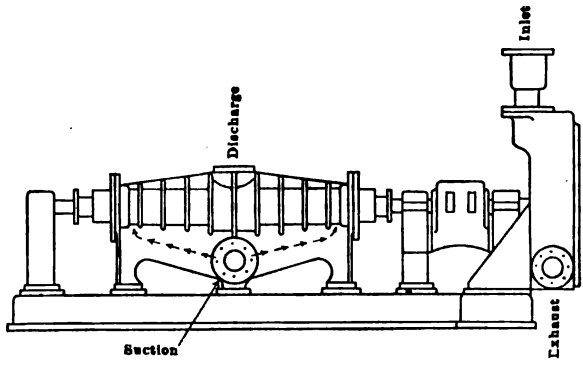


FIG. 23.—Quinby screw type H.P. oil service pump.

The Quinby screw pump which has recently been tested and approved for use in the Navy as a fuel oil service pump, is illustrated in Figs. 23, 24 and 25 and is described by Mr. Wm. H. Quinby as follows:

"Fig. 23 shows a general arrangement, complete with turbine; Fig. 24 represents a cross-section of the pump through one set of screws; Fig. 25 shows a pair of shafts with the screws and gears in mesh as they are in the pump.

"The four screws that act as pistons in propelling the liquid, are mounted in pairs on parallel shafts, and are so arranged that in each pair the thread of the screw projects to the bottom of the space between the threads of the opposite screw. The screw threads have flat faces and peculiarly undercut sides. The screws fit in the cylinder as shown in the cross-section, Fig. 24, and space enough is left between the screws and the cylinder and also between the faces of the intermeshing threads, to allow a close running fit without actual contact.

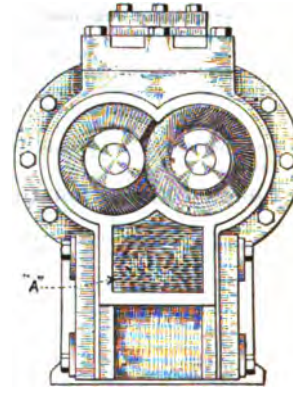


FIG. 24.—Quinby pump, cross section.

"The suction connection is at the bottom and opens into a chamber underneath the pump cylinder, as shown at "A" in Fig. 24. The liquid

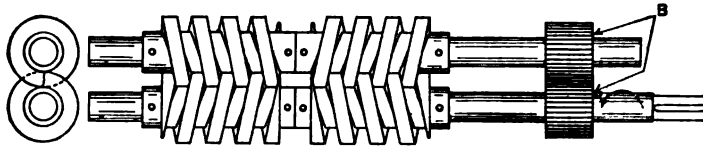


FIG. 25.—Quinby pump, detail of intermeshing undercut threads.

passes through this chamber to the two ends of the cylinder and is forced from the two ends toward the centre by the action of the two pairs of screws. The intermeshing portion of the opposite screw in each pair looks the liquid in between the threads; consequently each revolution delivers positively the contents of the space between one convolution of each of the threads less only the small amount of loss due to the clearance between the screws and between the screws and the cylinder. The discharge is in the middle of the top of the cylinder. The power to drive the pump is applied to one shaft and the second shaft is driven by means of a pair of gears shown at "B," Fig. 25. The necessary power may be applied by means of a belt and pulley or by means of a direct-connected motor or engine.

"The pump has **no internal packing**, no valves and no small moving parts. There is no end thrust or wear in the bearings due to the back pressure of the discharge, because the back pressure is delivered to the middle of the cylinder, and the endwise pressure upon the screws in one direction is exactly counterbalanced by an equal pressure in the opposite direction. The only opening from the pressure chamber is into the discharge pipe and the only packing is in the stuffing boxes where the two shafts pass through the cylinder heads.

"These pumps have been built to give a steady pressure of from 1 to 500 lbs. per sq. in., at capacities of from 5 to 3000 gallons per minute. Owing to the steady discharge pressure, no air banks are required on oil pressure lines.

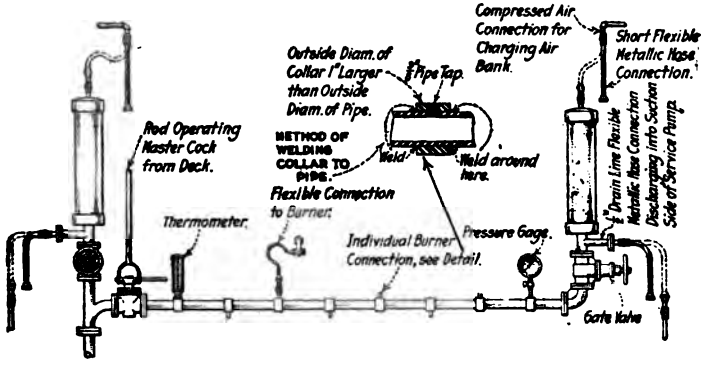

Compressed Air Connection for Charging Air Bank.
Short Flexible Metallic Hose Connection.
Outside Diam. of Collar 1" Larger than Outside Diam. of Pipe.
1" Pipe Tap.
Rod Operating Master Cock from Deck.
METHOD OF WELDING COLLAR TO PIPE.
Weld
Weld around here.
Flexible Connection to Burner.
1" Drain Line Flexible Metallic Hose Connection Discharging into Suction Side of Service Pump.
Thermometer.
Individual Burner Connection, see Detail.
Pressure Gage.
Gate Valve

FIG. 26.—Arrangement of oil piping on boiler front.

Fuel Oil Heaters. The object of heating fuel oil, and its effect on **viscosity**, is discussed under atomization, page 286. Heaters are provided for this purpose. For the very light, less viscous oils, heaters are of relatively little importance, but with the heavier very viscous grades of oil fuel, they assume a position of importance in the design of the complete equipment.

In every installation, **sufficient heater capacity should be provided to handle heavy oils at full power.** The most viscous oils will require heating in some cases to above 240 deg. F. At the same time they are very sluggish to heat. This capacity will be greatly in excess of that required for the light oils at high power and for either the heavy or light when cruising at reduced powers. If **one large heater** is used, it will be found very difficult to regulate the temperatures at capacities much below the designed maximum.

For this reason it is desirable to provide **several smaller heaters** connected in parallel or to provide some other convenient method of readily increasing or decreasing the amount of heating surface utilised, so that the portion in use can be operated at or near its maximum capacity. A sensitive control of the supply of steam to the heaters would complete an ideal arrangement for accurately controlling the oil temperature.

As the oil passes through the heater, some means of thoroughly mixing it by obstructions, such as baffling should be provided. Otherwise the oil will not be uniformly heated as it passes to the atomisers.

When no steam is available and the oil in use is very viscous, a small **charcoal-fired heater** will be necessary to heat the oil preparatory to lighting up. At the same time some means should be provided for draining the cold sluggish oil out of the pressure oil lines between the heater and the burners. This can best be accomplished by fitting a small ¼-in. drain line from the dead end of the supply line across the front of each boiler and leading to the suction of the service pump. If this is not provided, it will be necessary to drain this oil off into buckets. Fig. 26 illustrates a very good arrangement for providing a drain for both the oil line in lighting up and the air bank when recharging with compressed air.

Types of Heaters. There are a number of types of fuel oil heaters manufactured. The three types illustrated which are used extensively for marine installations cover the general principles of design of practically all types.

The **Cramp fuel oil heater,** manufactured by *The Wm. Cramp & Sons Ship & Engine Building Co.*, is illustrated in Fig. 27. This is a horisontal heater with **U tubes** which are rolled and expanded into a header. The oil enters the bottom chamber of the removable head. This chamber is separated by a division plate from the top chamber or outlet side of the heater. The tubes are seamless drawn steel, tested to 600 lbs. Steam for heating enters the top of the outer shell and surrounds the outside of the tubes. At the bottom and connected to the steam space is a drain for carrying off the condensate. This drain leads to a steam trap which discharges either to the filter tanks or to an inspection tank. Frequently on the heater or on the trap, or both, is located a water glass which will show oil on the top of the water in the glass, in case the heater leaks.

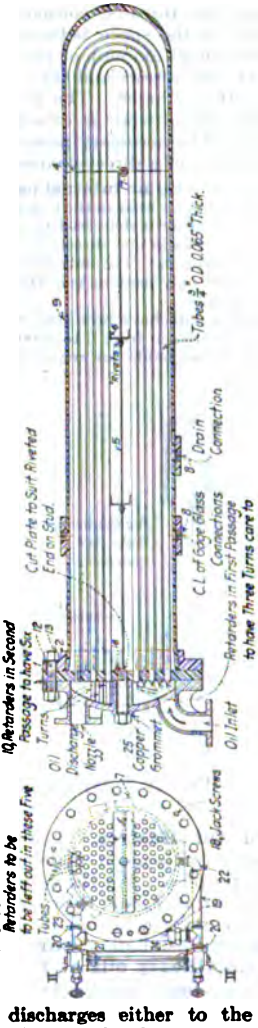

Tubes ⅝" OD 0.065" Thick
Cut Plate to Suit Riveted End on Shell
8" Drain Connection
C.L. of Gage Glass Connections
Retarders in First Passage to have Three Turns care to be taken to have Retarders Fitted as shown.
Retarders in Second Passage to have Six Turns
Oil Discharge Nozzle
Copper Grommet
Oil Inlet
Retarders to be left out in these Five Tubes
48 Jack Screws

FIG. 27.—Cramp H.P. fuel oil heater.

Fig. 28 shows the **Schutte & Koerting Co.'s "Z" Type** Oil Heater. It is a **straight tube** heater, steam passing through the tubes and oil passing around the outside of the tubes, a full heating effect being obtained by means of baffles. The tubes are seamless drawn steel, expanded into steel tube plates. A special arrangement is provided which permits of free expansion and contraction of the tube bundle. The whole tube bundle can be easily removed for cleaning.

Fig. 29 illustrates the **Reilly Multicoil Fuel Oil Heater.** The steam flows through the **spiral seamless steel tube coils.** The oil passes around the coils.

Most of the shipbuilding companies that install oil-burning apparatus make oil heaters.

Heating coils in storage tanks and bunkers are necessary to permit of the oil being pumped to the main heaters, where its viscosity is reduced sufficiently for atomizing; see **effect of heating oil**, p. 286 The tank heaters consist usually of coils of iron pipe about 1¼ in. diameter located near or around the pump suction. These coils must have sufficient capacity to heat the oil, as it enters the suction, sufficiently to reduce the viscosity to about 375 deg. Engler, or lower.

In the case of **fuel tanks or bunkers,** it would not seem necessary to heat the oil throughout the entire tank, and thus in the case of the double bottom, attempt "to warm up the whole ocean," provided local heating in the immediate vicinity of the pump suction can be carried out rapidly

DESCRIPTION OF FIG. 27.

NOMENCLATURE

Mark	Part	Material
11	Set of Gage Cocks	Comp. "M"
1	Tube Sheet	Wrought Steel Class "C" Boiler Plate
2	Flange	Forged Steel
4	Baffle Plate	Wrought Steel Class "C" Boiler Plate
5	Baffle Plate	
6	Stiffening Plate	
7	Pad	
8	Pads	
9	Shell	Seamless Drawn
10	Retarders	Wrought Steel Class "C"
11	Retarders	Shape $\frac{3}{64}$" × $\frac{3}{64}$" Thick
12	Studs	Wrought Steel Class "B"
13	Nuts	Wrought Iron
16	Studs and Nuts	Class "B"
17	Bolts and Nuts	Bolt Material
19	Pipes	Extra Heavy Brass
20	Nipples	
21	Tees	
22	Ell	
23	Ell	
24	Union	
25	Grommet	Copper
	Tubes	Seamless Drawn Steel Tubing
	Test Pressure (Water)	Sheel 400 Lbs. Per Sq. In. Head and Tubes 600 Lbs. Per Sq. In.

and uniformly enough to heat the oil as it is pumped away. In this case the size and heating surface in the heating coil should be proportioned to the rate at which the oil is pumped out of the tank rather than to the size of the tank.

The situation relative to **cargo tanks** is different, owing to the high rate of speed at which the oil is handled by the cargo pumps. Prevailing opinion seems to be opposed to carrying very viscous oil in the double bottoms, on account of the very large heaters necessary; and the usual practice is in favor of heating the oil throughout the entire tank previous to starting the pumps. Thus, in the cargo tanks of oil-carrying vessels, flat coils are often installed near the bottom of the tanks in addition to the coils around the suction, the combined heating surface of the two sets of coils being figured at the rate of 0.10 of a square foot of heating surface per barrel of oil of the total capacity of the tank. About 40 per cent of this surface is allotted to the suction coils and 60 per cent to the flat coils.

NOMENCLATURE FOR FIG. 28

Mark	Part	Material
A	Shell	Cast Steel Cl. "B" Annealed
B	Cover	Cl. "B" Annealed
C	Header	Cl. "B" Annealed
D	Tubes	⅝" O. D. Seamless Steel
E	Baffles	Steel Plate No. 10–14 B.W.G.

Strainers. Thorough straining of the oil before it reaches the pressure atomizer is essential, otherwise the small orifice and the tangential passages become clogged, thereby destroying the uniformity of atomization and reducing the capacity. Also the fine gritty sediment causes considerable wear of the orifices.

Basket type strainers are generally fitted on the suction lines. The mesh of the wire gauze used should be coarse; no finer than 28 mesh is recommended, as the primary object of the suction strainer is to protect the oil pump. Too fine a mesh is objectionable as it resists the flow of the relatively cool oil, causing excessive work on the part of the pumps and frequently resulting in lost suction.

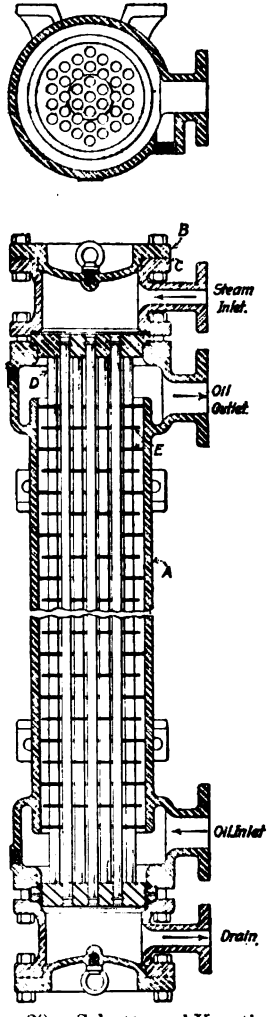


FIG. 28.—Schutte and Koerting "Z" oil heater.

Duplex high pressure basket type strainers are installed on the pressure side of the service pumps, generally between the pump and the oil heater, but in some cases on the discharge from the heater. The latter arrangement has a serious objection when the more viscous oils are used, due to the high temperature to which the oil is necessarily heated. In the merchant

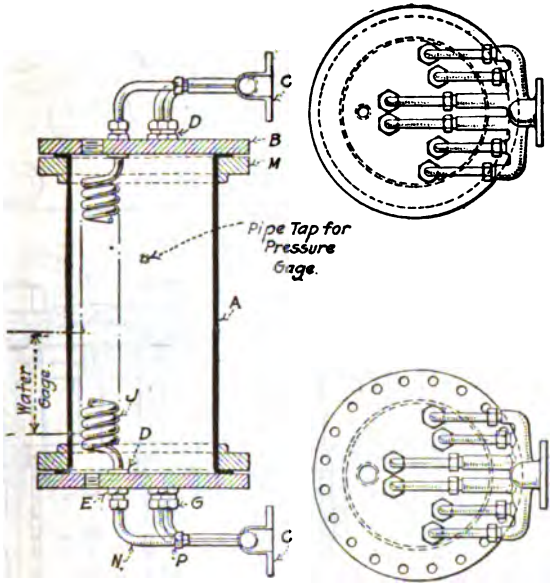


FIG. 29.—Reilly multicoil H.P. fuel oil heater.

Mark	Part	Material
A	Shell	Seamless Steel Tubing or Cast Steel
B	Head	Class "B" Boiler Plate
C	Manifold	Comp. "G"
D	Nipple	Comp. "G"
E	Lock Nut	Comp. "G"
G	Union Nut	Comp. "G"
J	Coil	1" O. D. Seamless Steel Tubing
M	Flange	Rolled Steel
N	Bend	1" O. D. Seamless Steel Tubing
P	Union Nut	Comp. "G"

ervice, where heavy oils are frequently heated above the flash point, there would be a real danger when it becomes necessary to cut out and open up a heater for cleaning. But there is also an advantage in locating the trainer on the discharge side of the heater. The heated oil is very fluid nd gives up the contained foreign substances more easily, also a much

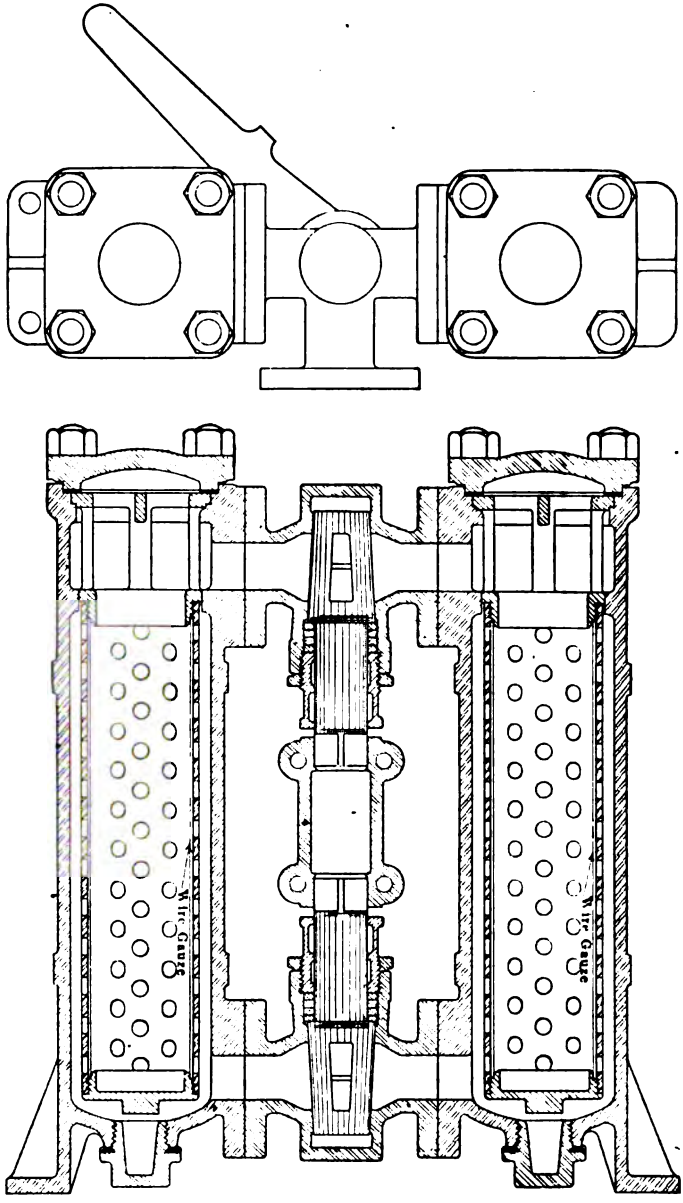
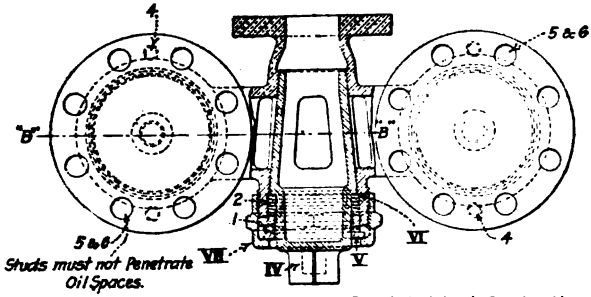
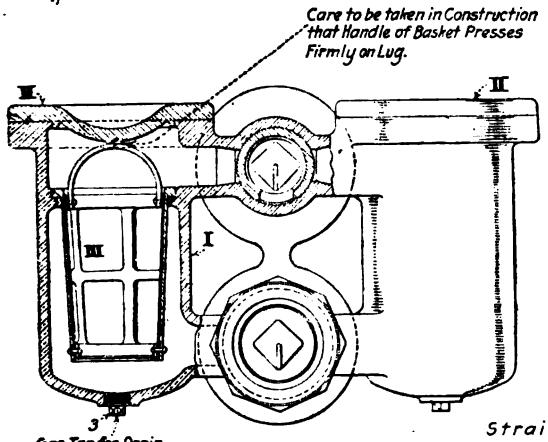


Fig. 30.—Schutte and Koerting H.P. Duplex fuel oil strainer.




Care to be taken in Construction that Handle of Basket Presses Firmly on Lug.
II
III
I
3
Gas Tap for Drain
Section B-B
Strainer.
Area through Basket 14.925 Sq In
Area through Wire Mesh 15.11 " "
Area of 2 in Nozzle 3.1416 Sq In
Ratio 4.81 to 1.

NOMENCLATURE

Mark	Description	Material
I	Body	
II	Cover	
III	Basket	
IV	Plug	
V	Gland	
VI	Ring	
VII	Nut	
1	Pin	Comp. "G"
2	Packing Ring	Vulca Beston
3	Pipe Plug	Comp. "Mn–R"
4	Jack Screws	Steel Class B Bolt Material
5	Studs	Steel Class B Bolt Material
6	Nuts	Wrought Iron
	Handle	Steel Class "C".
	Wire Mesh	Steel
	No. 30 wg., 0.014" Dia. Wire	

Test Pressure 600 Lbs.
Handle, Brass Strips, Screws in Basket Tinned.

FIG. 31.—Cramp's H.P. Duplex fuel oil strainer.

finer mesh gauze may be used. In the Naval service, brass wire gauze of 42 mesh to the inch is generally used for pressure strainers.

In addition to the above, many makes of pressure atomizers are equipped with **individual strainers.** If proper attention is devoted to the care and cleanliness of the large pressure strainers on the main supply line, strainers for individual atomizers are an unnecessary and expensive refinement.

Duplex strainers should be of sufficient **capacity** that one may be cut out for cleaning while operating under full power, so **constructed** as to be readily opened and cleaned and so **designed** that the flow of the oil shall be from the outside into the centre of the basket, in order that dirt and sediment will be deposited on the outside surface of the basket.

Figs. 30 and 31 illustrate two duplex pressure strainers that are extensively used in marine installations. The one shown in Fig. 30 is manufactured by the *Schutte & Koerting Co.* The one shown in Fig. 31 is manufactured by *The Wm. Cramp & Sons Ship & Engine Building Co.*

Oil Piping. In the **Naval service,** specifications require that the **pressure service piping** be seamless drawn steel tubing with "expanded on" steel flanges. Flanged joints are ground and are required to be oil tight under test without the use of gaskets; test pressure, 600 lbs. Suction **piping** is steel, lap-welded, and expanded into flanges. Thin, tough, fibrous paper gaskets are allowed; test pressure, 50 lbs. Oil piping should be exposed as much as possible to insure prompt detection of leaks.

For **merchant service,** extra-heavy welded-iron or steel pipe, with screwed joints and with extra-heavy galvanized-iron fittings, is used. Flanges are screwed on the pipe and manila paper, card board or special oil-proof packing may be used for gaskets. Rubber is not allowable on account of sulphur in the oil. Copper piping is not used on account of the sulphur, but brass and composition fittings, valves, unions, etc., may be safely used.

The suction piping should be large. The **Newport News Rule** limits the designed velocity of Mexican oil through suction pipes to not over 20 ft. per minute, the oil being heated to reduce the viscosity to about 300 deg. Engler. For discharge pipe lines they consider 100 ft. per minute allowable in small pipes, the viscosity being reduced to 15 deg. Engler or under.

Mr. C. F. Bailey of Newport News Shipbuilding and D. D. Co. states that "As we pipe from the service pumps to the oil burners we reduce the speed of the flow near the end of the lines, *i.e.*, we do not reduce the piping in proportion to the oil used, as we find it necessary to reduce the flow in order to maintain the pressure at the burners on the end of the line farthest from the service pumps. With Navy fuel oil, and where we do not make allowance for changing to heavy oil like Mexican oil, we allow about double the speed in the discharge pipe to the burners that is allowed with Mexican oil. For instance, in the *U. S. S. Pennsylvania*, the service-pump discharge to the heaters is proportioned for about 130 feet velocity per minute, the discharge from heaters to burners is proportioned for about 230 feet per minute in 2½-in. and 2-in. pipes, respectively. As the lines reduce we have in 1½-in. pipe, 185 feet; in 1¼-in. pipe, 195 feet; and in ⅜-in. pipe, to each burner, 128 feet; velocities per minute. Our Destroyer practice in regard to piping and oil speeds is substantially the same as that for the *U. S. S. Pennsylvania.*"

Valves. For valves on **suction lines** designed for viscous oils, the **gate** valve is preferable, on account of reduced friction. On **delivery lines** globe valves of a regrinding type give satisfaction. There is no occasion to use needle valves. Where **fine regulation** is required, as in some cases with steam atomizers, there are several types of valves which open gradually on slotted or V-shaped passages which give better and more consistent results than the needle type. All valves for **high-pressure** work should be extra heavy with bonnets screwed over, not into, the valve body. Specially designed and packed **plug cocks** may be used in small sizes for quick action.

Individual atomizers are frequently connected to the oil supply line by ¼ or ⅜-in. flexible piping. Extra heavy annealed copper pipe is used for this purpose but there is now on the market an excellent and very flexible oil tight braided tubing, known as "Riflex," that is superior to the copper pipe

Air Cushion Chambers. Fig. 26 shows oil supply pipe across front of boiler, drain lines and air banks as fitted with reciprocating fuel oil service pumps. It is a matter of opinion as to whether air chambers should be fitted at the discharge of the service pumps, on the supply line just before the oil reaches the atomisers or at the dead end of the supply line. In any case the flow of oil should be direct into the cushioning chamber and if a valve is fitted to cut out the chamber it should be a gate valve.

BOILER ROOM INSTRUMENTS

The draft gauge, commonly known as a **manometer,** is used to measure the pressure in the fire room above the atmosphere and also pressure differences in different parts of the boiler. It provides an excellent means of **measuring draft** in connection with the study of furnace efficiency and boiler design.

Besides carrying off the waste gases of combustion, the smoke pipe creates a draft or difference in pressure which, at natural draft, is the means depended on to draw the air for combustion into the furnace. This draft is dependent on the difference in weight between the column of hot air in the smoke pipe and the weight of an equal column of relatively cold outside air.

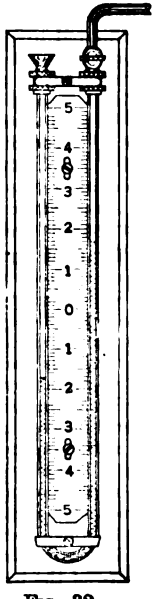


FIG. 32.

FIG. 33.

The scale is graduated to record the pressure or draft in inches of water. One inch of water at 62 deg. F. is equal to 0.036 lb. per square inch = 0.576 ounces per sq. in.

Fig. 32 illustrates the ordinary type of **U tube.** The pressure is the difference in the water level in the two tubes.

Fig. 33 shows a much better design. It is **direct reading** and the visible motion of the liquid is double that in the ordinary type.

Fig. 34 illustrates a **differential draft gauge** which permits of much more accurate readings. It is not practical to use it on board ship at sea as it requires careful leveling.

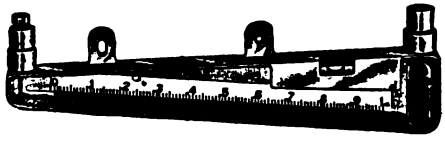


FIG. 34.

Oil Thermometers should be installed in the following locations: on the pressure oil supply line to each boiler, to record the temperature of the oil just before it reaches the atomizers (see Fig. 26); on the discharge from the oil heater, and, on the suction from each storage tank. In addition, a thermometer is frequently placed on the oil service pump discharge to the oil fuel heater.

Oil Pressure Gauges should be provided on oil supply lines at each boiler near the atomizers, and on the discharge from each pump.

Smoke Pipe and Furnace Temperature. In the study of boiler and furnace efficiency and in the computation of heat balances it is essential that the smoke pipe or uptake temperature of the waste gases be known, and it is desirable to know furnace temperature. **Nitrogen filled mercurial thermometers** graduated to 1000 deg. F. or **base metal thermocouples** are very satisfactory for smoke pipe temperature observations. The recording type **thermoelectric pyrometer** is frequently preferred as giving a continuous record.

For observations of furnace temperature the **optical pyrometer** is perhaps the best. The use of **precious metal thermocouples** is expensive on account of frequent renewals and is generally unsatisfactory as the temperatures of oil-fired boilers is close to the maximum range, and quite often is above the melting temperature of platinum. Of the many thermocouples, the **potentiometer type** is known to give excellent service.

Potentiometer Type of Pyrometer. When two dissimilar metals are joined, forming a completed circuit, due to their difference in potential and the difference in temperature of the two junctions an E.M.F. appears and there is developed a flow of current. The resultant net or effective E.M.F. will be proportional to the difference in temperature of the two junctions. Such a device is known as a thermocouple.

In the ordinary electrical pyrometer a galvanometer or millivoltmeter, inserted in the circuit, measures the current flowing. The deflection of the millivoltmeter, however, is dependent not only upon the electromotive force generated but also upon the resistance of the circuit and upon the adjustment and condition of the instrument.

In a pyrometer of the potentiometer type the E.M.F. developed by the thermocouple is balanced against a known E.M.F. and the reading is obtained when this balance is exact, that is, when no current flows. Under these conditions it does not matter what changes may take place in the resistance, length or size of the leads, or of the thermocouple.

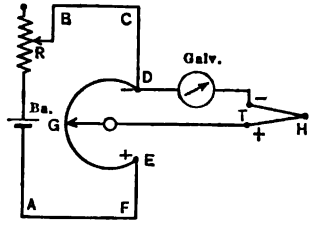


FIG. 35

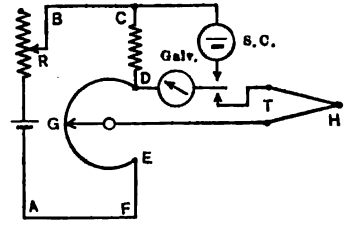


FIG. 36.

Fig. 35 shows a **wiring diagram** of the potentiometer. The thermocouple is at H, with its polarity as shown. It is connected to the main circuit of the potentiometer at the fixed point *D* and at the point *G*.

A current from the dry cell *Ba* is constantly flowing through the main, or so-called potentiometer circuit, *ABCDGEF*. The section *DGE* of this circuit is a slide wire of uniform resistance throughout. The scale is fixed on this slide wire. The current from *Ba* as it flows through *DGE* undergoes a fall in potential, setting up a difference in voltage, or an E.M.F. between *D* and *E*, and also between *D* and all other points on the slide wire. The polarity of the former E.M.F. is in opposition to the polarity of

the thermocouple which connects into the potentiometer at *D* and *G*. By moving *G* along the slide wire a point is found where the voltage between *D* and *G* in the slide wire is just equal to the voltage between *D* and *G* as generated by the thermocouple. A galvanometer in the thermocouple circuit indicates when the balance point is reached, since at this point the galvanometer needle will stand motionless when its circuit is opened and closed.

The voltage in the slide wire *G* will vary with the current flowing through it from the cell *Ba*; a means of standardising this should be provided. By introducing a standard cadmium cell S. C. Fig. 36 this can be effected. The cell is connected at *C* and *D* to the potentiometer circuit whenever the potentiometer current is to be standardised. At this time the galvanometer is placed in series with S. C. The variable rheostat *R* is then adjusted until the current flowing is such that as it flows through the standard resistance *CD*, the fall in potential between these two points is just equal to the voltage of the standard cell S. C. At this time the galvanometer will indicate a balance in the same way as when it was used with a thermocouple. By this operation the current in the slide wire *DGE* has been standardized.

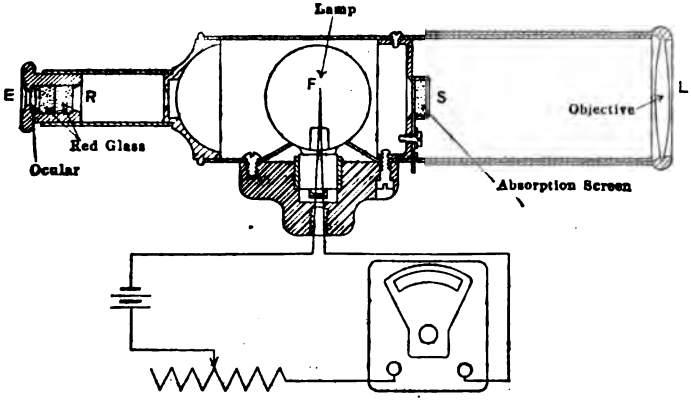


FIG. 37.

The Optical Pyrometer. For measuring temperatures from a dull red (about 1100 deg. F.) to the highest known temperature, an optical pyrometer using a balance method of comparing luminous radiation, is the most accurate. The greatest success has been obtained by separating out one color, usually red, from all the light emitted by an incandescent body and comparing the intensity of this one colored light with the intensity of the light of the same color from a standard source of light. The eye is very sensitive in judging which is the brighter of two areas, especially when one area is small and superimposed upon the other. The brightness of the two areas cause them to be made equal by varying the intensity of the standard of comparison.

The principle of an excellent type of optical pyrometer is illustrated in Fig. 37. *L* is a lens through which radiations from the incandescent body are brought to a focus at the point *F*. In the position of the image produced by the lens is placed a tungsten lamp filament. This filament receives current from two No. 6 dry batteries, contained in a portable case which also contains a rheostat and an accurate milliammeter. The incandescent filament and the image produced by the lens is observed through the eyepiece *E*.

By means of the rheostat the current through the filament is adjusted until the brightness of the filament is just equal to the brightness of the image produced by the lens *L*;

whereupon the filament blends or becomes indistinguishable in the background formed by the image of the heated object. When a balance has been attained, the observer notes the reading of the milliammeter. The temperature corresponding to the current is thus read from a calibration curve supplied with the instrument.

As the intensity of the light emitted at the higher temperatures becomes dazzling it is found desirable to introduce a piece of red glass in the eyepiece at *R*. This also eliminates any question of matching colors or the observer's ability to distinguish colors. It is of further value in dealing with bodies which do not radiate light of the same composition as that emitted by a black body, since the intensity of radiation of any one color from such bodies increases progressively in a definite manner as the temperature rises. The intensity of this one color can therefore be used as a measure of temperature for the body in question.

The practical advantages of the instrument are: (1) It is light and portable; (2) large surface of sight is not required; (3) distance of the observer from the hot object is not important; (4) adjustment is rapid; setting is precise; and (5) intensity, not color, of light is matched.

Smoke Chart. In order to have a uniform system of determining and recording the density of smoke produced, the smoke chart, otherwise known as **Ringelmann's Scale,** is used. There are six cards, numbered from 0 to 5; the first is entirely white, Nos. 1 to 4 are ruled into small squares by lines of varying thickness, and No. 5 is entirely black. The ruled cards have 17 horizontal and 10 vertical lines, spaced 10 mm. apart between centres. For card No. 1 the lines are 1 mm. in width; for No. 2, 2.3 mm.; for No. 3, 3.7 mm.; and for No. 4, 5.5 mm. No. 0 represents no smoke. Nos. 1 to 4 represent 20, 40, 60 and 80 per cent smoke respectively, and No. 5 represents 100 per cent smoke.

These cards are hung in a horizontal row about 50 feet or more from the observer and in line with the smoke pipe. At this distance the lines on the ruled cards are not visible and the cards appear to be of different shades of gray. The observer glances from the cards to the smoke and decides which one is nearest in shade to that of the smoke. Observations should be made and recorded in terms of percentage of smoke and the average taken as the average density of the smoke for the test.

Flue Gas Analysis is essential to the accurate determination of the gases of combustion in connection with apportioning of the heat losses. An intelligent use of the gas analysis instrument is the most valuable aid in obtaining the maximum furnace efficiency. It is a positive **gauge on the amount of excess air** and is probably of much more assistance when oil is burned than it is when coal is used as fuel. This instrument and its use is discussed on page 482.

Oil Meters should not be considered as replacing tank soundings, but they are of great assistance to the engineer in procuring hourly records of oil consumption in connection with computing curves of consumption of oil under various operating conditions. With proper precautions, they serve as an excellent check on the tank soundings. If means are provided for circulating the oil through the system prior to lighting up, care should always be taken to by-pass the meter, or the check on daily consumption will be destroyed.

Meters that are accurate within 1 per cent through a large range of capacity and pressures are now being manufactured. Two **representative types** that have given satisfactory service are the Empire Oil Meter and the Bowser Oil Meter, described hereafter.

The Empire Oil Meter is a positive displacement meter of the oscillating piston type, and consists of a **measuring chamber** enclosed in a case, and provided with the necessary **gear train** for transmitting the motion of the piston to the **registering mechanism.**

The measuring chamber of the Empire Oil Meter consists of an annular chamber, formed by an outside ring and two heads. On these heads circular projections form the inner wall of the annular chamber, a space being left between these projections to allow for the working of the web of the piston.

The necessary outlet and inlet ports are also formed in the heads of the measuring chamber. A bridge is placed in the annular chamber separating the inlet and outlet ports.

The piston is a cylindrical ring provided with a web which works with an oscillating motion around the central abutment of the measuring chamber and is provided with a projection on its center which acts as a guide for the piston and keeps it in joint, forming contact with both the inside and outside walls of the annular measuring chamber.

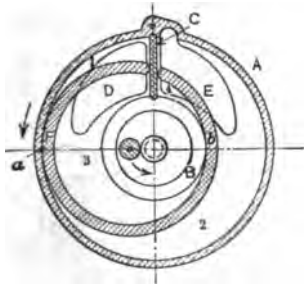


FIG. 38

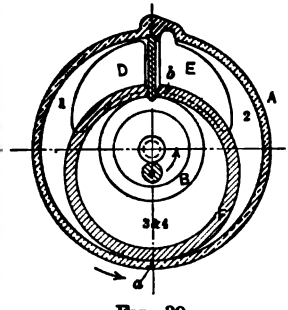


FIG. 39.

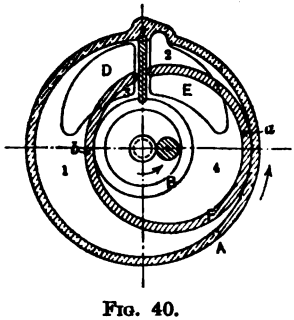


FIG. 40.

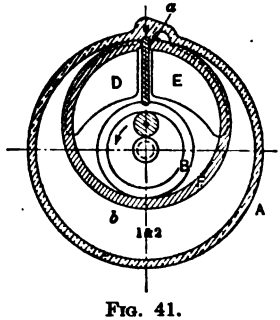


FIG. 41.

The accompanying diagrams, Figs. 38, 39, 40, 41, and 42, and the description, give more in detail the action of the measuring chamber of the meter.

The case chamber *A* is of cylindrical form and has an abutment *B* central with the case chamber, and the diaphragm or partition *C* makes a division between the abutment and the case wall. On each side of this abutment are ports *D* and *E*; *D* is the inlet port and *E* the outlet port.

The piston *F* is a split ring of cylindrical form having a slot which passes over the diaphragm *C*. This piston has a motion around the case in which the centre of the piston describes a circle around the centre of the case chamber, but as the piston does not itself rotate, but slides back and forth upon this slot on the diaphragm *C*. The meter has

been called an "oscillating meter," to distinguish it from meters in which the piston may rotate upon its own axis.

Referring to Fig. 38, the piston is shown as traveling in the direction of the arrow, and it will be seen that there is a contact of the piston at the point "*a*" with the case walls, and at "*b*" with the wall of the abutment. The piston also bears upon the diaphragm *C*. Therefore, the piston divides the case chamber into the spaces 1, 2, 3, and 4, in which the spaces 1 and 2 are formed outside of the piston, the chambers 3 and 4 are formed inside of the piston.

As the piston moves in the direction shown by the arrow in Fig. 38, inlet spaces 1 and 3, which communicate with the inlet port *D* are enlarging their volumes while outlet spaces 2 and 4, which communicate with the outlet port *E*, are reducing their volumes.

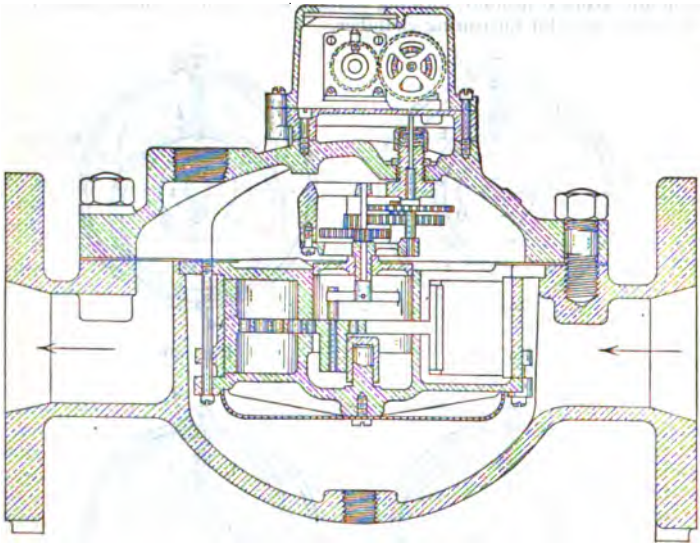


FIG. 42.—Empire H.P. oil meter.

At the position shown in Fig. 39, inlet space 1 continues to increase its volume. Space 4 becomes zero. Space 3, inside the piston, has reached its maximum volume and has closed communication with the inlet port *D*. At this point space 3, merging with space 4, changes from an inlet to an outlet space and opens communication with outlet port *E*. Outlet space 2 is rapidly reducing in volume. A new space 3 is beginning to form (seen in Fig. 40).

In Fig. 40 the piston has advanced another quarter cycle; space 1, still in communication with inlet port *D*, continues to enlarge; new inlet space 3 is enlarging; spaces 2 and 4, in communication with outlet port *E* are reducing in volume.

When position Fig. 41 is reached, space 1 has attained its maximum size and merging with space 2, becomes an outlet space; it opens communication with outlet port *E* A new space 1 begins to form (seen in Fig. 38).

The pressure upon the piston is all in the line connected by the points of contact, *a* and *b*, and this is at right angles to the motion of the piston.

The oil meter as arranged for use in the U. S. Navy is provided with a straight reading counter which reads in gallons and is driven by a pair of

bevel gears and a counter shaft, calibrating gears being installed between the countershaft and the register shaft.

The Empire Oil Meter has been made in a number of special sizes for heavy pressure service for the United States Navy. Among these sizes being the following:

Sizes, inches	1½	2	2½	3	3½
Capacities, gal. per min.	1 to 20	2 to 40	2 to 50	5 to 75	5 to 100

The Bowser Fuel Oil Meter of the positive displacement type, for accurately measuring the quantity of liquid flowing through a pipe, is shown in Figs. 43 and 44. This device consists essentially of a **rotating element** 2 eccentrically located in a body 1. Mounted in the rotor are two sets of **sliding**

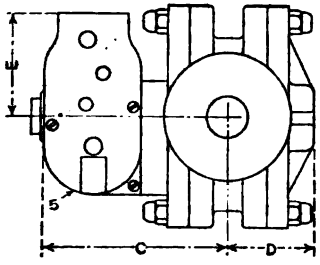


FIG. 43.

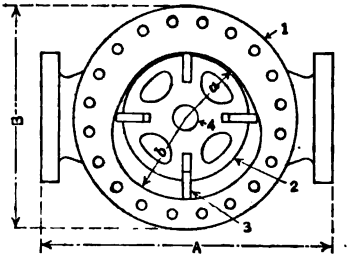


FIG. 44.

Bowser H.P. fuel oil meter.

blades 3, the edges of which are continuously in contact with the bore of the body. This bore is formed by an arc of minor radius *a* at the top and an arc of major radius *b* at the bottom, both struck from the centre upon which the rotor revolves. Each arc forms approximately 90 deg. of the perimeter of the bore and they are connected by curves. The inlet and outlet openings are located opposite each other and are rectangular in shape, extending the full 90 deg. on each side of the bore formed by the curves.

As the liquid enters the meter it fills the cavity formed by the difference between the minor and major radii of the bore, and due to the pressure exerted upon the unbalanced area of the blade the rotor is revolved. In revolving, the blades, which are 90 deg. apart, form a positive seal between the inlet and outlet ports, thereby causing a definite amount of liquid to be passed through the meter for every quarter revolution of the rotor.

This motion is transmitted by means of the rotor shaft (4) through carefully calibrated gearing to the dials which read directly in gallons. This counter mechanism is mounted on the head of the body (either right or left), and is contained in **casing** 5.

To reduce the power required to operate the meter to a minimum, special care is exercised in the design and machining of the bore in the body. The curves connecting the arcs are of the form known as uniformly accelerated and deaccelerated motion which insures the least friction of the sliding blades upon the bore and still maintains a positive seal.

This meter is built in nine sizes and three weights, to work under pressures up to and including 1000 pounds per sq. in. These meters are also designed to include a valve which can be set to allow a predetermined quantity of

liquid to be discharged through the pipe when it will automatically close, and if desired, automatically stop the pump.

For fuel oil and similar liquids Table 3 of **capacities** may be used to determine the size of meter to use.

Table 3.

Number of meter	Gallons per minute	*A* inches	*B* inches	*C* inches	*D* inches	*E* inches
0	5	6½	4⅜	3⅛	2	
1	9	10	6⅞	7⅛	2 9/16	4⅜
2	14	10	6⅞	7⅛	3 1/16	4⅜
3	30	12¼	9¼	7¾	4	4⅜
4	50	12¼	9¼	8¾	5	4⅜
5	100	19	13½	9 9/16	5⅞	4⅜
6	200	22¾	17	10 13/16	6⅝	4⅜
7	450	25	21	12	8⅞	4⅜
8	900	30	25½	15¾	11⅝	4⅜
9	1350	34	30	18	14	4⅜

Measurement of Boiler Water or Steam is rarely accomplished on board ships in service with any degree of accuracy. On trial trips, **calibrated tanks** are frequently installed on deck but these are removed upon the completion of the trials. In general, calibrated tanks and tanks on scales are too large and unhandy for permanent installation on board ship. **Feed tank soundings** serve to check the amount of water on hand and aid in keeping a record of the make up feed but they aoffrd no data of value in connection with boiler evaporative efficiency. A suitable instrument, for use on ship board, that will supply reasonably accurate data on the actual water evaporated into steam by the boilers would do more to save fuel than any other one instrument. This data in conjunction with fuel consumption would enable the engineer to set a standard of fire room efficiency.

The **venturi meter** meets the above requirements, but it necessitates the use of longer leads of straight pipe than are generally available on ship board. This objection also applies, in most cases, to the **steam flow meters** on the market. There is an additional objection to the majority of flow meters, though giving excellent service in stationary plants, where they can be accurately leveled, they are sensitive to motion on ship board. These meters are being rapidly developed and, it is believed, will soon be found in successful operation at sea.

OPERATION

Practical Requirements. The **oil** must be finely **atomized** just prior to coming in contact with the air for combustion. The requisite **quantity of air** must be intimately mixed with the oil and **excess air** reduced to a minimum. A high **furnace temperature** must be maintained, by the use of sufficient refractory surface, to assist in complete combustion. **Combustion** must be **completed** before the combustible gases come in contact with cooling surfaces and become chilled below the ignition temperature. Boiler **heating surfaces** must be **protected** from localised heating caused by a blow pipe action; heating should be uniformly distributed.

Prior to Lighting Fires. Ensure that the whole **oil system** is clean and free from dirt, waste, flakes of shellac, red lead, etc. Inspect and clean all **strainers**, this should be done frequently while underway; see that wire gauze is in good condition and that the proper mesh is used. Thoroughly clean **atomising tips**, place and properly adjust same in air registers. Close individual atomizer throttle valves. Drain or pump overboard, **water in oil tanks** that has settled out. If oil is very cold and viscous, and pumps have difficulty in getting a suction, use heating coils provided in the tanks. If **air chambers** are provided, drain and charge with compressed air.

Ensure that furnace and bilges are free from **explosive gases.** Pump out and steam out all bilges; wipe up all oil on floor plates and elsewhere. It is excellent practice to do this at least once each watch while steaming. Blow out furnace with air or steam, or, if neither are available, thoroughly ventilate.

Start up pumps and put oil pressure on lines. Inspect for **oil leaks.** By-pass meter until ready to light up, otherwise it will operate while air is being pumped out of the line and while oil is being circulated. Warm the oil with the heater. See that torch is provided and ready. If light oil is being used, everything is in readiness for lighting fires as no trouble is experienced in atomizing oils above 23 deg. Baumé at usual temperatures. When no steam is available, the hand oil pump is used until steam is generated.

A suitable **lighting torch** may be made by securing a few loose turns of asbestos wicking on the end of a rigid wire rod about 2½ to 3 feet in length.

Dampers in uptakes or stacks are not generally installed in oil burning boilers, but if provided, they should be wide open before fires are lighted.

Lighting Fires. Note that there is sufficient **pressure** on the oil and that it is at the proper **temperature** for good atomization. Make sure that the center atomizer has a small size orifice. Light torch and place near and just under the atomizer. Stand clear as a precaution against blow backs. Have **blowers** warmed up and just ready to turn over if steam is available. Open **atomizer throttle**, and the oil will immediately light up and readily burn. Be sure to note that the oil is being properly atomized. A **dirty atomizer** will cause heavy black streaks. The presence of considerable water is indicated by sputtering that is easily detected. **Every precaution should be taken to prevent unburned oil from collecting in the furnace as the hot brick gasifies it and there is serious danger of a violent explosion.** Proper precautions and common sense eliminate such dangers.

When extra burners are lighted it is preferable to use the torch each time while the furnace is still relatively cold. After steam is generated and the furnace is hot, **additional burners** may be safely **lighted** from adjacent burners in operation. No matter how hot the furnace may be, if all burners are temporarily secured, **do not attempt to relight from the incandescent brick work.** The most serious accidents have been occasioned by this practice.

Always **light the centre burner first,** then adjacent ones on each side. Keep the number in operation arranged symmetrically around the centre. This gives a more uniform distribution of combustible gases in the furnace.

It is advisable to avoid raising steam too rapidly as the rapid change in temperature is hard on the refractory lining tending to spall and loosen the brick. With express type boilers, rapid raising of steam from a cold boiler is more injurious to the brick work than it is on the boiler. With Scotch boilers this condition is entirely different.

Unless a boiler is to be cut out, at no time should the steam pressure be allowed to fall below 75 per cent of the authorized boiler working pressure. With a **low steam pressure** the blowers cannot be speeded up to supply air for lighting the additional burners necessary to meet increased demand for steam without making smoke and causing vibration.

Pressure on Oil. The oil pressure does not enter into the problem of **burning different oils.** The question is purely a matter of ascertaining the temperature requisite in the case of each grade of oil to obtain the proper viscosity. When this viscosity is maintained, the oil pressure may be the same with every grade of oil used. With atomizers properly designed and kept properly cleaned, oil pressures from 125 to 250 lbs. give practically, equally good atomization. **Increasing the pressure** above 125 lbs. merely **increases the capacity** which, other things being equal, is very nearly proportional to the square root of the pressure, *i.e.*, **to double the capacity will require four times the pressure.** However it must be remembered that excessive oil pressures require a corresponding increase in air pressures.

Difficulty in Lighting Fires. Trouble is to be expected with heavy oils in first lighting up. It will probably be necessary to install small charcoal oil heaters to be used, when no steam is available, for warming up heavy oil before lighting up. A further aid in lighting up under a dead boiler would be a half inch line tapped off the dead end of the oil line across the front of the boiler and led to an oil tank or to the service pump suction. Before lighting up, the valve on this line may be opened and the oil circulated through the system until hot oil from the heaters fills same up to the burners. This will eliminate the necessity of drawing off the cold oil in buckets at the burners to obtain heated oil in lighting fires.

Heavy Soot on Tubes. Heavy deposits of soot on tubes always accompanies heavy smoking, no matter what kind of oil is used, and is a sign of incomplete combustion. This condition is exaggerated under natural draft and at low air pressures and may be due to several causes; insufficient air for combustion, poor atomizers, oil too viscous, atomizers improperly adjusted, poor design of installation, unstrained oil, etc.

Heavy Smoke. Smoke is an index that something is wrong. When complete combustion is obtained smoke is eliminated. But it should be kept in mind that complete combustion in a boiler furnace does not necessarily mean maximum efficiency of the boiler. **Complete combustion with a minimum of excess air does mean maximum furnace and boiler efficiency.** For this reason it is often stated that the best practical results are obtained with a very light gray or brown smoke leaving the smoke pipe, because when the smoke pipes are smokeless it is difficult on board ship to determine the amount of excess air entering the furnace, and as this amount increases, the furnace efficiency falls off rapidly. A trace of light brown smoke emitted in a uniform manner from the smoke pipes generally indicates about 10 to 15 per cent excess air. To remove all traces of smoke will, in most cases, require an excess of 50 per cent or above.

If smoke is due to other causes than insufficient air, these causes must be eliminated before the smoke can be used as an indicator of the best practical furnace efficiency. For example, if one or two atomizers are dirty, **streaks of dense oil** enter the furnace, and pass out the smoke pipe partially consumed, causing smoke even though there may be a very high percentage of excess air. A trained eye can often tell the difference between smoke from such causes, and that due to lack of air, as it is generally irregular and streaked.

Carbon on atomising tips is the surest sign of their **improper location.** If the atomiser is shoved in too far, an eddy is formed in the entering air currents. A fine fog of oil is drawn back behind the face of the tip and is deposited on this face, on the side of the tip, and occasionally for some distance back on the burner pipe. This latter is an extreme case, where the atomizer has been shoved in several inches beyond the proper point. In extreme cases jets of ragged flame are also drawn back by this eddy, sometimes completely hiding the tip from view.

Carbon on Furnace Openings. If carbon forms **on edges of the furnace openings,** the atomizers have been drawn back in register too far, or as is often the case, the furnace opening is irregular due to patching up of the brick work, and edges of brick or mortar project into the outer film of the cone of oil. If carbon forms **on atomising tip and also on furnace opening** and there are no mortar or brick projections, the opening has been reduced to such an extent that the burner cannot be drawn back far enough to prevent the eddies in the air current described above. This is a fault of design and can be eliminated by increasing the furnace opening or by a new design of register. It can also be eliminated by using an atomizer that gives a smaller angle to the cone of oil, but this is not recommended unless there is sufficient depth to the furnace, to permit of complete combustion before any oil strikes the back wall. This is rarely the case in marine boilers.

Hard carbon deposits on tubes and on furnace sides and bottom are due to the oil striking the tubes or the brick before it has had time to burn. This is frequently experienced even with the light grades of oil, and is caused by two defects: one, design; the other, operation. Wing and bottom burners are sometimes placed too near the tubes and the furnace bottom. As a result oil strikes these surfaces and is carbonized before it has time to form an intimate mixture with the air and become completely consumed. Due to the necessity for cutting down the furnace volume in Naval boilers to a minimum this trouble is frequently met with.

Under operation, the causes are the same as those that produce excessive soot and smoke; namely, the failure to obtain an intimate mixture of the entering air and the oil at the register; this is more often caused by the improper location of the atomizing tips than anything else. It is sometimes due to too much excess air entering around the cone of oil and thereby cooling the flame down so that combustion is not obtained until some distance in the furnace; this permits the particles of oil thrown off at the greater angles from the axis of the tip to strike the furnace bottom, side walls or tubes near the front of the furnace before they are burned. This cools them below the combustion temperature. The lighter gases vaporize and burn; the carbon remaining adheres to the surfaces as a solid mass. The same occurs when there is insufficient air for complete combustion, but this need not be considered, as the tendency on board ship is toward too much excess air.

The noticeable increase in the amount of carbon deposits when viscous oils of asphaltic base, such as Mexican and California, are burned, is probably due to the large proportion of unsaturated hydrocarbons contained. These are heavy and slow burning. Other things being equal, they require a longer time in which to become completely consumed. Therefore they require a longer combustion path or some means must be resorted to that will hasten the combustion. This can be accomplished by increasing the velocity of the entering air for combustion. The area of the openings in the air controlling registers must be closed a proportional amount so that the volume of air remains the same. It is frequently stated by engineers that "more air is needed to burn Mexican oils." This is not correct and should be stated as follows: A greater air pressure is required to burn Mexican oils, as an increase in the air pressure is necessary to obtain an increase in the entering velocity.

PRECAUTIONS IN MANAGEMENT

While oil-burning boilers are subject to the same derangements and casualties that may occur in coal-burning plants, there are in addition a number of casualties which are peculiar to the former installation.

A flareback is due to the explosion of a mixture of oil vapor or gas and air in the furnace. Such an explosion may cause grave injury to the personnel and serious damage to the boiler, its fittings and attachments. A flareback is most apt to occur when lighting up, or when attempting to relight a burner from a hot brick wall.

Precautions to Prevent Flarebacks. (1) Oil must not be allowed to accumulate in the furnace. All oil on the furnace floor must be wiped up and the furnace blown through with steam or air before lighting the burners.

(2) Whenever a burner is extinguished, shut off the oil and blow through the furnace with steam or air before relighting the burner.

(3) Never attempt to relight a burner from a red-hot brick wall.

(4) Atomizer control cocks or valves must be kept tight at all times to prevent leakage of oil into the furnace from dead burners.

(5) The firemen handling the torch in lighting burners must stand well clear of the register to avoid injury in case a flareback occurs.

Procedure When Flareback Occurs. When a flareback occurs, close the master oil valve, shutting off all oil to the furnace. Speed up the blowers, if in operation, keep fireroom hatches closed, and stop the oil-service pumps.

Fires. In each fireroom fitted for oil burning there shall be provided fire-extinguishing apparatus, consisting of steam fire hose permanently coupled and of sufficient length to reach all parts of the fireroom, and either a box containing about two bushels of dry sand with a large scoop, or chemical fire extinguishers of the tank type.

Do not allow oil to accumulate in any place; particular care must be taken to guard against this accumulation in drip pans under pumps, in bilges, in the furnaces, and on the floor plates. Any oil spilled must be at once wiped up. Whether at sea or in port, bilges should be steamed out every watch and should be washed and pumped out at least once each day.

Although smoking is permitted on the fireroom floor plates in front of the furnaces by the Naval Instructions, in view of the possibility of carelessness on the part of the personnel, all smoking in the firerooms is generally prohibited.

Procedure in Case of Fire. In attempting to extinguish an oil fire never use water, as the only effect will be to spread the flames over a larger area. Fuel Oil is difficult to ignite in bulk and is not capable of spontaneous combustion. There is little danger of the oil in the tanks becoming ignited, but in such an event steam hose connections should be made to the tank and steam turned on to smother the flames.

In case of an oil fire in the bilges, close master oil valve, shutting off oil to the burners; stop the oil pump; connect up the foam fire extinguisher and steam hose and direct the hose streams on the fire to smother the flames. If the fire be extensive, and the personnel should be driven from the fireroom, all hatches, ventilators, or other openings into the compartment should be closed and steam should be turned into the compartment to smother the fire.

If the blowers are in operation, judgment must be used in deciding whether to keep them in operation or to immediately shut them down. They will feed air to the flames and thereby aid combustion, but conditions may occur when speeding up the blowers may be the one thing that will save the personnel by giving them an opportunity to escape. In general, the blowers should be secured as soon as possible.

Foamite Firefoam Method for Extinguishing Fire. In this method the fire is smothered by the application of **Firefoam**, a tough blanket made up of bubbles of carbon dioxide gas. This blanket will adhere to any material and float on the surface of gasoline, fuel oil, and all other inflammable liquids.

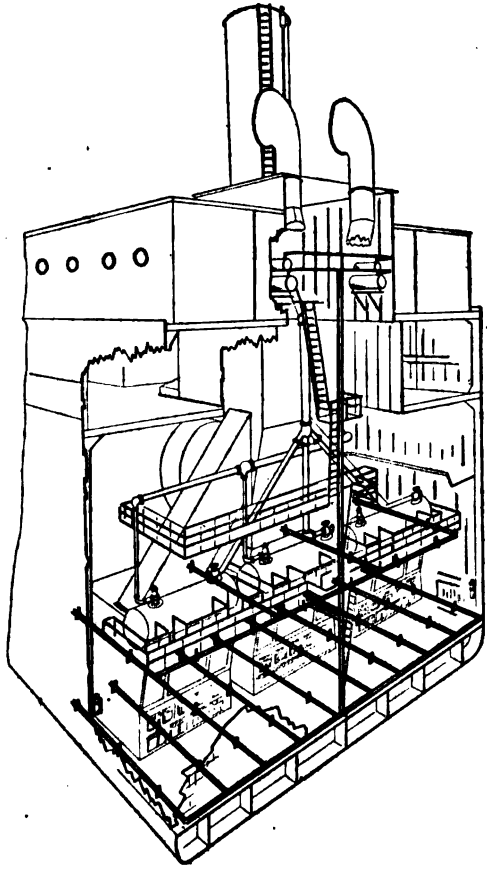


FIG. 45.—Firefoam Sprinkler System for Extinquishing Oil Fires.

The extinguishing medium, carbon dioxide, is held directly on the burning surface to the exclusion of all air and cannot be blown away by the wind or fire draft.

Firefoam is generated by the mixture of two solutions which react upon each other and expand to a volume from six to eight times the combined

volumes of the original solutions. The reaction evolves carbon dioxide, which, due to the presence of a foaming ingredient, Foamite, is not permitted to pass into the atmosphere but is held in the form of a blanket of foam and carried directly to the fire.

Application of Firefoam. For incipient fires the Firefoam can be applied by means of **hand apparatus** and **Chemical engines** which in outside appearance and operation resemble closely the ordinary types. In the protection of large oil bunkers of oil burning vessels and tank steamers the two solutions are pumped through a double pipe line which does not permit the two to mix until they reach the fire. For the protection of the boiler rooms of oil burning vessels, the Firefoam is applied by means of a Firefoam **Sprinkler System.**

Firefoam Sprinkler System. Fig. 45 shows a Firefoam sprinkler System for the protection of the boiler room of oil burning vessels. From the two **solution tanks** maintained under air pressure at all times, a **double pipe line** leads to **sprinkler heads** located throughout the boiler room and under the floor plates. At these heads the two solutions mix and Firefoam is discharged.

Operation. The system can be put into operation automatically by means of an auxiliary thermostatic system, or manually by means of controls located at a safe distance.

Firefoam Solutions. Solution "A" is made up by dissolving aluminum sulphate in water; Solution "B" by dissolving bicarbonate of soda and Foamite in water.

The percentage of concentration of the solutions varies with the different equipment in which the solutions are used. However, in every case, the determination is made from a computation of the relative weights of aluminum sulphate and bicarbonate of soda which will react chemically with each other. In the larger systems equal volumes of solutions are generally used; whereas, in the smaller equipment economy requires that the volume of the aluminum sulphate solution be reduced, a result which is accomplished by concentrating that solution.

Solution "A." The proper quantity of water should be measured out and after the temperature has been raised to approximately 150 deg. the aluminum sulphate should be added slowly—not dumped in—and the solution agitated for approximately five minutes.

Solution "B." The proper quantity of water should be measured out and after the temperature has been raised to approximately 100 deg. but never above 110 deg., the bicarbonate of soda should be added and thoroughly dissolved, after which the Foamite, a thick black liquid, is added. At any temperature above 110 deg. the bicarbonate of soda depreciates rapidly. For that reason every effort should be made to keep the water below that temperature.

Testing. The solutions should be tested for expansion approximately three times a year. Samples of the two solutions when mixed in equal volumes should give an expansion of from six to eight times the combined volumes of the two solutions which are used in the test.

The solutions, with proper care, should last from two to three years. In some cases they have retained their efficiency for more than five years, but the life of the solutions depends upon the care exercised in their preparation and maintenance. Such conditions as high temperature and excessive agitation only tend to shorten their life.

From a chemical analysis which should be made at least once a year can be determined what weight of the constituent chemicals should be added to restore the original efficiency of the solutions. There is a limit, however, beyond which no degree of addition is effective. In that case fresh solutions must be supplied.

Oil Leaks into Feed-water System. Fuel oil may leak into the feed-water system through a leak in the oil heaters or through a leak in the heating coils installed in the fuel-oil tanks. Such leaks can be detected by examina-

tion of the water drained from the steam side of the heater. Whenever leaks are found, the heaters must be by-passed; otherwise the oil will pass directly into the boilers through the feed-water system. The presence of oil in any boiler is dangerous, but it is particularly so in a small-tube boiler in which fuel oil is burned owing to the intense heat and the high rates of evaporation.

Damage to Brick Walls. Whenever a brick drops out in the furnace wall the boiler should be cut out, if practicable. If necessary to continue the boiler in operation until another can be connected up to replace it, burners adjacent to the location of the brick should be cut out to avoid damage to the boiler casing.

Testing Oil System. Whenever that part of the oil system subject to pressure has not been in use for a period of a week, or after joints in the piping have been remade, it shall be tested cold under a pressure at least equal to the working pressure, and a careful inspection shall be made for leaks before fires are lighted. All fuel-oil fittings shall at all times be kept in working order; and the air slides, doors, and valves shall be frequently moved when not in use to insure that they are in good condition and ready for use.

SECTION 4

MARINE BOILERS

BY

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CONTENTS

	Page
Development	333
Designation	339
Ideal Requirements	343
General Design	343
Effect on Design of	
Fuel	344
Draft	344
Steam Pressure	352
Steam Output	354
Size and Number of Units	357
Space and Weight	358
Boiler Sizes and Weights	358
Design of Scotch Boilers	385
Rules and Regulations Governing Boiler Construction	390
Factor of Safety	391
Hydraulic Tests	392
Riveted Joints	393
Calking	405
Welded Drums	406
Bumped or Dished Heads	406
Manholes and Handholes	408
Cylindrical Shells, Riveted	408
Flat Surface Stayed	410
Stays	413
Furnaces	415
Combustion Chamber Girders	420
Combustion Chamber Tube Sheet	421
Cylindrical Tube Sheets, Water Tube Boiler Shells	422
Diagonal Ligaments	422
Scotch Boiler Construction	426
Tubes	434
Boiler Setting	438
Boiler Mountings	440
Furnace Mountings	453
Superheaters	461
Care of Boilers	
Preservation of Idle Boilers	464
Prevention of Exterior Corrosion	464
Raising Steam	464
Removal of Oil	465
Internal Corrosion	466
Boiler Repairs	475
Fuel and Firing	477
Time Firing	478
Cleaning Fires	479
Air Required for Combustion	480
Analysis of Gases of Combustion	481
Removal of Scale	487
Boiler Tests	488
Boiler Materials, Inspection Requirements	514

MARINE BOILERS

BY

ERNEST H. PEABODY

Serious experimentation in the propulsion of vessels by steam became active in the closing years of the 18th century. Ingenious mechanisms driven by steam for operating oars and paddles were devised, soon followed by the use of steam-propelled paddle wheels. Considerable progress had been made on both sides of the Atlantic when Fulton brought out the first commercially successful steamboat, the "Clermont," which was given her initial trial on the Hudson River in 1807. Colonel John Stevens, of Hoboken, had already built several very practical boats; and, many years in advance of his time, had developed a steamboat operated by twin screws which had a successful trial on the Hudson in 1804. The remarkable genius of Stevens may be appreciated when it is considered that not only did he adopt the screw propeller and twin screws, but he employed a gearing between the engine and the propeller-shaft by which one engine operated both propellers. Furthermore, he developed a water-tube boiler to carry 50 lbs. gage pressure, an illustration of which is shown in Fig. 1. The boiler and propelling machinery were built in his own shops. He was already well advanced in the construction of a larger vessel, the "Phoenix," operated by paddle wheels, when the trial of the "Clermont," and the succeeding patent award to Fulton by the State of New York, giving him control of the waters of the Hudson, made it necessary for Stevens to take the "Phoenix" to Philadelphia. She thus achieved the honor of being the first steam vessel to make a voyage at sea.

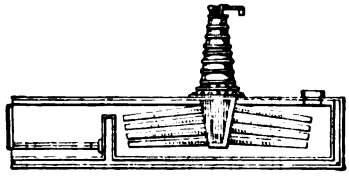

FIG. 1.—Stevens' Boiler.

Fulton's foresight in obtaining an engine and boiler designed by James Watt and built by Boulton & Watt, at Soho, Scotland (which he shipped to New York with the permission of the English government and installed in the "Clermont"), no doubt had much to do with his success in anticipating Stevens, the facilities for the fabrication of machinery being then much further advanced in Great Britain than in America.

The early experimentors in steam navigation naturally adopted the prevailing types of stationary boilers, among which was the vertical type called the "**haystack**" boiler from its resemblance in shape to a stack of hay, and the horisontal "**wagon**" boiler of Watt, named from its resemblance to the top of a wagon of the "prairie schooner" type. Smeaton conducted some interesting experiments on shore, at Long Benton, England, in 1772, with the "haystack" boiler fitted with a flue and carrying a pressure of 1½ lbs. above the atmosphere. The boiler of the "Clermont" was of copper, about "7 ft. deep and 8 ft. broad" by 20 ft. long, set in brickwork, and carried a pressure of 5 lbs. above atmosphere. It was provided with one square-section return flue running the entire length of the boiler, the grates being installed at the front, and the hot gases passing to the rear under the bottom of the shell and back through the flue to the funnel.

The natural development of the marine boiler was first along the line of modifications of the "wagon" boiler, with shells made square or with flat sides and rounded top, and with various arrangements of rectangular flues with various twists and turns, and also with flues of circular and elliptical section. A boiler of the flue type, carrying a pressure of 15 lbs. per sq. in., was installed in the S.S. "Sirius" in 1837. This vessel was one of the first to make a trans-Atlantic voyage entirely by steam power.

Within the following ten years the next development began to be introduced, consisting of the substitution of tubes for the rectangular and round sheet flues. Fire tubes had already been used, notably by Stephenson in the **locomotive boiler** of the "Rocket," in 1830, but the introduction of tubes in marine boilers had been slow, and it is interesting to note that so late as 1862 a lively controversy was under way as to the relative merits of the flue boiler versus the tubular boiler. We even find vessels constructed in the fifties fitted partly with flue boilers and partly with tubular boilers, the

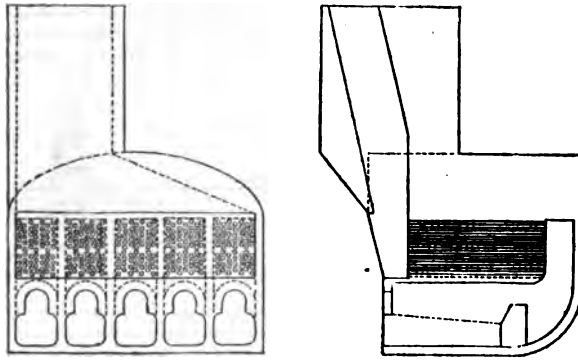

FIG. 2.—Stimer's Return-tubular Boiler.

designers evidently being reluctant to trust themselves entirely to the fire-tube construction. The same thing has been witnessed in our own day, where vessels have been fitted partly with Scotch boilers and partly with water-tube boilers, on the same reasoning.

The steam pressures carried in 1850 were surprisingly low. A list of sixty-four vessels (most of them sea-going) fitted with boilers designed and built in America shows that ten of them carried 10 to 12 lbs. gage pressure; twenty, 13 to 16 lbs.; five, 18 to 20 lbs.; six, 23 to 25 lbs.; seven, 28 to 30 lbs.; five, 32 to 35 lbs.; five, 36 to 40 lbs.; three, 45 lbs.; one, 50 lbs.; one, 80 lbs. and one, 90 lbs. Thus, more than half of these vessels carried pressures of 20 lbs. and under, while 53, or 84 per cent of the total number, carried steam pressures not over 35 lbs. Most of these boilers were of the circular-flue type. Some had shells of approximately square section, others oval; and a considerable number cylindrical, this being the case with all types carrying 30 lbs. and over. The return-tubular idea was well represented, the type shown in Fig. 2 being a typical example. This boiler was invented and patented by Alban C. Stimers, afterward Chief Engineer of the famous "Monitor" in the Civil War. The highest pressure of all (90 lbs.) was that of a vessel operating on Lake Erie,

whose boilers consisted of seven iron cylindrical shells, 30 in. in diameter by about 24 ft long, each fitted with two 12 in. diameter return flues, Fig. 3. These boilers were set on deck, the total heating surface being 3640 sq. ft. and the total grate surface 112 sq. ft. One of the anomalies of steam navigation is that this type of boiler has prevailed to the present day, and is now in

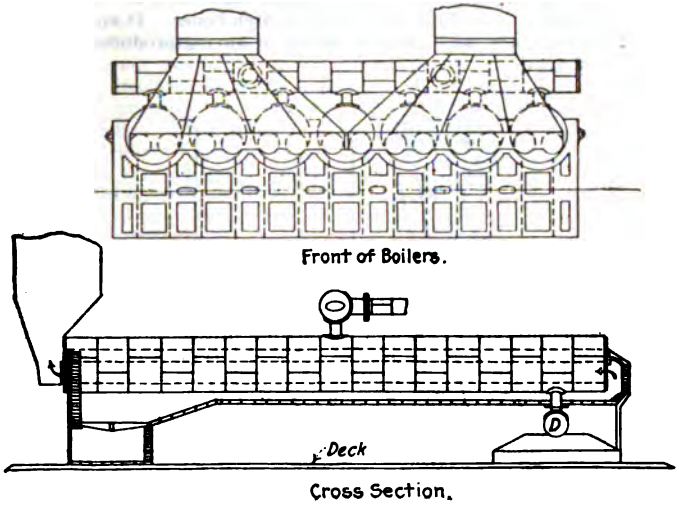

FIG. 3.—Early Inland River Type Boiler.

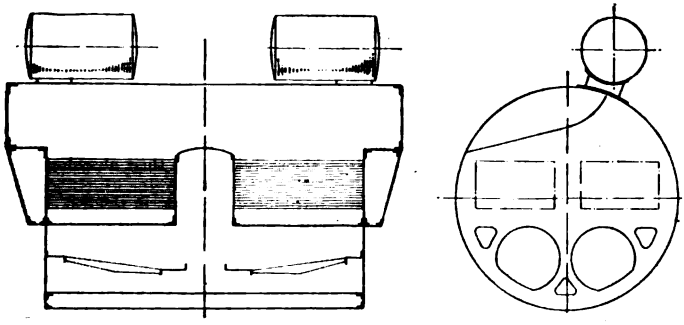

FIG. 4.—S. S. MacGregor Laird Boiler.

common use on inland rivers where muddy river water is used for feed and where the ease with which the boiler may be washed out every few days explains the common use of the design.

In 1862, the boiler shown in Fig. 4 was fitted in the S. S. "MacGregor Laird." This has been referred to by writers as the first example of the

Scotch boiler. However, previous to 1851, the boiler illustrated in Fig. 5 was designed by an American engineer, Mr. Erastus W. Smith, and constructed by Mott & Ayres, of New York, for the S.S. "Hermann." This boiler was fitted with a steam-dome through which the uptake passed (sometimes called a "**wet uptake**"), and the front tube-sheet was set back from the end of the boiler, both of which characteristics while not now in vogue were not uncommon in early designs of the Scotch boiler. It appears, therefore, that America may well claim the honor of having produced the first of this celebrated type.

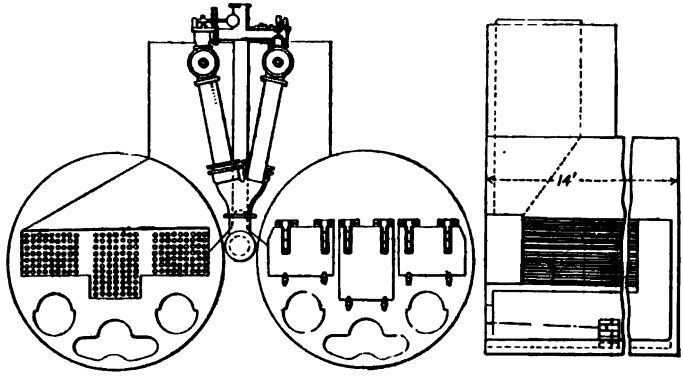

FIG. 5.—Smith's Boiler.

The compound engine, first successfully introduced by the progressive firm of Randolph & Elder (also builders of the "MacGregor Laird"), in the S.S. "Brandon," in 1854, had much to do with the gradual working up of steam pressures, as it was discovered that compounding was of little value unless accompanied by higher pressures than the then favorite 15 to 20 lbs. However, 35 or 40 lbs. was considered good enough, and the attention of marine engineers at this time (the late fifties) was taken up with the problem of improving the economies by increasing the temperature of the steam by superheating rather than by a material increase in pressures. The superheater continued to be used until the early seventies, with pressures up to 60 lbs., when difficulties with lubrication and trouble with the superheaters themselves led to their abandonment for the time.

In 1874, the triple-expansion engine was introduced by Kirk, in the S.S. "Propontis," and proved a still further incentive toward higher steam pressures. It is a commentary on the conservatism of the marine engineering of the day that as late as 1875 there were many advocates of the simple as opposed to the compound engine, and that serious consideration was even given to the project of designing the propelling machinery of war ships to operate at pressures of 15 lbs. abs. (atmos. press.), that is, depending on the vacuum, so that in the case of a boiler or steam pipe being perforated by a shot, the steam would not rush out and injure the personnel.

Within the period of twenty years from 1875 to 1895, the triple-expansion engine fully established its superiority over the compound type for sea-going vessels The type of boiler also became settled, the Scotch boiler, with its circular shell and convenient return-tube arrangement establishing a recognised superiority over the earlier flue construction. These two facts, accompanied by the introduction during the same period of open-hearth steel plate

for boiler manufacture, account for a very remarkable increase in steam pressures. In 1875, the prevailing practice was represented by 60 to 80 lbs. per sq. in. In 1895, it had risen to an average of 160 to 165 lbs. (required by the triple-expansion engine), with a few instances of pressures as high as 180 lbs. This rapid increase in working pressures is all the more remarkable in view of the slow development toward higher pressures in the preceding years.

Meantime, it is interesting to note the development of the water-tube boiler. The great flexibility of this type early led to the introduction of freak designs, inaccessible for cleaning and repair, and which, on account of high steam pressures and defective design in numerous instances led to disastrous results. A successful design by John Faron was, however, installed in the trans-Atlantic steamer "Atlantic," in 1849, followed by similar installations in the "Pacific," "Baltic," and "Arctic," Fig. 6. The full advantages of the water-tube principle were not realised in this boiler, as it did

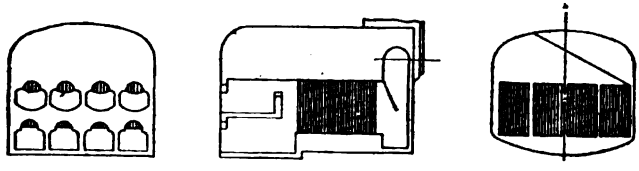

FIG. 6.—Faron's Boiler.

not eliminate the large semi-rectangular shell; and the design gave way to the superior features of the cylindrical boilers of the flue and fire-tube type. In 1851, Belleville began the interesting development of the boiler which bears his name. He was followed by Lagrafel, Niclausse, Du Temple, Thornycroft, Yarrow, Cowles and others, some of whom produced designs which have survived. In 1893, a water-tube coil boiler, designed by Charles Ward, was installed in the U.S.S. "Monterey," under the initiative of Admiral George W. Melville, who during his period of service as Engineer-in-Chief of the United States Navy from 1887 to 1903, took a prominent part in the famous "Battle of the Boilers" which led to the ultimate elimination of the shell type of boiler from all fighting ships and the substitution of water-tube boilers.

The first Babcock & Wilcox Marine Boiler was designed by Stephen Wilcox, and installed in the yacht "Reverie," in 1889. This design was improved and perfected by William D. Hoxie, who in 1895 brought out the "Alert" type (so called from the fact that it was first installed in the U.S.S. "Alert"), Fig. 7. This boiler, owing to its simple, effective design and accessibility, and the fact that it is constructed entirely of forged steel pressure parts, straight tubes and expanded joints, and its adaptability for working pressures of 250 to 300 lbs. and over, has had phenomenal success in naval vessels and in the merchant marine.

The World War, now closing, has resulted in a greatly extended use of water-tube boilers in cargo and merchant ships, the most successful of the various types being designed closely along the lines of the Hoxie patent. Many of these vessels have also, with marked success, been fitted with superheaters, the modern type of which was re-introduced into marine practice

by The Babcock & Wilcox Company in the Lake steamer "D. G. Kerr," in 1902, and in the U. S. Navy in the U.S.S. "Indiana," in 1904. Specially designed superheaters are also being extensively fitted in Scotch boilers, the objections which led to the elimination of the earlier types of superheaters having been entirely overcome.

The advantages of high pressure and superheated steam have long been recognized by marine engineers. Superheat, as noted, is again being adopted; and while to some extent quadruple-expansion engines led to the use of pres-

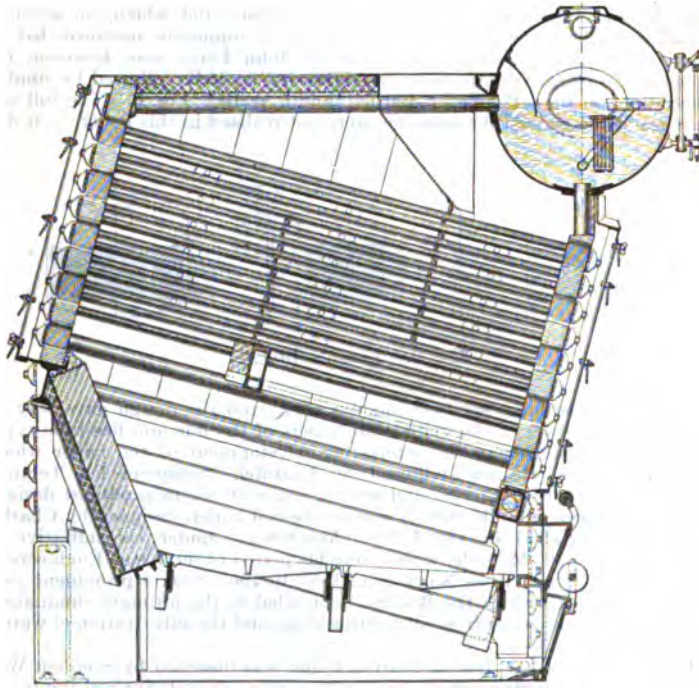

FIG. 7.—Babcock & Wilcox Boiler, "Alert Type."

sures of 250 lbs. with well-designed water-tube boilers, the increase in steam pressures has not been at all rapid since 1895. The common pressures to-day (1919) for new work may be considered as 190 lbs. for Scotch boilers and 200 to 210 lbs. for water-tube boilers. The introduction of steam turbines has been partly responsible for this slow increase in pressures, recent development having been more along the lines of improved vacuum rather than toward higher working pressures. It is probable, however, that necessity for increased economies and reduction in weights will again result in the upward climb of working pressures for which the water-tube boiler is exceedingly well adapted.

DESIGNATION OF BOILERS

Marine Boilers are designated in different ways, according to more or less self-evident characteristics.

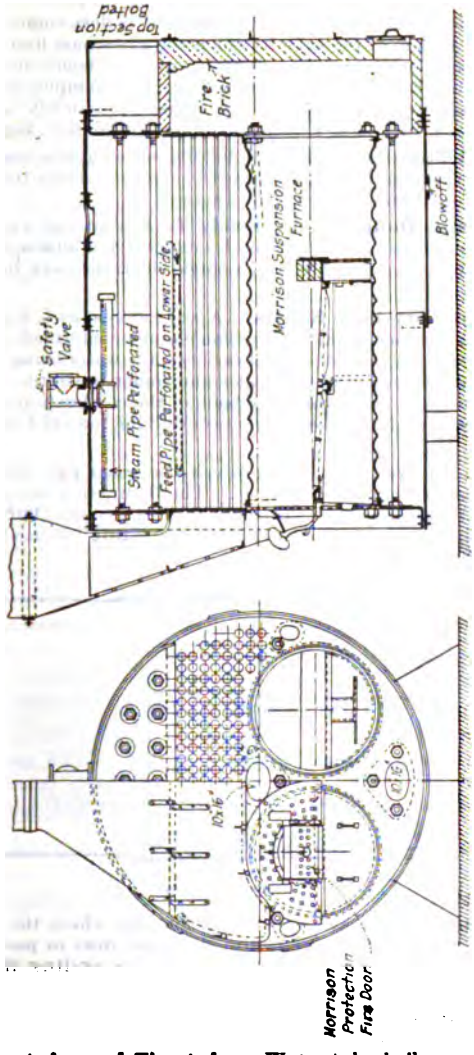

FIG. 8.—Dry-back Boiler.

1. Water-tube and Fire-tube. Water-tube boilers are those in which the water passes through the tubes and the hot gases circulate around them.

Fire-tube boilers are those in which just the opposite takes place, fire passing through the tubes which are immersed in water.

2. Externally Fired and Internally Fired. Externally fired are those in which the furnace, while within the boiler setting proper, is external to the heating surface of the boiler. Most water-tube boilers come under this class; also fire-tube boilers of the type in which the gases pass first under the shell and then through tubes or flues. Watt's "wagon" boiler and the horizontal return tubular (land type in brick setting) are examples of this construction. Internally fired boilers have the furnace practically surrounded by the heating surface, such as in the Scotch boiler, locomotive, leg type, etc.

3. Tubular and Tubulous Boilers. In this sense tubular signifies fire-tube boilers while tubulous is a term formerly given to water-tube boilers; the latter designation is now not much in vogue.

4. Single-end and Double-end. Boilers fired from one end or from both ends; the term usually refers to Scotch boilers, the double-ended boiler being practically two boilers combined in a single shell, the back heads being omitted.

5. Wet-back and Dry-back Boilers. A wet-back boiler is an internally fired boiler in which the back of the combustion chamber is formed by a water-leg or water space between two flat stayed surfaces, the water being comprised in the main circulatory system of the boiler and under pressure. The term dry-back designates the type in which the back of the combustion chamber is formed by fire-brick, usually backed by non-conducting material and a sheet-metal casing. (See Fig. 8.)

6. Flue Boiler. A fire-tube type in which the gases pass through rectangular- or circular- or oval-section flues. The distinction between a flue and a tube is usually fixed at about 5 in. diameter. The term "**tube**" is used when the diameter is 5 in. or less, and the term "**flue**" is used when the diameter is more than 5 in.

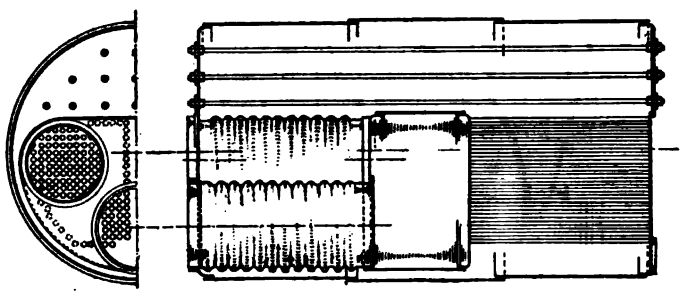

FIG. 9.—Gunboat Boiler.

7. Drop-flue Boiler.—An arrangement of flues by which the gases pass into the upper flues first and drop down to the lower rows in passing to the uptake. This type saves some headroom over the **ascending-flue boiler,** in which the gases enter the lower flues first.

8. Horizontal Return Tubular Boiler. The horizontal fire-tube type in which the gases pass to a combustion chamber in the rear and then return through tubes to the uptake at the front of the boiler. The term usually

refers to the externally fired dry-back boiler, set in brickwork, of a type similar to the "Mississippi River" boiler. This type is not much used in marine practice except on river steamers. The return-tube construction, however, is exemplified in the Scotch boiler.

9. Scotch Boiler.—This boiler receives its name on account of its popularity with shipbuilders on the Clyde. It is of the cylindrical-shell, internally fired type, fitted with corrugated furnaces, wet-back combustion chamber and return fire tubes. The cylindrical furnaces, which contain the grate-bars, bridge-wall, etc., are corrugated to give strength against collapsing pressure and to provide for longitudinal expansion and contraction.

10. Gunboat Boiler. The cylindrical fire-tube type in which the diameter is made relatively small to reduce headroom, the boiler being increased in length to give the necessary heating surface. The tubes, instead of returning from the combustion chamber to the front of the boiler, continue on to the back, where the uptake is situated. (See Fig. 9.)

11. Locomotive Boiler. The locomotive boiler as used in marine practice is an application of the standard locomotive type to river steamers and also to light-draft high-speed vessels. It has been entirely displaced in that service by water-tube boilers, but for a time was employed on account of its comparatively light weight. This type met with considerable success in some of the early British Destroyers.

12. Leg Boiler. The internally fired boiler in which the furnace walls are constructed of flat stayed-surface water-legs. This application is usually made in connection with cylindrical shells fitted with fire tubes, as in the case of the locomotive boiler, which is an example of the type.

13. Shell Boiler. The fire-tube type of boiler in which a large cylindrical shell is used under pressure, as, for example, in the Scotch boiler, as distinguished from boilers of the water-tube type, which use small-diameter steam-and-water drums.

14. Fire-box Boiler. A boiler in which the furnace is located in a so-called "**box**" of flat surfaces which form the furnace walls. It is similar to the Locomotive and Leg boilers except that it has return tubes.

15. Lobsterback Boiler, Fig. 10, has a cylindrical shell; box-shaped furnaces with water-legs at one end of the shell; flues leading to the back connection and return flues or tubes to the funnel, which is located about midlength of the boiler. This type is adapted to go under deck in shallow river boats.

16. Express Boiler. A term used to designate very lightly constructed water-tube boilers fitted with tubes 1 in. to 1¼ in. in diameter, which may be highly forced. This is the common type for Destroyers and very high-speed vessels.

17. Coil Boiler. A boiler of the water-tube type, in which the tube is in the form of a coil, usually in the form of a portion of a helix.

18. Porcupine Boiler. A vertical shell from which water-tubes of the Field type extend on all sides, like the quills of a porcupine, and around which the gases pass as they ascend to the uptake. The Field tube, invented by Field in 1866, consists of a tube having one end open to the circulation and the other closed, and another tube of smaller diameter inside, so arranged as to deliver water to a point near the closed end.

19. Flash Boiler. A boiler containing little or no excess water, and having no water line, and in which the feed water is injected in amounts

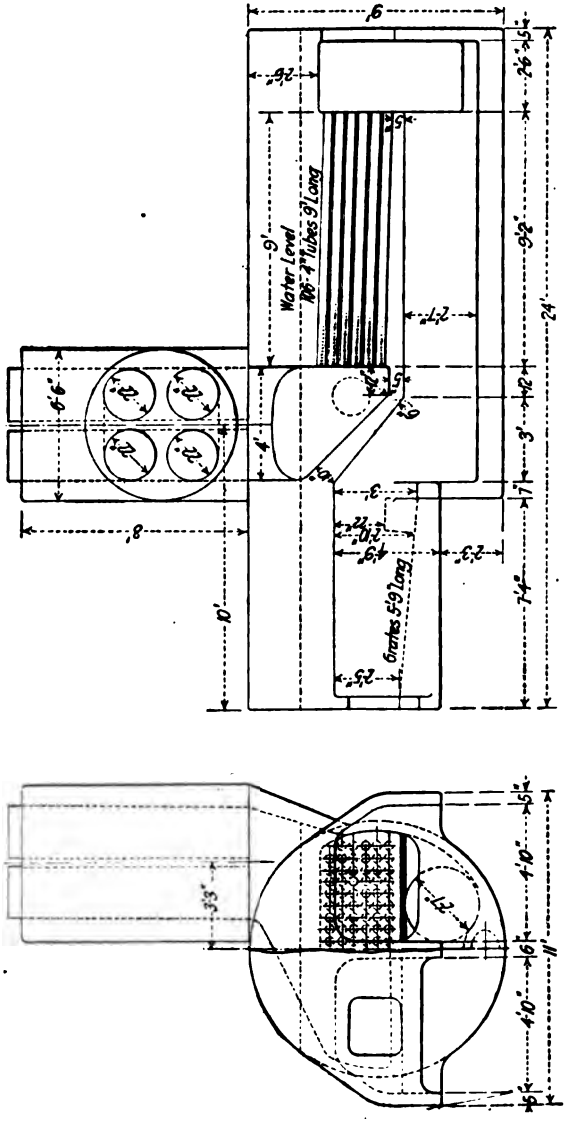

FIG. 10.—Lobsterback Boiler.

substantially equivalent to the demand for steam. This boiler takes its name from the fact that the water flashes into steam almost immediately on entering the boiler. The type is attractive for marine purposes on account of the reduction in weight due to the small amount of water carried. It has so far, however, not been successful in any but very small units.

20. Sectional Boiler. One in which the water spaces are divided into small parts, none of which is subject to disastrous explosion. This is an essential feature of "**Safety**" boilers. The term "Sectional" is sometimes used to designate the vertical divisions of the tube-bank of certain classes of water-tube boilers, but this point has been adjudicated, and the meaning held to be that of the broader definition given above. All water-tube boilers are therefore sectional in character.

21. "Pipe" Boilers. Those made of pipe through which the water circulates. Frequently used to designate all characters of water-tube boilers.

The infinite variety in which water-tubes can be grouped, bent and interlaced, renders the term "water-tube" altogether too broad to be used to designate a type. A water-tube boiler of robust structure, built of forged-steel pressure parts, straight tubes and expanded joints, is much nearer in its salient features to the Scotch-boiler class than to the type in which small bent tubes or screwed pipes are crowded into small space with little furnace volume, no combustion chamber and inadequate baffling. The latter type is inaccessible and uneconomical and is foredoomed to failure in severe service, while the former is able to meet the most strenuous demands and is safe, economical, and approximately half the weight of the Scotch design. This type of water-tube boiler is destined to supplant the excessively heavy, unsafe and expensive Scotch boiler in the merchant marine as it has already done in naval service.

IDEAL REQUIREMENTS FOR MARINE BOILERS

Ideal requirements for marine boilers may be summarized as follows: ability to carry high pressures without danger of disastrous explosion; accessibility for readily cleaning and overhauling all parts; lightness and compactness so as to "stow" well on shipboard; efficiency and economy in fuel consumption; simplicity of design and avoidance of special features which complicate repairs and necessitate a large stock of spare parts.

The late *Rear Admiral George W. Melville* stated that it was his conclusion that the thoroughly satisfactory water-tube boiler should possess, among others, the following characteristics: Reasonable lightness, with scantlings sufficient to promise reasonable longevity; an adequate amount of water, so that failure of the feed supply or any inattention thereto would not immediately cause trouble; accessibility for cleaning and repairs on both water and fire sides; straight tubes, with no screw joints in the fire but the simple expanded joints so well tested out for years; no cast metal, either iron or steel, subjected to pressure; ability to raise steam quickly; high economy of evaporation; economy of space; interchangeability of parts, and, as far as possible, the use of regular commercial sizes, so that repair material could be procured anywhere; the ability to stand severe forcing without injury; the ability to stand abuse; that is, to be of rugged construction and not so delicate as to require skilled mechanics to run it; safety against disastrous explosion, meaning that only the part of the boiler which gave way would be damaged.

GENERAL DESIGN

The factors which determine the principal features **of boiler proportions and design** are (1) kind of fuel, (2) draft, (3) steam pressure, (4) steam output, (5) size and number of units, and (6) available space and weight.

1. Kind of Fuel, Effect on Design. The first requirement in generating steam is to properly consume a sufficient amount of fuel. There is no use in adding heating surface if the necessary heat cannot be generated in the furnace. This simple axiom is oddly enough often overlooked. The boiler furnace is therefore the starting point in any calculations on the production of power by steam, and obviously the kind of fuel qualifies the furnace design. The character of the fuel further has an influence on the proper arrangement of the heating surface, and particularly on the baffling of the gases and areas of the gas passages, those fuels which can be burned with smaller amounts of air for combustion requiring smaller gas passages and smaller uptake areas. Volatile fuels such as oil or volatile coals require larger furnaces. If smoke and soot and ashes are produced in combustion, the necessary cleaning facilities will be required.

2. Draft, Effect on Design. Whatever the future may hold, atmospheric air is now the universal means of supplying the necessary oxygen for combustion. The rate at which this air is supplied depends on the intensity of the draft. **Natural draft,** so called, is the suction produced by the funnel, and the **intensity of the draft** depends mainly on two things, (1) the temperature of the escaping gases and (2) the height of the funnel. The gases must leave the boiler at a temperature not lower than the temperature of the steam. They should not exceed this in a well-designed boiler by more than 100 deg. F. to 200 deg. F., depending on the rate of driving. As the temperature of steam at 200 lbs. pressure is 388 deg. F., this would make the proper uptake temperature approximately 500 deg. F. to 600 deg. F. They frequently, however, reach a temperature of 800 deg. F. to 1000 deg. F. in some types, a condition which is promoted by neglect in cleaning the boiler, leaky baffles and insufficient or inefficient heating surface, etc. If the waste gases are increased from a temperature of 500 deg. F. to a temperature of 800 deg. F. the effective draft at the base of a 100-ft. funnel will be increased from approximately 0.47 to about 0.63 in. of water.

The **influence of the wind** is an important factor in creating draft, this effect being practically independent of gas temperature and funnel height. A good **head wind** will increase the draft a tenth or even two-tenths of an inch of water column. The effect of the wind becomes negligible when the wind is **"following"** the ship and blowing at the same rate of speed; and as this condition may be frequently met at sea, the designer must provide for this condition, any gain due to favorable winds being merely that much margin in securing good steaming conditions.

Draft intensity is due to the unbalanced pressure of the weight of the heated column of gas within the funnel and the weight of a column of cool air of the same height outside the funnel. Thus, the temperature of the outside air is another factor in draft intensity, cold air giving a higher draft than warm air. This, however, may be ignored in the design, as the change is slight and is partly balanced by the check to combustion due to the chilling effect of the cold air on the furnace temperature. Barometric pressure is still another factor in creating draft intensity, a low barometer giving less draft than a high pressure. At sea level this difference is negligible.

If the various data are available, the **draft** that will be obtained **theoretically** may be determined by the **Peabody Draft Formula** which was devised by the writer as a means of simplifying the calculation:

$$D = \frac{HB}{100}\left(\frac{2250}{t+460} - \frac{2660}{T+460}\right), \text{ where}$$

D = Draft in hundredths of an inch water column;
H = Height of funnel in feet above point of draft measurement;
B = Barometer reading in inches of mercury;
t = Temperature of outside air, deg. F.;
T = Temperature of gases inside funnel, deg. F.

The effect of friction, gas eddies, the cooling effect of air leakage, etc., combine to reduce the intensity of the draft obtained in practice, so that the theoretical results must be multiplied by a variable coefficient (depending upon the conditions) to obtain the actual draft at the base of the funnel. A coefficient of 0.8 represents fairly average practice.

The following handy rule will be nearly enough correct to serve in the large majority of cases: **The actual draft intensity expressed in hundredths of an inch of water column at the base of funnel or in the uptake at the boiler damper will be approximately equal to one-half the vertical distance in feet from the point of measurement to the top of funnel.**

When the draft intensity is increased by means of fans or blowers it is called **Forced Draft.** It is evident from the above draft formula that any desired draft intensity may be obtained if the funnel can be made high enough. Thus, a funnel, say, 300 feet high would give a draft, at its base, of approximately 1¼ inches of water column. Such dimensions are of course prohibitive on shipboard, and blowers are resorted to to give these higher drafts. There is, however, no real physical difference between forced and natural draft, the distinction being limited to the means of production.

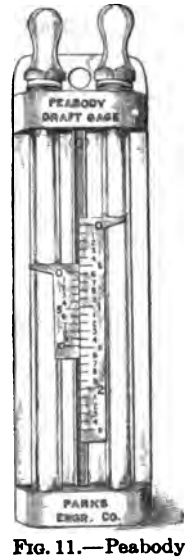

FIG. 11.—Peabody Draft Gage.

Considering draft intensity as a difference in gas pressure, there is no difference in the flow of gases from a higher to a lower pressure whether this difference in pressure be produced naturally by a funnel or artificially by fans. In other words, there is no real distinction between a push and a pull, except in cases where the forced draft is applied to the ashpit only (as in the case of ashpit draft) instead of the entire setting, as in the case of closed fireroom, etc. In that event, the pressure in the ashpit being higher than the pressure around the boiler casing, any slight leakage in the latter affects the course of the gases.

In view of the fact that a funnel must always be provided to take away the waste gases, the forced-draft installation is in reality a combination of forced and natural draft, and it is the sum total of the draft pressures (that is, the algebraic difference) that is effective in supplying air to the fuel.

In doing this certain resistances must be overcome, such as friction over the heating surface and through the fuel bed, or, in the case of oil fuel, through the restricted openings into the furnace. It is the custom of some engineers to base calculations, on the **draft measured in the boiler furnace,** or the difference between the furnace pressure and the pressure outside the furnace. The objection to this lies in the fact that the draft intensity as measured by a draft gage is greatly affected by the resistance the air meets in getting into the furnace. For example, in the case of a coal fire, if the fuel bed is thin and even and free from ash and clinker, the rate of combustion is at its maximum while the apparent *furnace* draft as measured by gage is at its minimum. As the fire bed builds up and becomes choked with clinker, the combustion rate falls off, but the furnace draft increases.

The same thing is true of the **draft as measured at the uptake or boiler damper,** but at this point the changes are less marked for the reason that the friction through

the boiler is added to that of the friction of the gases entering the furnace and the percentage of change with varying fire bed conditions is greatly reduced. While the friction of the gases through a boiler setting varies somewhat with the rate of forcing, design and type of the boiler and the proportion of heating surface to grate surface, these changes are less important than the changed resistance of the fire bed and its erratic effect on the draft gage. For these reasons, the writer believes that it is more practical to base calculations on draft difference between the *uptake* and the fireroom rather than between the fireroom and the furnace. The fireroom pressure is used instead of the ashpit pressure, as at high rates of forcing there is a slight drop in pressure in the passage of the air through the ash-doors.

The following results were obtained in a large number of tests, trial trips, etc., with Babcock & Wilcox Marine Boilers, with semi-bituminous mine-run coal, hand fired:

Draft Required at Different Rates of Combustion

Coal burned per square foot of grate per hour, pounds	Draft required. Difference between boiler damper and fireroom, inches	Coal burned per square foot of grate per hour, pounds	Draft required. Difference between boiler damper and fireroom, inches
7.5	0.10	27.5	0.82
10.0	0.15	30.0	0.95
12.5	0.21	5.0	1.28
15.0	0.28	40.0	1.65
17.5	0.37	45.0	2.07
20.0	0.45	50.0	2.53
22.5	0.55	60.0	3.52
25.0	0.68		

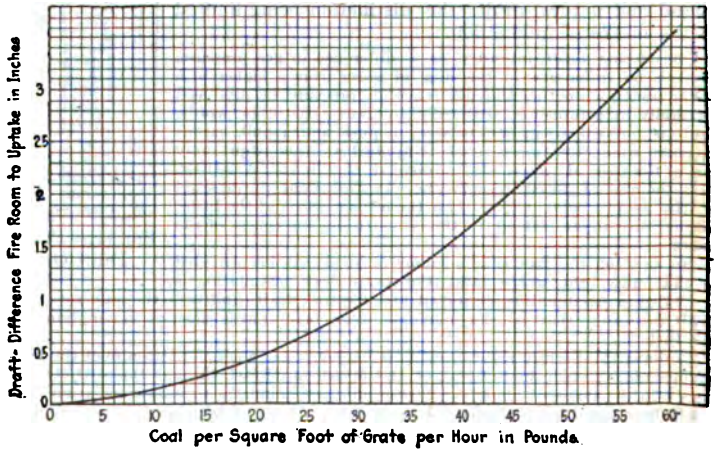

FIG. 12.

These figures are represented graphically in Fig. 12. It is to be noted that this chart may be used for forced draft as well as natural draft, as the ordinates indicate the algebraic difference between draft pressures in the uptake at the outlet from the boiler and the fireroom.

Forced draft for increasing the rate of combustion may be variously applied. In the **Closed-ashpit arrangement**, the blowers force air under pressure to the ashpit, which is made as far as practicable tight against the passage of air except through the furnace grates and the fire bed, thus accelerating the combustion rate. In this combination the relation of the draft pressure to that of the atmosphere becomes a vital factor. The natural draft or suction is a minus pressure (*i.e.*, below the atmospheric pressure), while the pressure in the ashpit is a positive pressure (*i.e.*, greater than the atmosphere). The pressure around the boiler setting remains the same as the atmosphere. The neutral point in the passage of the gases at which the pressure changes from plus to minus (*i.e.*, becomes equal to the atmosphere) depends upon the amount of the natural and forced draft, and also on the resistance of the fire bed and the boiler. Thus, with a strong natural draft and a low ashpit pressure, this neutral point may be in the fire bed itself, and the furnace pressure will be minus. Under these conditions, when the fire-doors are opened, air flows into the furnace. , But if the ashpit pressure is increased or the suction lessened, the neutral point may pass on into the boiler passages, or even into the uptake, in which case there will be a positive pressure in the furnace. Then when the fire-door is opened, the hot gases will blow out into the fireroom and this may reach a point where the fires cannot be handled properly. Under these conditions, the "blast" in the ashpit must be shut off or reduced temporarily while the fires are being charged or sliced, and turned on again when the fire-doors are closed. Automatic devices for shutting off the forced draft when the fire-doors are opened are frequently employed with ashpit draft.

An ingenious method of reducing the tendency for hot gases to blow out the fire-door is to admit a portion of the air directly to the furnace over the fire bed through jets around the fire-door opening. These jets discharging toward the rear, create a flow of air inward through the fire-door, thus opposing the hot gases that tend to flow outward. This device is effective to a limited degree in enabling higher ashpit pressures to be carried than would otherwise be possible.

Under usual conditions on shipboard, where ashpit draft is used, it is found that about 0.5 in. or 0.75 in. in the ashpit can be carried without interference with the firing. Above this pressure, the blast must be closed down each time the doors are opened. High forced draft on the closed-ashpit system has the objection that any slight gas leaks in the furnace or boiler setting will permit hot gases to escape outward under the effect of the plus gas pressure. This increases the wear and tear on the setting and causes added heat and discomfort in the fireroom.

The **Balanced-draft** system is a method of proportioning the ashpit and suction draft pressure to give a negative or a very slight positive pressure in the furnace at all times. There is a patented device for doing this automatically, and it is claimed that an improvement in economy and longer life to the setting is secured, as the furnace temperature is not lowered by the inrush of cold air on opening the fire-doors and the temperature of the casing and setting remains at a constant and moderate temperature. The rate of combustion is limited with this system, and it has not been extensively employed in marine work.

Forced draft on the **Closed-fireroom** system consists of making the entire fireroom and stokehold tight against air leakage, and forcing air into this space under pressure by means of **blowers**, usually located outside the compartment. **Deflectors** are frequently employed at the blower discharge for preventing the rush of air into the fireroom, and diffusing it. This is particularly necessary when the room pressure is high, as in the case of naval

work with oil fuel, where pressures of 6 in. to 8 in. are employed. In this case, the **loss of head due to velocity** of the air is an important feature and demands more study in the arrangement and form of air ducts than the problem usually receives. Closed-fireroom forced draft has all the advantages of natural draft, barring the inconvenience due to the necessity of providing all entrance and exit doors with air locks, *i.e.*, vestibules with double doors. The **rates of combustion** which may be obtained are, however, limited only by the ability to handle the fires, and there is no blowing out of hot gas from the fire-doors or setting, all the flow being inward, as the higher pressure is always in the fireroom. **Tight boiler casings** are a necessity, however, and there is some loss due to cold air entering the furnace door when open.

Induced draft is the system of forced draft which consists of the application of large blowers designed to draw the hot gases out of the uptakes and discharge them into the funnel, a damper being employed in the uptake to make certain that all the gases flow through the fans. This is equivalent in effect to high natural draft. The blowers are required in this case to handle hot gases, but for moderate rates of forcing the scheme is effective.

A system of induced draft termed the **Prat Ejector Draft** System has also been devised whereby a jet of cold air is discharged upward in the centre of the funnel for the purpose of accelerating the draft suction. This is a revival of an old idea, and while effective for low rates of driving, it has not been widely used.

Steam Jets in the funnel or uptakes readily increase the draft, but unless properly designed they are very wasteful of steam, and in any event the loss of fresh water renders this method of forced draft prohibitive except for harbor vessels. (See Fig. 13.)

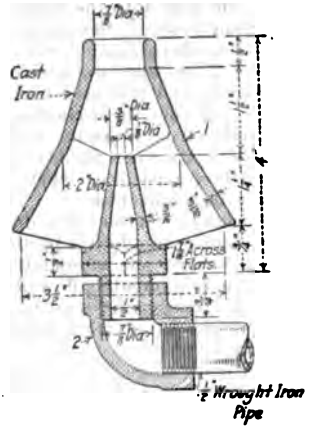

FIG. 13.—Steam Jet.

Steam jets may also be used for forcing air into closed ashpits, thus creating a positive pressure instead of a suction. This arrangement will never be used in marine work except as an emergency measure. However, it has one particular feature worthy of mention: Very poor fuels which run high in ash and tend to make a bad clinker may be burned satisfactorily with ashpit steam jets, as the steam in passing through the fuel bed softens the clinker and renders it porous to the passage of air. Thus fuels may be used which could not be employed with ordinary draft or which if so employed would give only a very low rate of combustion.

Ventilators. Although of utmost importance, the necessity of using ventilators of **adequate size** is often overlooked. Obviously, the air must be delivered to the fireroom before it can get to the furnace, or, if forced-draft fans are used to drive the air into the boiler room, the air must be delivered to the fans through ducts of adequate size. If the fans have to suck the air through restricted or circuitous passages, or if the natural funnel-draft is called upon to assist the air at high velocity down small ventilators, very decided losses in capacity result; losses which could easily be prevented. This is especially true of natural-draft installations. Ventilators **should be located** with cowls on deck in *unobstructed* places so that they will get the air freely, and they should be carried down into the firerooms with as *few*

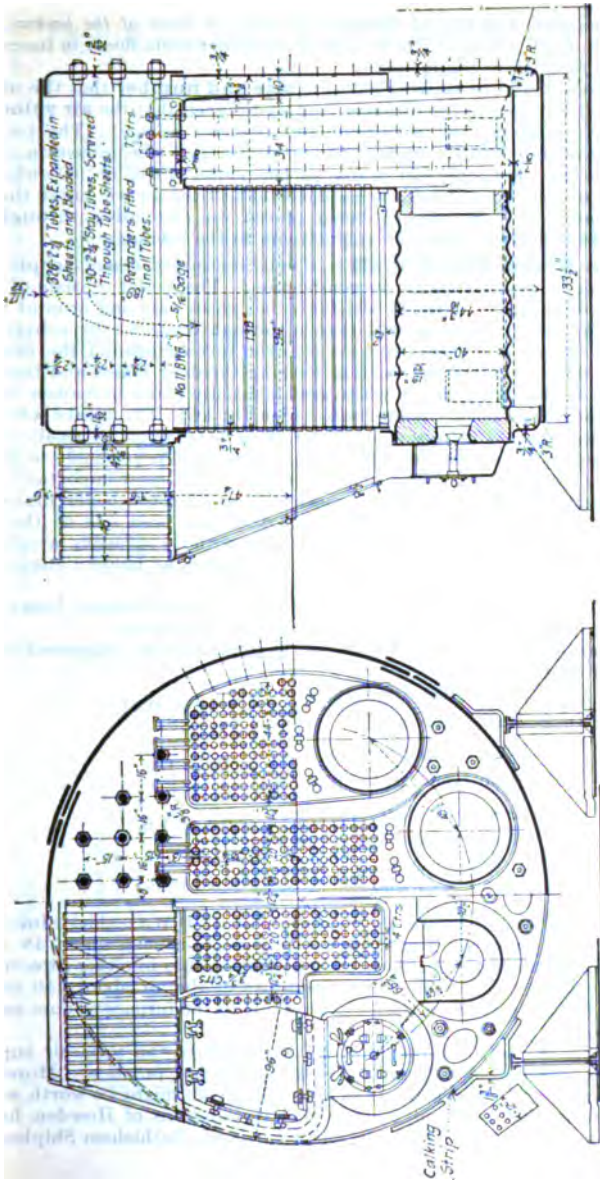
306-2½" Tubes, Expanded in Shells and Headed
130-2¾" Stay Tubes, Screwed Through Tube Sheets
68 Retarders fitted in all Tubes
No 11 B.W.G.
Caking Strip

Fig. 14.—Scotch Boiler with Howden Heater.

bends as possible, and should *discharge directly in front of the boilers*, and *low enough down* (not over 7 or 8 ft. from the boiler-room floor) to insure the air getting directly to the furnaces.

Ventilators should be so arranged as to **size and number** that the air will be evenly distributed to the furnaces, and so designed that the **air velocities** down the ventilators *shall not exceed* 1000 *feet per minute*. The total air should be calculated on the basis of the maximum rate of combustion, allowing not less than 300 cubic feet of free air per pound of coal or 260 cubic feet per pound of oil; and it should be assumed that all the air will enter through the ventilators, no dependence being placed on circulation through the fidley or through doors from the engine room to the fireroom.

Howden Heated Forced Draft is a combination of closed ashpit draft and a heater for heating the air for combustion. The heater consists of tubes located in the uptakes, through which the hot gases pass and around which the air is forced by blowers, and thence through ducts to the ashpits. A portion of the air is diverted to pass into the furnace around the fire-door opening; that is, through the castings forming the door jambs and lintels.

Howden heated forced draft when used is usually fitted to Scotch boilers, to which the design readily lends itself. (See Fig. 14.) The uptake where the heater is located is at the front of the boiler, from which ducts are easily carried to the ashpit and the latter being inside the boiler shell, is not liable to air and gas leakage (the main objection to ashpit draft). But most important of all, the high temperature of the waste gases leaving a Scotch boiler makes the air-heater very effective. Thus, Howden draft has been one of the most influential factors in keeping the Scotch boiler, with its excessive weight and high cost, within the range of fuel efficiencies required by modern engineering practice.

Mr. Charles F. Bailey, Engineering Director of the Newport News Shipbuilding and Dry Dock Company, states in this connection:

"The proportions of forced-draft air heaters may be expressed in the following ratios:

$$\frac{\text{Air heating surface}}{\text{Main boiler heating surface}} = 0.26 \text{ to } 0.31.$$

$$\frac{\text{Air heating surface}}{\text{Main boiler grate area}} = 10 \text{ to } 12.$$

$$\frac{\text{Area through heater tubes}}{\text{Area through boiler tubes}} = 0.8 \text{ to } 1.0.$$

$$\frac{\text{Area through heater tubes}}{\text{Grate area}} = 0.15 \text{ to } 0.19.$$

"We use tubes 3-inch outside diameter, No. 12 B.W.G. thick almost exclusively. The tubes are placed vertically and are usually about 48 inches long, the variation being not more than about 6 inches in either direction.

"The heaters usually raise the temperature of the air about 150 deg. to 175 deg., the temperature of the air as supplied to the furnace seldom exceeding 250 deg.

"The ratio of the air supplied above the grate to the total air supplied varies considerably, but is from 0.15 to 0.2 in our best practice. More careful consideration of this feature than is usually given might be worth while."

The data in Tables 1 and 2 on actual installations of Howden heaters are given by Mr. Robert Warriner, Chief Engineer, Bethlehem Shipbuilding Corporation.

Table 1. **Howden Draft System Data**

Plant	U.I.W.	Harlan	S.P.	Harlan	Harlan	From a Howden print	Fore river
Hull number	147	117	4.177	3,450	3,459	....	1,376
Air heating surface, square feet	706	864	765	714	665	600	575
Boiler heating surface, square feet	2,710	2,755	2,750	2,376	2,412	2,277	2,475
Grate area, square feet	55.0	78.0	55.0	57.8	56.4	56.4	55.0
I.H.P. (total)	2,600	2,000	2,600	2,700	2,500	2,440	2,600
A.H.S. / B.H.S.	0.261	0.314	0.278	0.301	0.275	0.264	0.232
B.H.S. / G.A.	49.2	35.3	50.0	41.1	42.7	40.3	45.0
Air heating tube area, square feet	10.48	10.42	9.17	8.62	9.59	9.15	9.88
Boiler tube area, square feet	16.50	13.76	13.49	10.24	10.38	9.7	9.81
A.H.T.A. / B.T.A.	0.635	0.758	0.680	0.842	0.923	0.942	1.006
G.A. / A.H.T.A.	5.25	7.49	6.00	6.70	5.88	6.16	5.57
A.H.S. / G.A.	12.83	11.07	13.90	12.35	11.79	10.65	10.45
A.H.S. / I.H.P.	0.813	0.864	0.884	0.794	0.799	7.38	0.664
B.H.S. / I.H.P.	3.13	4.13	3.17	2.64	2.89	2.80	2.86
A.H.S. + B.H.S. / I.H.P.	3.94	5.43	4.06	3.43	3.69	3.54	3.52
Width of air heater box	38"	31⅝"	31⅝"	29⅝"	33⅝"	31⅝"	34⅜"
Length of air heater tubes	36"	48"	48"	48"	42"	40"	41"
Size of air heater tubes	2½"–#12	2¾"–#12	2¾"–#12	2¾"–#12	2¾"–#14	2¾"–#14	3¼"–#12
Number of air heater tubes	372	300	264	248	264	250	198
Size of boiler tubes	3"	3"	3"	2½"	2¾"	2½"	2½"
Diameter of furnace	3'–4⅜"	4'–3"	3'–4⅜"	3'–4"	3'–4"	3'–4"	3'–4"
Number of boilers	3	2	3	3	3	3	3
Diameter of stack, inside	7'–6"	5'–6"	6'–9"	6'–8"	6'–3"	8'–6"	6'–6"
Grate area / Stack area	3.74	6.56	4.61	4.97	5.52	3.72	4.98

Table 2. Air Velocities in Howden Heater System

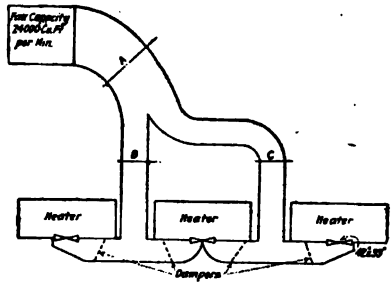

FIG. 15.—Air Ducts for Howden Heater.

I.H.P. 2600,
Oil per I.H.P. = 1.00 lb. per hr.
Cu. ft. of air per lb. of oil = 310 at 60 deg. F.

$$\text{Cu. ft. of air required per min.} = \frac{2600 \times 1.00 \times 310}{60} = 13{,}420$$

Temp. of engine room, 95 deg. F.
Temp. of air in ducts, 100 deg. F. } Assumed.
Temp. of air in heater, 250 deg. F.

Location	Size	Area, sq. ft.	Approx. temp., deg. F.	Cu. ft. per min.	Velocity, ft. per min.
A	33" ×43⅜"	9.94	100	14,450	1,453
B	16½" ×43⅜"	4.97	100	7,225	1,453
C	16½" ×43⅜"	4.97	100	7,225	1,453
Entrance to heaters	42" ×33"	9.62	100	4,817	501
Heater space	12'-9" ×2'-10¾"	25.55net	250	6,110	239
Exit from heaters	2" ×13" ×34¾"	6.27	250	6,110	974

Ellis & Eaves Forced Draft is an application of induced draft in combination with heaters in the uptakes for heating the air for combustion. In this case the fans handle the waste gases, partially cooled by the air heater. (See Fig. 16.)

A combination of closed fireroom forced draft with air heaters has also been patented, but has found little application in practice for the reason that the best types of water-tube boilers, which are the types employed at high capacity under closed-fireroom conditions, have such high heat absorbing qualities that the uptake temperatures are low, and thus very large air heaters are required to obtain a fair degree of air heating, and even then the financial return due to the saving of waste heat is insufficient to pay a satisfactory return on first cost and upkeep expense. (See Tests, Cincinnati Boiler, p. 505.)

3. Steam Pressure, Effect on Design. In the large majority of cases the **working pressure required at the engine** fixes the boiler pressure, due allowance being made for frictional drop in steam lines including loss through bends and fittings. With steam velocities of 5000 to 8000 feet per minute, this pressure drop will vary from 5 to 15 lbs. in the smaller installations with short piping connections, and from 20 to 25 lbs. in larger powered and highly forced equipment consisting of a multiplicity of boiler units in several compartments. **If superheaters are fitted,** an additional drop of a few pounds pressure must be allowed for.

In exceptional cases boilers built for high steam pressure with small steam-and-water spaces have been successfully employed by operating main engines through reducing valves, the boiler pressure being considerably higher than necessary to overcome ordinary friction in pipes. This results in a small amount of superheat due to wire-drawing, and provides the equivalent of a reservoir of steam to meet fluctuating loads.

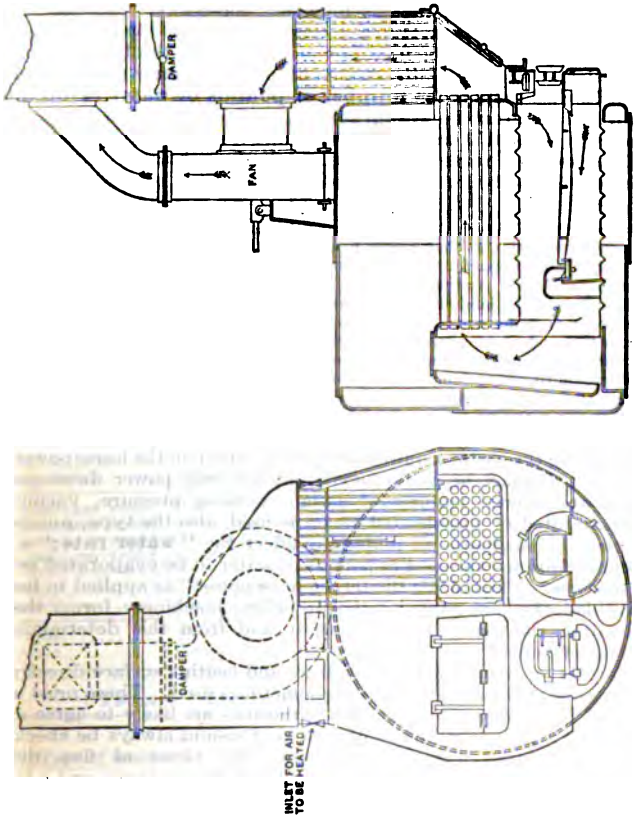

FIG. 16.—Ellis and Eaves System.

The pressures for which **water-tube boilers** may be constructed without departing from good practice are very high. War vessels frequently are fitted with the better type of this generator carrying more than 300 lbs., while in special cases, 500 lbs. has been employed. One thousand pounds would not require any insurmountable modifications in design. **Scotch boilers**, on the other hand, reach the limit of proper construction at about 200 lbs. Above that either the plates become enormously thick for boiler work or the diameter of the shell must be greatly reduced; and weights (already excessive) become prohibitive. Advanced designers have indeed called

for Scotch boilers to carry 235 and even 250 lbs.; but this increase in steam pressures has brought with it not only unjustifiable weights, but excessive cost of maintenance.

4. Steam Output, Effect on Design. The boilers are required to furnish the necessary steam to operate the propelling and auxiliary machinery at full capacity. "Horsepower," "indicated horsepower" or "shaft horsepower" are the properly accepted terms for defining the capacity of the engine. They should not be used in connection with the boiler except as a basis for calculating the required steam output. **The only proper measure** of this in English units is the **number of pounds of water (avoirdupois) per hour which must be evaporated** to produce the steam to operate the machinery, at the actual working pressure and from the actual feed temperature. **The term "horsepower" as applied to boilers is, in fact, a misnomer,** for the boiler cannot produce horsepower or power of any kind unless it explodes.

In land practice, the term "boiler-horsepower" is accepted and specifically defined by Engineering Societies, so that a term loosely used in early days now signifies the definite amount of heat imparted to the steam in evaporating an equivalent of 34½ lbs. of water per hour "from and at 212 deg. F." Stationary boiler manufacturers have further accustomed the land engineering fraternity to consider 10 sq. ft. of heating surface in a boiler as being equal to one horsepower. They labor along with these ideas, and discuss this and that "percentage of rating," etc., in a strenuous endeavor to keep pace with the times. Marine engineers should avoid this archaic anomaly. The boiler makes the steam to run the engine, and the engine, when it runs, makes the power, and the relative amount of power actually developed with the steam, depends entirely upon the efficiency of the engine. Furthermore, as noted above, heating surface will not make steam except as it is accompanied by a furnace capable of burning the fuel.

In estimating the steam required per hour as based on the horsepower of the main engines, it must be remembered that not only power developed, but size, type and special features of design, working pressure, vacuum and whether saturated or superheated steam is used, also the type, number and arrangement of auxiliaries all materially affect the "**water rate**;" *i.e.*, the steam consumption expressed in pounds of water to be evaporated per hour per horsepower. Thus, while the term "horsepower" as applied to boilers is to be avoided, the horsepower of the propelling machinery forms the basis for estimating the required steam output, and from this determining the proper size of boiler.

Rough rules for fixing the limits of grate and heating surface directly from the horsepower of the main engines are sometimes used. These must always be qualified by conditions and no two authorities are likely to agree exactly on the results. Rough estimates of this kind should always be checked by more accurate data. Table 3 represents the views of five different authorities, whose rules have been compared and the maximum and minimum figures set down. They show the range which these rough estimates may take in the hands of various practical engineers. The difference in funnel height accounts in part for the variation, but the results emphasize the need of closer calculations. It may be noted that the minimum estimates for heating surface per horsepower for forced draft are less than for natural draft. This represents Scotch boiler practice, where the maximum amount of heating surface compared to grate surface which can be provided is limited and where the use of the Howden heater compensates to some extent for the loss in efficiency as the boiler is forced. If, however, the design of the boiler permits, it is advisable to keep the heating surface practically the same for both forced and natural draft, as obviously, the steam requirements are about identical in both cases.

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Based on Horsepower of Main Engines

Type of vessel	Propelling machinery	Height of funnel above grate, feet	Kind of fuel	Kind of draft	H.P. per sq. ft. of grate	Sq. ft. of heating surface per H.P.
Tug boat	Compound eng.	30–50	Anth.	Natural	5–8	3.75—6.0
Tug boat	Compound eng.	30–50	egg or	Forced	7–10	3.50—4.5
Tug boat	Triple exp. eng.	30–50	Bit.	Natural	6–10	3.75—5.0
Tug boat	Triple exp. eng.	30–50	coal	Forced	9–14	3.0—4.0
Small cargo vessel	Triple exp. eng.	50–75	Bit. coal	Natural	8–10	3.5—4.0
Small cargo vessel	Triple exp. eng.	50–75	Bit. coal	Forced	12–14	3.0—4.0
Small cargo vessel	Geared turbine	50–75	Bit. coal	Natural	10–12	3.0—3.75
Small cargo vessel	Geared turbine	50–75	Bit. coal	Forced	14–16	2.6—3.3
Large cargo vessel	Triple exp. eng.	60–90	Bit. coal	Natural	9–11	3.00—4.0
Large cargo vessel	Triple exp. eng.	60–90	Bit. coal	Forced	13–14.5	2.75—3.8
Large cargo vessel	Quad. exp. eng.	60–90	Bit. coal	Natural	10–12	2.75—3.75
Large cargo vessel	Quad. exp. eng.	60–90	Bit. coal	Forced	14–16	2.50—3.75
Large cargo vessel	Geared turbine	60–90	Bit. coal	Natural	10–12	3.00—3.75
Large cargo vessel	Geared turbine	60–90	Bit. coal	Forced	14–16	2.50—3.65
Tanker or large cargo vessel	Triple exp. eng.	60–90	Oil	Natural	.....	3.00—3.75
Tanker or large cargo vessel	Triple exp. eng.	60–90	Oil	Forced	.....	2.65—3.6
Tanker or large cargo vessel	Geared turbine	60–90	Oil	Natural	.....	2.80—3.5
Tanker or large cargo vessel	Geared turbine	60–90	Oil	Forced	.....	2.40—3.5
Fast passenger vessel	Geared turbine	70–120	Bit. coal	Natural	12–14	3.00—3.75
Fast passenger vessel	Geared turbine	70–120	Bit. coal	Forced	14–16	2.50—3.60
Fast passenger vessel	Geared turbine	70–120	Oil	Natural	.....	2.80—3.50
Fast passenger vessel	Geared turbine	70–120	Oil	Forced	.....	2.40—3.50
Battleship	Direct turbine	80–90	Oil	Forced	.....	1.5—2.0
Battleship	Geared turbine	80–90	Oil	Forced	.....	1.0—2.0
Destroyer	Direct turbine	40–45	Oil	Forced	.....	1.1
Destroyer	Geared turbine	40–45	Oil	Forced	.....	1.0

Table 4. Steam Consumption for all Purposes

Type of engine	Indicated shaft horse power	Steam consumption per I.H.P. or S.H.P. All purposes	Plus 20 per cent, approximately
Compound engines	400–800	20	24
Triple engines	1200–2000	17	20
Triple engines	3000–5000	16	19
Quadruple	3000–5000	15	18
Geared turbine	1500–2500	14	17
Geared turbine	3000–6000	13	16

In estimating the steam required per hour as based on the horsepower of the main engines, it is customary to include the steam used by the auxiliaries, etc.; in other words, to use a water rate, based on the maximum horsepower of the propelling engines, which will include the steam required for all purposes. The water rate or steam consumption of the machinery of merchant vessels in Table 4 is given by *Mr. Charles F. Bailey*, who states that

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the boiler is added to that of the friction of the gases entering the furnace and the percentage of change with varying fire bed conditions is greatly reduced. While the friction of the gases through a boiler setting varies somewhat with the rate of forcing, design and type of the boiler and the proportion of heating surface to grate surface, these changes are less important than the changed resistance of the fire bed and its erratic effect on the draft gage. For these reasons, the writer believes that it is more practical to base calculations on draft difference between the *uptake* and the fireroom rather than between the fireroom and the furnace. The fireroom pressure is used instead of the ashpit pressure, as at high rates of forcing there is a slight drop in pressure in the passage of the air through the ash-doors.

The following results were obtained in a large number of tests, trial trips, etc., with Babcock & Wilcox Marine Boilers, with semi-bituminous mine-run coal, hand fired:

Draft Required at Different Rates of Combustion

Coal burned per square foot of grate per hour, pounds	Draft required. Difference between boiler damper and fireroom, inches	Coal burned per square foot of grate per hour, pounds	Draft required. Difference between boiler damper and fireroom, inches
7.5	0.10	27.5	0.82
10.0	0.15	30.0	0.95
12.5	0.21	5.0	1.28
15.0	0.28	40.0	1.65
17.5	0.37	45.0	2.07
20.0	0.45	50.0	2.53
22.5	0.55	60.0	3.52
25.0	0.68		

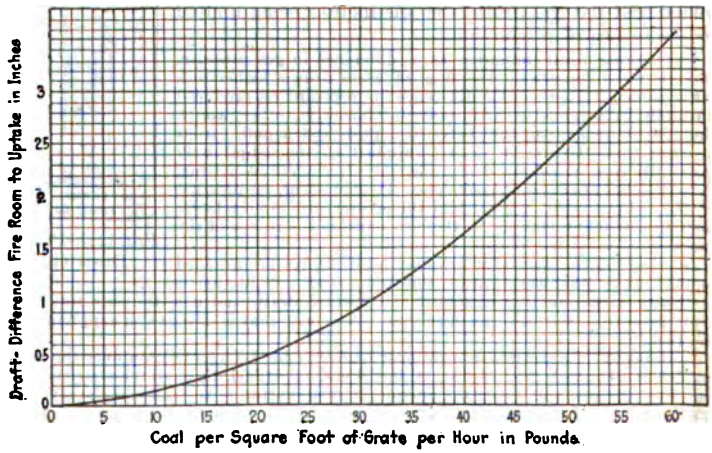

FIG. 12.

These figures are represented graphically in Fig. 12. It is to be noted that this chart may be used for forced draft as well as natural draft, as the ordinates indicate the algebraic difference between draft pressures in the uptake at the outlet from the boiler and the fireroom.

Forced draft for increasing the rate of combustion may be variously applied. In the **Closed-ashpit arrangement**, the blowers force air under pressure to the ashpit, which is made as far as practicable tight against the passage of air except through the furnace grates and the fire bed, thus accelerating the combustion rate. In this combination the relation of the draft pressure to that of the atmosphere becomes a vital factor. The natural draft or suction is a minus pressure (*i.e.*, below the atmospheric pressure), while the pressure in the ashpit is a positive pressure (*i.e.*, greater than the atmosphere). The pressure around the boiler setting remains the same as the atmosphere. The neutral point in the passage of the gases at which the pressure changes from plus to minus (*i.e.*, becomes equal to the atmosphere) depends upon the amount of the natural and forced draft, and also on the resistance of the fire bed and the boiler. Thus, with a strong natural draft and a low ashpit pressure, this neutral point may be in the fire bed itself, and the furnace pressure will be minus. Under these conditions, when the fire-doors are opened, air flows into the furnace. But if the ashpit pressure is increased or the suction lessened, the neutral point may pass on into the boiler passages, or even into the uptake, in which case there will be a positive pressure in the furnace. Then when the fire-door is opened, the hot gases will blow out into the fireroom and this may reach a point where the fires cannot be handled properly. Under these conditions, the "blast" in the ashpit must be shut off or reduced temporarily while the fires are being charged or sliced, and turned on again when the fire-doors are closed. Automatic devices for shutting off the forced draft when the fire-doors are opened are frequently employed with ashpit draft.

An ingenious method of reducing the tendency for hot gases to blow out the fire-door is to admit a portion of the air directly to the furnace over the fire bed through jets around the fire-door opening. These jets discharging toward the rear, create a flow of air inward through the fire-door, thus opposing the hot gases that tend to flow outward. This device is effective to a limited degree in enabling higher ashpit pressures to be carried than would otherwise be possible.

Under usual conditions on shipboard, where ashpit draft is used, it is found that about 0.5 in. or 0.75 in. in the ashpit can be carried without interference with the firing. Above this pressure, the blast must be closed down each time the doors are opened. High forced draft on the closed-ashpit system has the objection that any slight gas leaks in the furnace or boiler setting will permit hot gases to escape outward under the effect of the plus gas pressure. This increases the wear and tear on the setting and causes added heat and discomfort in the fireroom.

The **Balanced-draft** system is a method of proportioning the ashpit and suction draft pressure to give a negative or a very slight positive pressure in the furnace at all times. There is a patented device for doing this automatically, and it is claimed that an improvement in economy and longer life to the setting is secured, as the furnace temperature is not lowered by the inrush of cold air on opening the fire-doors and the temperature of the casing and setting remains at a constant and moderate temperature. The rate of combustion is limited with this system, and it has not been extensively employed in marine work.

Forced draft on the **Closed-fireroom** system consists of making the entire fireroom and stokehold tight against air leakage, and forcing air into this space under pressure by means of **blowers**, usually located outside the compartment. **Deflectors** are frequently employed at the blower discharge for preventing the rush of air into the fireroom, and diffusing it. This is particularly necessary when the room pressure is high, as in the case of naval

apply only to boilers where the shell is in one rim, both heads being flanged in. Where the heads are flanged out, or there is a middle circumferential seam, due allowance should be made. The formulæ give the weight of the bare boiler, and allowance for the weight of fittings and grate bars should be added. This has been done in the weights given in the tables.

Gross weight of boiler in pounds, $W = D^2 \times L \times \left(\frac{P^2}{1666} + 22\right)$ for separate combustion chamber type and

$W = D^2 \times L \times \left(\frac{P^2}{1600} + 20\right)$ for common combustion chamber type.

Weight of water to level 4 in. above combustion chamber top in tons, $Ww = D^2 \times L \times 0.007 + 3.5$ for common combustion chamber type and $Ww = D^2 \times L \times 0.0075 + 4.5$ for separate combustion chamber type, where D = diameter of boiler in ft.; L = length of boiler over heads in ft.; P = working pressure in lbs. per sq. in. above atmos.

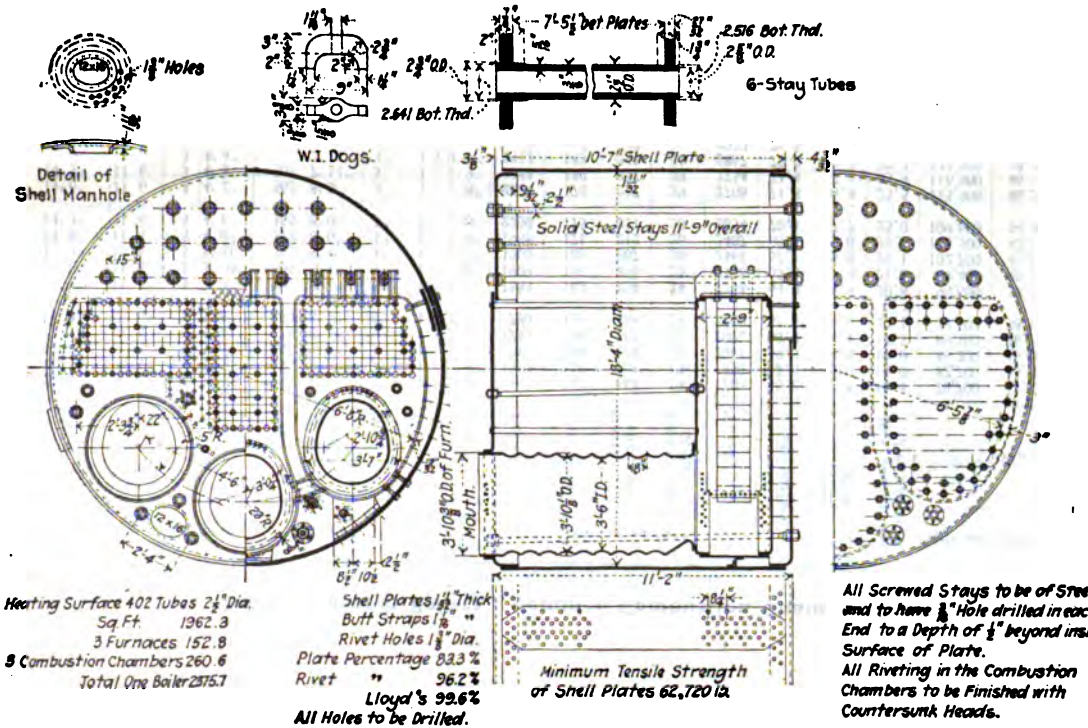
1⅛" Holes
Detail of Shell Manhole
W.I. Dogs.
2.641 Bot. Thd.
7'-5⅛" bet Plates
2.516 Bot. Thd.
2⅝" O.D.
6-Stay Tubes
10'-7" Shell Plate
Solid Steel Stays 11'-9" Overall
10'-4" Diam.
3'-10⅜" O.D. of Furn.
Mouth
3'-9⅛" O.D.
3'-6" I.D.
11'-2"
Heating Surface 402 Tubes 2½" Dia.
Sq. Ft. 1962.3
3 Furnaces 152.8
3 Combustion Chambers 260.6
Total One Boiler 2375.7
Shell Plates 1⅛" Thick
Butt Straps 1⅛" "
Rivet Holes 1⅛" Dia.
Plate Percentage 83.3%
Rivet " 96.2%
Lloyd's 99.6%
All Holes to be Drilled.
Minimum Tensile Strength of Shell Plates 62,720 lb.
All Screwed Stays to be of Steel and to have ⅜" Hole drilled in each End to a Depth of ½" beyond inside Surface of Plate.
All Riveting in the Combustion Chambers to be Finished with Countersunk Heads.

FIG. 18.—Scotch Boiler, Separate Combustion Chamber Type.

Table 6. Scotch Boilers. Common Combustion Chamber Type

All Grate Bars 6 Ft. 3 In. Long

Dimensions		Furnaces		Tubes			Combustion chamber		Heating surface					Grate surface	H.S. / G.S.	H.S. per lin. in. tubes and furn.	Approximate weight, lbs.		
Diam.	Length	No.	Diam.	No.	Length	Diam.	No.	Depth	Tubes	Furnaces	Comb. chamber	Front tube sheet	Total				Boiler, dry	Water	Total
12′ 0″	11 6	3	3′ 4″	228	8′ 2″	3″	1	2′ 6″	1462	147	161	19	1789	56.2	31.9	16.4	79,700	33,850	113,550
12′ 3″	11 6	3	3′ 5″	236	8′ 2″	3″	1	2′ 6″	1513	150	168	19	1850	57.8	32.0	18.0	82,700	35,200	117,900
12′ 6″	11 6	3	3′ 6″	267	8′ 2″	3″	1	2′ 6″	1712	154	174	21	2061	59.3	34.8	19.0	86,200	36,100	122,300
12′ 9″	11 6	3	3′ 7″	273	8′ 1″	3″	1	2′ 7″	1733	156	181	22	2092	60.9	34.4	19.5	89,500	36,780	126,280
13′ 0″	11 6	3	3′ 8″	282	8′ 1″	3″	1	2′ 7″	1790	159	188	23	2160	62.5	34.6	20.1	92,500	38,320	130,820
13′ 3″	11 6	3	3′ 9″	292	8′ 1″	3″	1	2′ 7″	1853	163	194	24	2234	64.0	34.9	20.8	95,600	39,700	135,300
13′ 6″	11 6	3	3′ 10″	304	8′ 1″	3″	1	2 7″	1929	166	198	25	2318	65.6	35.4	21.6	99,400	40,550	139,950
13′ 9″	11 6	3	3′ 11″	312	8′ 0″	3″	1	2′ 8″	1960	168	207	26	2361	67.1	35.2	22.1	102,700	41,450	144,150
14′ 0″	11 6	3	4′ 0″	318	8′ 0″	3″	1	2′ 8″	1998	171	212	26	2407	68.8	35.0	22.6	106,500	42,550	149,050
14′ 3″	11 6	3	4′ 1″	355	8′ 0″	3″	1	2′ 8″	2230	175	221	28	2654	70.3	37.7	25.0	109,800	44,400	154,200
14′ 6″	11 6	3	4′ 2″	362	8′ 0″	3″	1	2′ 8″	2274	178	227	29	2708	71.8	36.3	25.5	113,500	46,200	159,700
14′ 9″	11 6	3	4′ 3″	368	7′ 11″	3″	1	2′ 9″	2288	180	237	29	2734	73.4	37.2	25.9	116,900	46,800	163,700
15′ 0″	11 6	3	4′ 4″	388	7′ 11″	3″	1	2′ 9″	2412	183	241	31	2867	75.0	38.2	27.3	121,000	48,200	169,200
15′ 3″	11 6	3	4′ 5″	396	7′ 11″	3″	1	2′ 9″	2462	187	249	32	2930	76.5	38.3	27.8	124,700	49,350	174,050

Table 7. Scotch Boilers. Separate Combustion Chamber Type

All Grate Bars 6 Ft. 3 In. Long

Dimensions		Furnaces		Tubes			Combustion chambers		Heating surface				Grate surface	H.S. / G.S.	H.S. per lin. in. tubes and furnaces	Approximate weight, lbs.		
Diam.	Length	No.	Diam.	No.	Length	Diam.	No.	Depth	Tubes	Furnaces	Comb. chambers	Total				Boiler, dry	Water	Total
14′ 0″	11′ 6″	3	3′ 7″	285	7′ 9″	3″	3	2′ 9″	1734	147	242	2123	60.9	34.8	20.2	107,000	46,800	153,800
14′ 3″	11′ 6″	3	3′ 8″	292	7′ 9″	3″	3	2′ 9″	1777	151	249	2177	62.5	34.8	20.7	110,800	48,350	159,150
14′ 6″	11′ 6″	3	3′ 9″	295	7′ 9″	3″	3	2′ 9″	1795	155	255	2205	64.0	34.4	20.9	114,400	50,000	164,400
14′ 9″	11′ 6″	3	3′ 10″	309	7′ 9″	3″	3	2′ 9″	1880	157	262	2299	65.6	35.0	21.8	118,400	51,400	169,800
15′ 0″	11′ 6″	3	3′ 11″	317	7′ 9″	3″	3	2′ 9″	1929	161	275	2365	67.1	35.2	22.4	121,900	51,750	172,650
15′ 3″	11′ 6″	3	4′ 0″	347	7′ 9″	3″	3	2′ 9″	2111	164	275	2550	68.8	37.0	24.4	126,100	53,800	179,900
15′ 6″	11′ 6″	3	4′ 1″	352	7′ 9″	3″	3	2′ 9″	2142	168	280	2590	70.3	36.8	24.8	129,800	55,400	185,200
15′ 9″	11′ 6″	3	4′ 2″	369	7′ 9″	3″	3	2′ 9″	2245	171	286	2702	71.8	37.6	26.0	134,300	57,200	191,500
16′ 0″	11′ 6″	3	4′ 3″	376	7′ 9″	3″	3	2′ 9″	2288	175	294	2757	73.4	37.5	26.5	138,200	58,500	196,700
16′ 3″	11′ 6″	3	4′ 4″	387	7′ 9″	3″	3	2′ 9″	2355	178	305	2838	75.0	37.8	27.2	143,000	61,700	204,700
16′ 6″	11′ 6″	4	3′ 8″	414	7′ 9″	3″	4	2′ 9″	2519	202	351	3072	83.2	36.9	29.2	148,700	65,000	213,700
16′ 9″	11′ 6″	4	3′ 9″	439	7′ 9″	3″	4	2′ 9″	2671	206	357	3234	85.3	37.9	30.9	152,900	65,800	218,700
17′ 0″	11′ 6″	4	3′ 10″	445	7′ 9″	3″	4	2′ 9″	2883	211	361	3455	87.5	39.4	33.2	156,700	67,300	224,000

Table 8. Scotch Boilers. Separate Combustion Chamber Type

All grate Bars 5 Ft. 6 In. Long

Dimensions		Furnaces		Tubes			Combustion chambers		Heating surface				Grate surface	H.S./G.S.	H.S. per lin. in. tubes and furn.	Approximate weight, lbs.		
Diam.	Length	No.	Diam.	No.	Length	Diam.	No.	Depth	Tubes	Furnaces	Comb. chambers	Total				Boiler, dry	Water	Total
14′ 0″	12′ 0″	3	3′ 7″	370	8′ 3″	2¼″	3	2′ 9″	1,997	156	242	2,395	53.6	44.7	21.7	111,500	49,100	160,600
14′ 3″	12′ 0″	3	3′ 8″	378	8′ 3″	2¼″	3	2′ 9″	2,041	161	249	2,451	55.0	44.5	22.2	114,800	50,700	165,500
14′ 6″	12′ 0″	3	3′ 9″	382	8′ 3″	2¼″	3	2′ 9″	2,062	164	255	2,481	56.3	44.1	22.5	119,100	52,250	171,350
14′ 9″	12′ 0″	3	3′ 10″	404	8′ 3″	2¼″	3	2′ 9″	2,181	168	262	2,611	57.7	45.2	23.7	123,100	53,800	176,900
15′ 0″	12′ 0″	3	3′ 11″	412	8′ 3″	2½″	3	2′ 9″	2,224	171	275	2,670	59.1	45.2	24.2	126,700	54,200	180,900
15′ 3″	12′ 0″	3	4′ 0″	451	8′ 3″	2½″	3	2′ 9″	2,435	174	275	2,884	60.5	47.6	26.3	131,200	56,200	187,400
15′ 6″	12′ 0″	3	4′ 1″	455	8′ 3″	2½″	3	2′ 9″	2,456	178	280	2,914	61.8	47.2	26.6	134,800	57,900	192,700
15′ 9″	12′ 0″	3	4′ 2″	480	8′ 3″	2½″	3	2′ 9″	2,591	182	286	3,059	63.2	48.4	28.0	139,600	59,800	199,400
16′ 0″	12′ 0″	3	4′ 3″	484	8′ 3″	2½″	3	2′ 9″	2,613	185	294	3,092	64.6	47.8	28.2	143,700	61,200	204,900
16′ 3″	12′ 0″	3	4′ 4″	505	8′ 3″	2½″	3	2′ 9″	2,726	189	305	3,220	66.0	48.8	30.6	148,200	64,550	212,750
16′ 6″	12′ 0″	4	3′ 8″	538	8′ 3″	2½″	4	2′ 9″	2,904	215	351	3,470	73.3	47.3	31.5	154,200	68,000	222,200
16′ 9″	12′ 0″	4	3′ 9″	570	8′ 3″	2½″	4	2′ 9″	3,077	219	357	3,653	75.1	48.6	33.3	158,000	68,600	226,600
17′ 0″	12′ 0″	4	3′ 10″	578	8′ 3″	2½″	4	2′ 9″	3,120	224	361	3,705	77.0	48.2	33.7	163,000	70,450	233,450

Almy Boilers. Tables 9 to 13 give some of the sizes of Almy boilers.

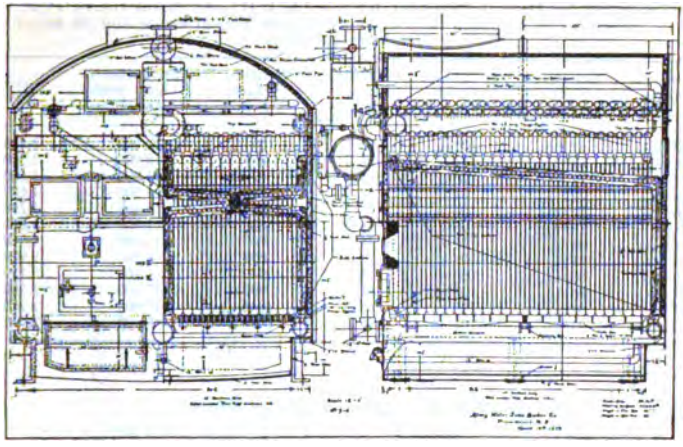

FIG. 19.—Almy Boiler.

Table 9. Class A (Launch Type), Almy Boilers

Tubes 1 1/16 In. Outside Diameter, 11 In. Thick (3/4-in. Standard Pipe). Based on 12 In. from Ashpan

Width of casing, in.	Length of casing, in.	Height of casing, in.	Grate area, sq. ft.	Heating surface, sq. ft.	Approximate weight, dry, lbs.	Water, lbs.
26 1/16	26 1/16	49 3/16	2.6	87	1150	100
28 15/16	26 1/16	49 7/16	3.0	94	1240	105
28 15/16	28 15/16	50	3.4	104	1370	115
31 13/16	28 15/16	50 1/2	3.9	114	1490	125
31 13/16	31 13/16	50 7/8	4.4	125	1500	135
34 11/16	31 13/16	51 1/8	4.9	135	1620	145
34 11/16	34 11/16	51 3/8	5.4	147	1820	155
37 9/16	34 11/16	51 5/8	6.0	158	2020	170
37 9/16	37 9/16	51 7/8	6.6	172	2150	185

Table 10. Class B, Almy Boilers

Tubes 1 5/16 In. Outside Diameter, 0.18 In. Thick (1-in. Extra Heavy Pipe). Based on 20 In. from Ashpan to Top of Grate. 20 In. Height of Fire Box

Width of casing, in.	Length of casing, in.	Height of casing, in.	Grate area, sq. ft.	Heating surface, sq. ft.	Approximate weight, dry, lbs.	Water, lbs.
44 25/32	44 25/32	78 1/4	9.1	257	4800	400
48 3/16	44 25/32	78 7/8	9.9	275	5060	430
48 3/16	48 3/16	79 1/8	10.9	297	5360	460
51 19/32	48 3/16	79 3/4	11.8	316	5640	490
51 19/32	51 19/32	80	12.8	338	5970	520
55	51 19/32	80 1/2	13.8	358	6270	550
55	55	80 1/4	14.9	382	6630	590
58 13/32	55	81 3/8	16.0	404	6960	620
58 13/32	58 13/32	81 1/2	17.2	430	7330	660
61 13/32	58 13/32	82 1/8	18.4	449	7690	690
61 13/16	61 13/16	82 3/4	19.6	475	8090	730
65 1/32	61 13/16	83 1/2	20.9	499	8470	760

Table 11. Class C, Almy Boilers

Tubes 1⅜ In. Outside Diameter, 0.2 In. Thick (1½-in. Extra Heavy Pipe). Based on 24 In. from Bottom of Ashpan to Top of Grate and 26 In. Height of Fire Box

Width of casing, in.	Length of casing, in.	Height of casing, in.	Grate area, sq. ft.	Heating surface, sq. ft.	Approximate weight with water, lbs.	Water, lbs.
67 15/32	67 15/32	102¾	21.7	591	12,500	1,440
72 3/16	67 15/32	103 3/16	23.5	629	12,900	1,520
72 3/16	72 3/16	103¾	25.5	670	14,000	1,680
76 29/32	72 3/16	104 3/16	27.5	711	14,500	1,680
76 29/32	76 29/32	104 5/16	29.7	761	15,400	1,780
81 5/16	76 29/32	105¼	31.8	803	15,900	1,860
81 5/16	81 5/16	105½	34.1	849	16,500	1,940
86 1/32	81 5/16	106¼	36.4	894	16,800	2,020
86 1/32	86 1/32	106½	38.8	949	17,300	2,120
91 1/16	86 1/32	107 1/16	41.3	997	17,600	2,200
91 1/16	91 1/16	107½	43.9	1048	17,900	2,280
95 25/32	91 1/16	108	46.5	1098	18,200	2,350

Table 12. Class D, Almy Boilers

Tubes 1 5/16 In. Outside Diameter, 0.18 In. Thick (1-in. Extra Heavy Pipe). Based on 20 In. from Bottom of Ashpans to Top of Grate and 26 In. Height of Fire Box. Single Fire Box.

Width of casing, in.	Length of casing, in.	Height of casing, in.	Grate area, sq. ft.	Heating surface, sq. ft.	Approximate weight, dry, lbs.	Water, lbs.
55 1/16	51 25/32	92⅞	12.3	479.4	8,100	680
55 1/16	55 1/16	93 1/16	13.3	510.9	8,500	720
58 25/32	55 1/16	93¾	14.4	537.2	9,000	760
58 25/32	58 25/32	93 15/16	15.5	573.8	9,500	810
61⅞	58 25/32	94½	16.6	601.8	9,900	850
61⅞	61⅞	94¾	17.8	636.3	10,400	900
65 9/32	61⅞	95 3/16	19.0	665.3	10,800	940
65 9/32	65 9/32	95½	20.3	703.9	11,300	990
68 11/16	65 9/32	96 1/16	21.6	734.9	11,700	1,030
68 11/16	68 11/16	96 5/16	22.9	772.5	12,200	1,080
72 3/32	68 11/16	97	24.3	805.0	12,500	1,120
72 3/32	72 3/32	97¼	25.7	844.1	13,000	1,170
75½	72 3/32	97⅞	27.2	877.9	13,400	1,220
75½	75½	98	28.7	918.7	12,600	1,270
75½	78 29/32	98¼	30.2	962.5	14,300	1,330
78 29/32	78 29/32	98⅞	31.8	994.4	14,700	1,370
78 29/32	82 5/16	99	33.4	1,041.8	15,100	1,420
78 29/32	85 23/32	99¼	35.0	1,084.1	15,500	1,470

Table 13. Class E, Almy Boilers

Tubes 1 5/16 In. Outside Diameter, 0.18 In. Thick (1-in. Extra Heavy Pipe). Based on 20 In. from Bottom of Ashpans to Top of Grate and 26 In. Height of Fire Boxes. Two Fire Boxes.

Width of casing, in.	Length of casing, in.	Grate area	Total heating surface	Total height	Approximate weight, dry, lbs.	Water, lbs.
79 15/32	75 1/2	27.25	1133.8	102.9	16,400	1700
79 15/32	78 29/32	28.7	1185.6	103.1	17,000	1780
79 15/32	82 1/16	30.15	1237.6	103.4	17,700	1860
79 15/32	85 23/32	31.6	1289.8	103.6	18,300	1940
79 15/32	89 1/8	33.05	1342.2	103.8	19,000	2020
79 15/32	92 17/32	34.51	1394.8	104.0	19,600	2100
86 9/32	75 1/2	30.27	1202.7	105.0	16,300	1810
86 9/32	78 29/32	31.88	1256.6	105.2	17,900	1890
86 9/32	82 1/16	33.5	1310.7	105.5	18,600	1970
86 9/32	85 23/32	35.11	1365.0	105.7	19,200	2050
86 9/32	89 1/8	36.72	1419.4	105.9	19,900	2130
86 9/32	92 17/32	38.34	1474.1	106.1	20,600	2210
93 1/32	75 1/2	33.3	1274.1	107.1	18,100	1920
93 1/32	78 29/32	35.07	1330.1	107.4	18,800	2000
93 1/32	82 1/16	36.84	1386.3	107.6	19,500	2080
93 1/32	85 23/32	38.62	1442.6	107.8	20,200	2170
93 1/32	89 1/8	40.39	1499.2	108.0	20,800	2250
93 1/32	92 17/32	42.17	1556.0	108.2	21,500	2340
93 1/32	95 15/16	43.94	1612.9	108.4	22,100	2420
93 1/32	99 11/32	45.71	1670.1	108.6	22,700	2510
99 29/32	75 1/2	36.32	1347.5	109.2	19,000	2020
99 29/32	78 29/32	38.26	1405.5	109.5	19,700	2110
99 29/32	82 1/16	40.19	1463.8	109.7	20,300	2200
99 29/32	85 23/32	42.13	1522.2	109.9	21,100	2290
99 29/32	89 1/8	44.06	1580.8	110.1	21,700	2370
99 29/32	92 17/32	46.0	1639.7	110.3	22,400	2460
99 29/32	95 15/16	47.93	1698.7	110.5	23,000	2550
99 29/32	99 11/32	49.87	1757.9	110.7	23,700	2640
106 23/32	75 1/2	39.35	1417.1	111.4	19,900	2130
106 23/32	78 29/32	41.44	1477.3	111.6	20,500	2220
106 23/32	82 1/16	43.54	1537.6	111.8	21,300	2310
106 23/32	85 23/32	45.64	1598.2	112.0	21,900	2400
106 23/32	89 1/8	47.73	1658.2	112.2	22,600	2490
106 23/32	92 17/32	49.83	1719.9	112.4	23,300	2580
106 23/32	95 15/16	51.92	1781.0	112.6	23,900	2680
106 23/32	99 11/32	54.02	1842.3	112.8	24,600	2770

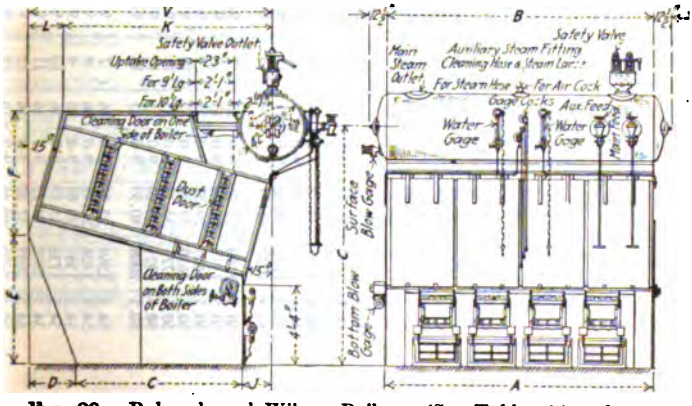

FIG. 20.—Babcock and Wilcox Boiler. (See Tables 14 and 15.)

Table 14. Babcock & Wilcox Boiler (see Fig. 20, p. 367)

Tube 4 In. Outside Diam., 9 Ft. Long

Grate surface, sq. ft.	Heating surface, sq. ft.	Ratio	Length of tube, ft. in.	No. of sections	No. of tubes per section	Dimensions											Approx. weights		
						A, ft. in.	*B*, ft. in.	*C*, ft. in.	*D*, in.	*E*, ft. in.	*F*, ft. in.	*G*, ft. in.	*J*, in.	*K*, ft. in.	*L*, in.	*V*, ft. in.	Boiler	Water	Total
42.3	1031	24.37	9-0	13	7	8- 1½	8- 2¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	43100	7500	50600
42.3	1158	27.38	9-0	13	8	8- 1½	8- 2¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	46700	8100	54800
42.3	1285	30.38	9-0	13	9	8- 1½	8- 2¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	49600	8700	58300
42.3	1412	33.38	9-0	13	10	8- 1½	8- 2¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	52400	9300	61700
42.3	1537	36.34	9-0	13	11	8- 1½	8- 2¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	55800	9800	65600
42.3	1665	39.36	9-0	13	12	8- 1½	8- 2¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	59300	10300	69500
42.3	1794	42.41	9-0	13	13	8- 1½	8- 2¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	62600	11000	73600
46.4	1111	23.94	9-0	14	7	8- 8½	8- 9¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	46400	8000	54400
46.4	1247	26.88	9-0	14	8	8- 8½	8- 9¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	49000	8600	57600
46.4	1384	29.83	9-0	14	9	8- 8½	8- 9¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	52000	9300	61300
46.4	1521	32.78	9-0	14	10	8- 8½	8- 9¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	55300	9900	65200
46.4	1656	35.69	9-0	14	11	8-8½	8- 9¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	58300	10600	68900
46.4	1792	38.62	9-0	14	12	8- 8½	8- 9¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	61300	11100	72400
46.4	1929	41.57	9-0	14	13	8- 8½	8- 9¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	64300	11600	75900
50.5	1190	23.56	9-0	15	7	9- 3½	9- 7⅜	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	48700	8500	57200
50.5	1337	26.48	9-0	15	8	9- 3½	9- 7⅜	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	51500	9300	60800
50.5	1483	29.37	9-0	15	9	9- 3½	9- 7⅜	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	54900	9900	64800
50.5	1630	32.28	9-0	15	10	9- 3½	9- 7⅜	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	58000	10600	68600
50.5	1775	35.15	9-0	15	11	9- 3½	9- 7⅜	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	61400	11300	72700
50.5	1921	38.04	9-0	15	12	9- 3½	9- 7⅜	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	64800	12000	76800
50.5	2068	40.95	9-0	15	13	9- 3½	9- 7⅜	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	68200	12700	80900
54.5	1269	23.28	9-0	16	7	9-10½	9-11¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	51100	9100	60200
54.5	1426	26.17	9-0	16	8	9-10½	9-11¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	53900	9800	63700
54.5	1581	29.01	9-0	16	9	9-10½	9-11¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	57500	10500	68000
54.5	1738	31.89	9-0	16	10	9-10½	9-11¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	60700	11300	72000
54.5	1892	34.72	9-0	16	11	9-10½	9-11¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	64200	12100	76300
54.5	2049	37.60	9-0	16	12	9-10½	9-11¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	67700	12900	80600
54.5	2205	40.46	9-0	16	13	9-10½	9-11¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	71200	13700	84900

58.6	1348	23.00	9-0	17	7	10- 5½	10- 9½	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	53900	9600	62800
58.6	1515	25.85	9-0	17	8	10- 5½	10- 9¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	56300	10400	66700
58.6	1680	28.67	9-0	17	9	10- 5½	10- 9¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	60300	11300	71600
58.6	1846	31.50	9-0	17	10	10- 5½	10- 9¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	63500	12000	75500
58.6	2011	34.32	9-0	17	11	10- 5½	10- 9¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	67200	12900	80100
58.6	2177	37.15	9-0	17	12	10- 5½	10- 9¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	70700	13800	84500
58.6	2343	39.98	9-0	17	13	10- 5½	10- 9¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	74200	14700	88900
62.7	1498	22.78	9-0	18	7	11- 0½	11- 1¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	55500	10200	65700
62.7	1604	25.58	9-0	18	8	11- 0½	11- 1¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	58700	11100	69800
62.7	1779	28.37	9-0	18	9	11- 0½	11- 1¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	62500	11900	74400
62.7	1955	31.18	9-0	18	10	11- 0½	11- 1¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	66200	12700	78900
62.7	2129	33.96	9-0	18	11	11- 0½	11- 1¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	69000	13500	82500
62.7	2305	36.76	9-0	18	12	11- 0½	11- 1¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	71800	14300	86100
62.7	2480	39.55	9-0	18	13	11- 0½	11- 1¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	74600	15100	89700
66.8	1507	22.56	9-0	19	7	11- 7½	11-11⅜	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	57900	10800	68700
66.8	1693	25.34	9-0	19	8	11- 7½	11-11⅜	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	61100	11700	72800
66.8	1878	28.11	9-0	19	9	11- 7½	11-11⅜	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	65200	12600	77800
66.8	2064	30.90	9-0	19	10	11- 7½	11-11⅜	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	69000	13400	82400
66.8	2247	33.64	9-0	19	11	11- 7½	11-11⅜	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	73000	14300	87300
66.8	2433	36.42	9-0	19	12	11- 7½	11-11⅜	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	77000	15200	92200
66.8	2619	39.21	9-0	19	13	11- 7½	11-11⅜	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	81000	16100	97100
70.9	1586	22.37	9-0	20	7	12- 2½	12- 6¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	59600	11400	71000
70.9	1782	25.13	9-0	20	8	12- 2½	12- 6¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	63700	12100	75800
70.9	1977	27.88	9-0	20	9	12- 2½	12- 6¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	67700	13200	80900
70.9	2172	30.63	9-0	20	10	12- 2½	12- 6¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	71900	14100	86000
70.9	2365	33.36	9-0	20	11	12- 2½	12- 6¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	76000	15100	91100
70.9	2561	36.12	9-0	20	12	12- 2½	12- 6¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	80100	16100	96200
70.9	2756	38.87	9-0	20	13	12- 2½	12- 6¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	84200	17100	101300
75.0	1665	22.20	9-0	21	7	12- 9½	12-10¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	62000	11900	75900
75.0	1871	24.95	9-0	21	8	12- 9½	12-10¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	65000	12900	77900
75.0	2076	27.68	9-0	21	9	12- 9½	12-10¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	70200	13900	84100
75.0	2281	30.41	9-0	21	10	12- 9½	12-10¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	74500	14800	89300
75.0	2484	33.12	9-0	21	11	12- 9½	12-10¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	79000	15800	94800
75.0	2689	35.85	9-0	21	12	12- 9½	12-10¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	83500	16800	100300
75.0	2894	38.59	9-0	21	13	12- 9½	12-10¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	88000	17800	105800

Table 14. Babcock & Wilcox Boiler (Continued) (see Fig. 20, p. 367)

Tube 4 In. Outside Diam., 9 Ft. Long

Grate surface, sq. ft.	Heating surface, sq. ft.	Ratio	Length of tube, ft. in.	No. of sections	No. of tubes per section	Dimensions										Approx. weights		
						A, ft. in.	*B*, ft. in.	*C*, ft. in.	*D*, in.	*E*, ft. in.	*F*, ft. in.	*G*, ft. in.	*J*, in.	*K*, ft. in.	*L*, in.	*V*, ft. in.	Boiler	Water	Total
79.0	1744	22.08	9-0	22	7	13- 4½	13- 5¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	64300	12100	76400
79.0	1960	24.81	9-0	22	8	13- 4½	13- 5¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	68600	13100	81700
79.0	2174	27.52	9-0	22	9	13- 4½	13- 5¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	73000	14100	87100
79.0	2390	30.25	9-0	22	10	13- 4½	13- 5¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	77500	15200	92700
79.0	2602	32.94	9-0	22	11	13- 4½	13- 5¾	9-4	14	7-1¼	6 -9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	81700	16200	97900
79.0	2817	35.66	9-0	22	12	13- 4½	13- 5¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	85900	17200	103100
79.0	3032	38.38	9-0	22	13	13- 4½	13- 5¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	90100	18200	108300
83.1	1825	21.96	9-0	23	7	13-11½	14- 0¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	66500	12700	79200
83.1	2049	24.66	9-0	23	8	13-11½	14- 0¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	71100	13800	84900
83.1	2273	27.35	9-0	23	9	13-11½	14- 0¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	74600	14900	89500
83.1	2498	30.06	9-0	23	10	13-11½	14- 0¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	80000	15900	95900
83.1	2720	32.73	9-0	23	11	13-11½	14- 0¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	84700	16900	101600
83.1	2945	35.44	9-0	23	12	13-11½	14- 0¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	89400	17900	107300
83.1	3170	38.15	9-0	23	13	13-11½	14- 0¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	94100	18900	113000
87.2	1904	21.83	9-0	24	7	14- 6½	14- 7¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	68700	13200	81900
87.2	2138	24.52	9-0	24	8	14- 6½	14- 7¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	73500	14300	87800
87.2	2372	27.20	9-0	24	9	14- 6½	14- 7¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	78100	15500	93600
87.2	2607	29.90	9-0	24	10	14- 6½	14- 7¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	83000	16600	99600
87.2	2839	32.56	9-0	24	11	14- 6½	14- 7¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	87600	17700	105300
87.2	3073	35.24	9-0	24	12	14- 6½	14- 7¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	92200	18800	111000
87.2	3307	37.92	9-0	24	13	14- 6½	14- 7¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	96800	19900	116700
91.3	1983	21.72	9-0	25	7	15- 1½	15- 2¾	9-4	14	7-1¼	4-10½	11- 1½	13¼	10-3¾	15½	11- 7¼	71100	13700	84800
91.3	2228	24.40	9-0	25	8	15- 1½	15- 2¾	9-4	14	7-1¼	5- 4¼	11- 7¼	14¾	10-3¾	17	11- 8¾	76000	15000	91000
91.3	2471	27.06	9-0	25	9	15- 1½	15- 2¾	9-4	14	7-1¼	5-10	12- 1	16½	10-3¾	18¾	11-10½	80700	16200	969000
91.3	2716	29.75	9-0	25	10	15- 1½	15- 2¾	9-4	14	7-1¼	6- 4	12- 7	18	10-3¾	20¼	12- 0	85700	17300	102000
91.3	2957	32.39	9-0	25	11	15- 1½	15- 2¾	9-4	14	7-1¼	6- 9¾	13- 0¾	19½	10-3¾	21¾	12- 1½	90500	18500	109000
91.3	3201	35.06	9-0	25	12	15- 1½	15- 2¾	9-4	14	7-1¼	7- 3½	13- 6½	21	10-3¾	23¼	12- 3	95300	19700	115000
91.3	3446	37.74	9-0	25	13	15- 1½	15- 2¾	9-4	14	7-1¼	7- 9¼	14- 0¼	22½	10-3¾	24¾	12- 4½	100100	20900	121000

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Table 15. Babcock & Wilcox Boiler (see Fig. 20, p. 367)

Tube 4 In. Outside Diam., 10 Ft. Long

Grate surface, sq. ft.	Heating surface, sq. ft.	Ratio	Length of tube, ft. in.	No. of sections	No. of tubes per section	Dimensions											Approx. weights		
						A, ft. in.	B, ft. in.	C, ft. in.	D, in.	E, ft. in.	F, ft. in.	G, ft. in.	J, in.	K, ft. in.	L, ft. in.	V, ft. in.	Boiler	Water	Total
42.3	1144	27.04	10–0	13	7	8– 1½	8– 2¾	9–0	2–6½	7-2	5– 1	11– 4¾	14	11–4⅛	16⅜	12– 8½	48500	8100	56600
42.3	1284	30.35	10–0	13	8	8– 1½	8– 2¾	9–0	2–6½	7-2	5– 6¾	11–10½	15½	11–4⅛	17⅝	12–10	51700	8800	60500
42.3	1424	33.66	10–0	13	9	8– 1½	8– 2¾	9–0	2–6½	7-2	6– 0½	12– 4¼	17¾	11–4⅛	19⅝	12–11¾	54000	9400	63400
42.3	1565	37.00	10–0	13	10	8– 1½	8– 2¾	9–0	2–6½	7-2	6– 6¾	12–10	18¾	11–4⅛	21⅛	13– 1¼	57000	10000	67000
42.3	1704	40.28	10–0	13	11	8– 1½	8– 2¾	9–0	2–6½	7-2	7– 0	13– 3¾	20¼	11–4⅛	22⅝	13– 2¾	60000	10700	70700
42.3	1845	43.62	10–0	13	12	8– 1½	8– 2¾	9–0	2–6½	7-2	7– 5¾	13– 9½	21¾	11–4⅛	2– 0⅛	13– 4¼	63000	11400	74400
42.3	1988	47.00	10–0	13	13	8– 1½	8– 2¾	9–0	2–6½	7-2	7–11½	14– 3¼	23½	11–4⅛	2– 1⅞	13– 6	66000	12100	78100
46.4	1231	26.53	10–0	14	7	8– 8½	8– 9¾	9.0	2–6½	7-2	5– 1	11– 4¾	14	11–4⅛	16⅜	12– 8½	51000	8600	59600
46.4	1382	29.78	10–0	14	8	8– 8½	8– 9¾	9–0	2–6½	7-2	5– 6¾	11–10½	15½	11–4⅛	17⅝	12–10	54000	9300	63300
46.4	1534	33.06	10–0	14	9	8– 8½	8– 9¾	9–0	2–6½	7-2	6– 0½	12– 4¼	17¾	11–4⅛	19⅝	12–11¾	56600	10000	66600
46.4	1686	36.34	10–0	14	10	8– 8½	8– 9¾	9–0	2–6½	7-2	6– 6¾	12–10	18¾	11–4⅛	21⅛	13– 1¼	60000	10700	70700
46.4	1835	39.55	10–0	14	11	8– 8½	8– 9¾	9–0	2–6½	7-2	7– 0	13– 3¾	20¼	11–4⅛	22⅝	13– 2¾	63000	11300	74300
46.4	1986	42.80	10–0	14	12	8– 8½	8– 9¾	9–0	2–6½	7-2	7– 5¾	13– 9½	21¾	11–4⅛	2– 0⅛	13– 4¼	66000	11900	77900
46.4	2138	46.08	10–0	14	13	8– 8½	8– 9¾	9–0	2–6½	7-2	7–11½	14– 3¼	23½	11–4⅛	2– 1⅞	13– 6	69000	12500	81500
50.5	1319	26.12	10–0	15	7	9– 3½	9– 7⅜	9–0	2–6½	7-2	5– 1	11– 4¾	14	11–4⅛	16⅜	12– 8½	53300	9200	62500
50.5	1482	29.35	10–0	15	8	9– 3½	9– 7⅜	9–0	2–6½	7-2	5– 6¾	11–10½	15½	11–4⅛	17⅝	12–10	56500	9900	66400
50.5	1643	32.53	10–0	15	9	9– 3½	9– 7⅜	9–0	2–6½	7-2	6– 0½	12– 4¼	17¾	11–4⅛	19⅝	12–11¾	59500	10600	70100
50.5	1806	35.76	10–0	15	10	9– 3½	9– 7⅜	9–0	2–6½	7-2	6– 6¾	12–10	18¾	11–4⅛	21⅛	13– 1¼	62800	11400	74200
50.5	1967	38.95	10–0	15	11	9– 3½	9– 7⅜	9–0	2–6½	7-2	7– 0	13– 3¾	20¼	11–4⅛	22⅝	13– 2¾	66200	12100	78300
50.5	2129	42.16	10–0	15	12	9– 3½	9– 7⅜	9–0	2–6½	7-2	7– 5¾	13– 9½	21¾	11–4⅛	2– 0⅛	13– 4¼	69600	12800	82400
50.5	2291	45.37	10–0	15	13	9– 3½	9– 7⅜	9–0	2–6½	7-2	7–11½	14– 3¼	23½	11–4⅛	2– 1⅞	13– 6	73000	13500	86500
54.5	1407	25.82	10–0	16	7	9–10½	9–11¾	9–0	2–6½	7-2	5– 1	11– 4¾	14	11–4⅛	16⅜	12– 8½	55600	9700	65300
54.5	1581	29.01	10–0	16	8	9–10½	9–11¾	9–0	2–6½	7-2	5– 6¾	11–10½	15½	11–4⅛	17⅝	12–10	58700	10400	69100
54.5	1753	32.17	10–0	16	9	9–10½	9–11¾	9–0	2–6½	7-2	6– 0½	12– 4¼	17¼	11 4⅛	19⅝	12–11¾	62100	11200	73300
54.5	1926	35.34	10–0	16	10	9–10½	9–11¾	9–0	2–6½	7-2	6– 6¾	12–10	18¾	11–4⅛	21⅛	13– 1¼	65500	12100	77600
54.5	2079	38.48	10–0	16	11	9–10½	9–11¾	9–0	2–6½	7-2	7– 0	13– 3¾	20¼	11–4⅛	22⅝	13– 2¾	69000	12800	81800
54.5	2271	41.67	10–0	16	12	9–10½	9–11¾	9–0	2–6½	7-2	7– 5¾	13– 9½	21¾	11–4⅛	2– 0⅛	13– 4¼	72500	13500	86000
54.5	2443	44.83	10–0	16	13	9–10½	9–11¾	9–0	2–6½	7-2	7–11½	14– 3¼	23½	11–4⅛	2– 1⅞	13– 6	76000	14200	90200

58.6	1495	25.51	10-0	17	7	10- 5½	10- 9¾	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	58200	10200	68400
58.6	1629	28.65	10-0	17	8	10- 5½	10- 9¾	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17¾	12-10	61000	11100	72100
58.6	1863	31.80	10-0	17	9	10- 5½	10- 9¾	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11⅜	64700	11900	76600
58.6	2046	34.91	10-0	17	10	10- 5½	10- 9¾	9-0	2-6½	7-2	6- 6¼	12-10	18¾	11-4½	21¼	13- 1¼	68200	12800	81000
58.6	2228	38.02	10-0	17	11	10- 5½	10- 9¾	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	23⅝	13- 2¾	72100	13500	85600
58.6	2412	41.16	10-0	17	12	10- 5½	10- 9¾	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0¼	13- 4¼	76000	14200	90200
58.6	2596	44.30	10-0	17	13	10- 5½	10- 9¾	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	79900	14900	94800
62.7	1583	25.25	10-0	18	7	11- 0½	11- 1¾	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	60500	10700	71200
62.7	1778	28.36	10-0	18	8	11- 0½	11- 1¾	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17¾	12-10	63500	11600	75100
62.7	1978	31.45	10-0	18	9	11- 0½	11- 1¾	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11⅜	67400	12500	79900
62.7	2167	34.56	10-0	18	10	11- 0½	11- 1¾	9-0	2-6½	7-2	6- 6¼	12-10	18¾	11-4½	21¼	13- 1¼	71000	13300	84900
62.7	2360	37.64	10-0	18	11	11- 0½	11- 1¾	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	23⅝	13- 2¾	75100	14200	89300
62.7	2554	40.73	10-0	18	12	11- 0½	11- 1¾	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0¼	13- 4¼	79000	15100	94100
62.7	2749	43.84	10-0	18	13	11- 0½	11- 1¾	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	82900	16000	98900
66.8	1671	25.01	10-0	19	7	11- 7½	11-11⅜	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	62700	11200	73900
66.8	1877	28.10	10-0	19	8	11- 7½	11-11⅜	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17¾	12-10	65900	12200	78100
66.8	2082	31.17	10-0	19	9	11- 7½	11-11⅜	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11⅜	70100	13100	83200
66.8	2287	34.24	10-0	19	10	11- 7½	11-11⅜	9-0	2-6½	7-2	6- 6¼	12-10	18¾	11-4½	21¼	13- 1¼	74000	14000	88000
66.8	2490	37.28	10-0	19	11	11- 7½	11-11⅜	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	23⅝	13- 2¾	78000	15000	93000
66.8	2696	40.36	10-0	19	12	11- 7½	11-11⅜	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0¼	13- 4¼	82000	16000	98000
66.8	2902	43.44	10-0	19	13	11- 7½	11-11⅜	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	86000	17000	103000
70.9	1759	24.81	10-0	20	7	12- 2½	12- 6⅜	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	64200	11700	75900
70.9	1975	27.86	10-0	20	8	12- 2½	12- 6⅜	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17⅞	12-10	68400	12700	81200
70.9	2191	30.90	10-0	20	9	12- 2½	12- 6⅜	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11⅜	73600	13700	87300
70.9	2408	33.96	10-0	20	10	12- 2½	12- 6⅜	9-0	2-6½	7-2	6- 6¼	12-10	18¾	11-4½	21¼	13- 1¼	77000	14700	91700
70.9	2622	36.98	10-0	20	11	12- 2½	12- 6⅜	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	23⅝	13- 2¾	80500	15700	96200
70.9	2838	40.03	10-0	20	12	12- 2½	12- 6⅜	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0¼	13- 4¼	84000	16700	100700
70.9	3054	43.07	10-0	20	13	12- 2½	12- 6⅜	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	87500	18700	106200
75.0	1847	24.63	10-0	21	7	12- 9½	12-10¾	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	67500	12200	79700
75.0	2074	27.65	10-0	21	8	12- 9½	12-10¾	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17⅞	12-10	70700	13300	84000
75.0	2301	30.68	10-0	21	9	12- 9½	12-10¾	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11⅜	75100	14200	89300
75.0	2528	33.71	10-0	21	10	12- 9½	12-10¾	9-0	2-6½	7-2	6- 6¼	12-10	18¾	11-4½	21¼	13- 1¼	79800	15400	95200
75.0	2753	36.71	10-0	21	11	12- 9½	12-10¾	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	23⅝	13- 2¾	84100	16400	100500
75.0	2980	39.73	10-0	21	12	12- 9½	12-10¾	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0¼	13- 4¼	88400	17400	105800
75.0	3207	42.76	10-0	21	13	12- 9½	12-10¾	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	92700	18400	111100

Table 15. Babcock & Wilcox Boiler (Continued) (see Fig. 20, p. 367)

Tube 4 In. Outside Diam., 10 Ft. Long

Grate surface, sq. ft.	Heating surface, sq. ft.	Ratio	Length of tube, ft. in.	No. of sections	No. of tubes per section	Dimensions											Approx. weights		
						A, ft. in.	*B*, ft. in.	*C*, ft. in.	*D*, in.	*E*, ft. in.	*F*, ft. in.	*G*, ft. in.	*J*, in.	*K*, ft. in.	*L*, ft. in.	*V*, ft. in.	Boiler	Water	Total
79.0	1934	24.48	10-0	22	7	13- 4½	13- 5¾	9-0	2-6½	7-2	5- 1	11- 4¾	14	11-4½	16¾	12- 8½	68700	12700	81400
79.0	2173	27.51	10-0	22	8	13- 4½	13- 5¾	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17⅛	12-10	73400	13800	87200
79.0	2410	30.51	10-0	22	9	13- 4½	13- 5¾	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11 4½	19¾	12-11¾	78000	14800	92800
79.0	2649	33.53	10-0	22	10	13- 4½	13- 5¾	9-0	2-6½	7-2	6- 6¾	12-10	18¾	11-4½	21¼	13- 1¼	82600	16000	98600
79.0	2883	36.49	10-0	22	11	13- 4½	13- 5¾	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	22¾	13- 2¾	87100	17000	103100
79.0	3122	39.52	10-0	22	12	13- 4½	13- 5¾	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0½	13- 4¼	91600	18000	109600
79.0	3359	42.52	10-0	22	13	13- 4½	13- 5¾	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	96100	19000	115100
83.1	2023	24.34	10-0	23	7	13-11½	14- 0¾	9-0	2-6½	7-2	5-1	11- 4¾	14	11-4½	16⅜	12- 8½	71200	13300	84500
83.1	2271	27.33	10-0	23	8	13-11½	14- 0¾	9-0	2-6½	7-2	5- 6¾	11-10½	15½	11-4½	17⅞	12-10	75400	14500	89900
83.1	2520	30.32	10-0	23	9	13-11½	14- 0¾	9-0	2-6½	7-2	6- 0½	12- 4¼	17¼	11-4½	19⅝	12-11¾	80700	15500	96200
83.1	2768	33.31	10-0	23	10	13-11½	14- 0¾	9-0	2-6½	7-2	6- 6¾	12-10	18¾	11-4½	21½	13-1¾	85400	16800	102200
83.1	3015	36.28	10-0	23	11	13-11½	14- 0¾	9-0	2-6½	7-2	7- 0	13- 3¾	20¼	11-4½	22⅝	13- 2¾	91000	17900	108900
83.1	3264	39.28	10-0	23	12	13-11½	14- 0¾	9-0	2-6½	7-2	7- 5¾	13- 9½	21¾	11-4½	2- 0⅛	13- 4¼	96600	19000	115600
83.1	3513	42.27	10-0	23	13	13-11½	14- 0¾	9-0	2-6½	7-2	7-11½	14- 3¼	23½	11-4½	2- 1⅞	13- 6	102200	20100	122300
87.2	2111	24.21	10-0	24	7	14- 6¼	14- 7⅜	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16⅞	12- 9	73500	13900	87400
87.2	2370	27.18	10-0	24	8	14- 6¼	14- 7⅜	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18¾	12-10½	78000	15100	93100
87.2	2629	30.15	10-0	24	9	14- 6¼	14- 7⅜	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20⅛	13- 0⅛	83200	16200	99400
87.2	2902	33.28	10-0	24	10	14- 6¼	14- 7⅜	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21⅝	13- 1¾	88200	17500	105700
87.2	3146	36.08	10-0	24	11	14- 6¼	14- 7⅜	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23⅛	13- 3¼	93000	18800	111800
87.2	3405	39.05	10-0	24	12	14- 6¼	14- 7⅜	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0⅝	13- 4¾	97800	20100	117900
87.2	3665	42.03	10-0	24	13	14- 6¼	14- 7⅜	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2⅜	13- 6½	102600	21400	124000
91.3	2199	24.09	10-0	25	7	15- 1½	15- 2¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16⅞	12- 9	76700	14500	91200
91.3	2470	27.05	10-0	25	8	15- 1½	15- 2¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18⅜	12-10½	84400	15700	96100
91.3	2739	30.00	10-0	25	9	15- 1½	15- 2¾	9-0	2-7	7-0	6- 2½	12- 4¼	17½	11-4½	20⅛	13- 0⅛	85900	16800	102700
91.3	3010	32.97	10-0	25	10	15- 1½	15- 2¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21⅝	13- 1¾	90600	18300	108900
91.3	3278	35.90	10-0	25	11	15- 1½	15- 2¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23⅛	13- 3¼	96000	19500	115500
91.3	3548	38.86	10-0	25	12	15- 1½	15- 2¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0⅝	13- 4¾	101400	20700	122100
91.3	3818	41.82	10-0	25	13	15- 1½	15- 2¾	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2⅜	13- 6½	106800	21900	128700

95.4	2286	23.96	10-0	26	7	15- 8½	15- 9¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16'4	12- 9	78000	15100	93100
95.4	2568	26.92	10-0	26	8	15- 8½	15- 9¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18'6	12-10½	82700	16300	99000
95.4	2849	29.86	10-0	26	9	15- 8½	15- 9¾	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20'6	13- 0¼	88500	17500	106000
95.4	3129	32.80	10-0	26	10	15- 8½	15- 9¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21'6	13- 1¾	94000	19000	113000
95.4	3408	35.72	10-0	26	11	15- 8½	15- 9¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23'6	13- 3¼	98600	20300	118900
95.4	3690	38.68	10-0	26	12	15- 8½	15- 9¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0'6	13- 4¾	103200	21600	124800
95.4	3970	41.61	10-0	26	13	15- 8½	15- 9¾	9-0	2-7	7-0	8- 1½	14- 3¾	23½	11-4½	2- 2¾	13- 6½	107800	22900	130700
99.5	2374	23.86	10-0	27	7	16- 3½	16- 4¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16'8	12- 9	80400	15700	96100
99.5	2667	26.80	10-0	27	8	16- 3½	16- 4¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18'6	12-10½	85800	17000	102800
99.5	2958	29.73	10-0	27	9	16- 3½	16- 4¾	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20'6	13- 0¼	91300	18400	109700
99.5	3250	32.66	10-0	27	10	16- 3½	16- 4¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21'6	13- 1¾	96700	19800	116500
99.5	3539	35.57	10-0	27	11	16- 3½	16- 4¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23'6	13- 3¼	102000	21100	123100
99.5	3831	38.50	10-0	27	12	16- 3½	16- 4¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0'6	13- 4¾	107300	22400	129700
99.5	4123	41.44	10-0	27	13	16- 3½	16- 4¾	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2¾	13- 6½	112600	23700	136300
103.5	2463	23.80	10-0	28	7	16-10½	16-11¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16'8	12- 9	82800	16300	99100
103.5	2765	26.72	10-0	28	8	16-10½	16-11¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18'6	12-10½	87500	17600	105100
103.5	3068	29.64	10-0	28	9	16-10½	16-11¾	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20'6	13- 0¼	94000	19100	113100
103.5	3370	32.56	10-0	28	10	16-10½	16-11¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21'6	13- 1¾	99600	20500	120100
103.5	3671	35.47	10-0	28	11	16-10½	16-11¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23'6	13- 3¼	105000	21900	126900
103.5	3973	38.39	10-0	28	12	16-10½	16-11¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0'6	13- 4¾	110400	23300	133700
103.5	4276	41.31	10-0	28	13	16-10½	16-11¾	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2¾	13- 6½	115800	24700	139500
107.6	2551	23.71	10-0	29	7	17- 5½	17- 9¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16'8	12- 9	85200	16900	102100
107.6	2864	26.62	10-0	29	8	17- 5½	17- 9¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18'6	12-10½	90000	18200	108200
107.6	3177	29.53	10-0	29	9	17- 5½	17- 9¾	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20'6	13- 0¼	96700	19800	115500
107.6	3491	32.44	10-0	29	10	17- 5½	17- 9¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21'6	13- 1¾	102000	21200	123200
107.6	3801	35.33	10-0	29	11	17- 5½	17- 9¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23'6	13- 3¼	107000	22700	129700
107.6	4115	38.24	10-0	29	12	17- 5½	17- 9¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0'6	13- 4¾	112000	24200	136200
107.6	4429	41.16	10-0	29	13	17- 5½	17- 9¾	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2¾	13- 6½	117000	25700	142700
111.7	2638	23.62	10-0	30	7	18- 0¼	18- 1¾	9-0	2-7	7-0	5- 3	11- 4¾	14	11-4½	16'8	12- 9	87600	17400	105000
111.7	2963	26.53	10-0	30	8	18- 0¼	18- 1¾	9-0	2-7	7-0	5- 8¾	11-10½	15½	11-4½	18'6	12-10½	92500	18900	111400
111.7	3287	29.43	10-0	30	9	18- 0¼	18- 1¾	9-0	2-7	7-0	6- 2½	12- 4¼	17¼	11-4½	20'6	13- 0¼	99000	20500	119500
111.7	3611	32.33	10-0	30	10	18- 0¼	18- 1¾	9-0	2-7	7-0	6- 8¼	12-10	18¾	11-4½	21'6	13- 1¾	105000	21900	126900
111.7	3932	35.20	10-0	30	11	18- 0¼	18- 1¾	9-0	2-7	7-0	7- 2	13- 3¾	20¼	11-4½	23'6	13- 3¼	111000	23400	134400
111.7	4257	38.11	10-0	30	12	18- 0¼	18- 1¾	9-0	2-7	7-0	7- 7¾	13- 9½	21¾	11-4½	2- 0'6	13- 4¾	117000	24900	141900
111.7	4581	41.01	10-0	30	13	18- 0¼	18- 1¾	9-0	2-7	7-0	8- 1½	14- 3¼	23½	11-4½	2- 2¾	13- 6½	123000	26400	149400

Summary of General Design. Having established the steam consumption per hour and the necessary fuel to generate same, first ascertain if the space available is large enough to get in sufficient **grate surface** to burn the fuel under natural draft, with the height of funnel which may be installed.

It may be assumed that natural draft is preferable to forced draft, except in the case of Scotch boilers where the air heaters give increased evaporative economy. Blowers are objectionable if they can be avoided, being a constant source of upkeep expense and added worry and involving the use of additional fuel for operation.

If sufficient grate cannot be provided to burn the coal natural draft, fix upon the **necessary rate of combustion,** and provide forced draft. It is well to keep the combustion rate as low as possible to avoid difficulty in stoking, not over 25 lbs. of coal per square foot of grate per hour in cargo vessels, and not over 30 lbs. in fast passenger ships and 40 lbs. in war vessels.

Having fixed upon the grate surface, next select the amount of **heating surface** in the boiler necessary to absorb an adequate proportion of heat from the hot gases. The best practice prescribes an evaporation of from 4 to 6 lbs. of water per hour per square foot of heating surface, actual conditions, in the cargo trade. It may be as high as 10 lbs. in war vessels, which are called upon for maximum power only occasionally, and 15 lbs. in Destroyers and Battle Cruisers, where the ultimate power on the weight must be obtained.

The writer is aware that designers of merchant ships frequently put in boilers so small that they must be forced to 7 or 8 lbs. of water per square foot of heating surface, but in this connection it is well to note that the harder a boiler is forced, the poorer will be the boiler efficiency, and the more frequent the necessary cleaning periods and the higher the upkeep costs.

Even more important is the fact that plenty of boiler capacity makes a ship an "easy steamer," and she develops a reputation which is a great asset to the owners; it is always easy to get firemen, and the crew like the ship and stay on the job. The very opposite is the case if the boilers have to be constantly forced to the limit, and extra work by the personnel is regularly demanded.

It is obvious that the ratio of heating surface to grate surface will vary with the rate of combustion, and may run from 25 to 60 sq. ft. of heating surface to 1 sq. ft. of grate surface. Certain special conditions of boiler construction affect the exact ratio to some extent; but the controlling factors are: enough grate to burn the coal and enough heating surface to absorb the heat; and the ratio of the two is a resultant, not a primary figure. It would be folly to use a ratio of 50 or more for natural draft and a low rate of combustion; and equally foolish to use 30 to 1 in a boiler that was to operate with forced draft and a high rate of combustion. Yet these things are not infrequently done in practice.

Area of Gas Passages. The term **calorimeter,** meaning areas of gas passages, is fortunately fast becoming obsolete. It was used in the time of Watt to designate the area in square inches of the return flue through which passed all the gases of combustion. The calorimeter (in square inches) divided by the length of the flue in feet was called the **vent,** and many calculations were worked out involving these features for various "horsepowers."

Theoretically, the area of the gas passages should be determined by the velocities of the gases. Obviously, however, as the gases in their passage through the boiler setting are constantly being cooled from a temperature at the throat of the furnace (*i.e.*, the point where the gases leave the furnace or combustion chamber and enter the tube heating surface) of 2500 deg. F.

or more to a temperature of, say, 500 deg. F. at the uptake, it is impossible to maintain even approximately a uniform velocity throughout. It will be seen, also, that different rates of operation result in very great variations in gas velocity. The water-tube boilers by reason of the introduction of suitable baffling arrangements, approach the condition of uniform gas velocities much more nearly than Scotch boilers, where, plainly, the area through the return tubes remains constant while the temperature of the gases is being very considerably reduced.

It has been found that the velocity of the gases has a material effect on the transmission of heat through the metal forming the heating surfaces. However, in practice, gas velocities may change through very wide limits, with results that are considered satisfactory. The rule for stationary water-tube boilers has been given (see *Mechanical Engineers' Hand-Book—Marks*), making the area of gas passages at the throat of the furnace "10 square inches per nominal horsepower." This results in a velocity of about 1300 feet per minute at "rated capacity" when the boiler is evaporating 3½ lbs. of water F. & A. 212 deg. F. per square foot of heating surface per hour, and over 4000 feet per minute at "200 per cent above rating," equivalent to an evaporation of 10½ lbs. of water F. & A. 212 deg. F. per hour per square foot of heating surface. The rule given by Mr. Meyer (page 387), making the area through the tubes of Scotch boilers not less than 24 square inches for each square foot of grate surface, gives gas velocities in most cases considerably greater than the above, probably equalling 5000 feet per minute at the higher rates at which Scotch boilers can be forced. When it is considered that the velocity of the cool gases leaving the boiler surface may often be reduced to less than 500 feet a minute, it is evident that the gas velocities through a boiler setting vary over a very wide range, and form no very safe basis for calculation. The problem is somewhat simplified, however, by the fact that high gas velocities are easily produced, merely resulting in the necessity for greater draft pressures to force the gases through the boiler.

It is quite common to base the area of the gas passages on the grate surface, in which the "rule of thumb" may be stated as follows:

⅙ of the grate when the ratio of heating surface to grate surface is....30:1;
⅕ of the grate when the ratio of heating surface to grate surface is....40:1;
¼ of the grate when the ratio of heating surface to grate surface is....50:1.

This is approximately equivalent to **1 sq. ft. of gas area to each 200 sq. ft. of heating surface. This is a safe rule for natural draft,** but it should be borne in mind that the areas of the gas passages may be made very much less by the mere expedient of resorting to higher draft pressures.

Baffles and Retarders. The function of baffles introduced in water-tube boilers is for the double purpose of creating sufficiently high gas velocities and also for distributing the gases properly over the heating surfaces. Care should be taken to arrange these baffles so that they can be cleaned if their position is such as to allow soot to accumulate. These baffles may be made of firebrick moulded to fit the tubes and laid on the tubes in such position as to close off the gas passages, in which case they are held in position by gravity or secured by mechanical means. Cast-iron baffles moulded to fit, or wrought iron shapes, may be used if contact with the tube is sufficiently good to carry away the heat which might otherwise distort the baffle. Cast-iron is nearly always used in the cross-baffles; that is, those for directing the gases across the tubes, in which case firebrick protectors are frequently employed as a safeguard, particularly in the hottest part of the setting.

As the gas areas in fire-tube boilers are comprised in the aggregate tube area, an improved distribution of the gases and better contact with the heating surface may be secured by the introduction of so-called **retarders**, (Fig. 21), which usually consist of sheet-iron strips practically filling the interior diameter of the tube and given one or more complete twists, thus giving the gases a rotary passage through the tubes, delaying their passage and bringing them into more intimate contact with the tube surface. Two or three or, as a maximum, four convolutions in the length are used with coal, while this may be

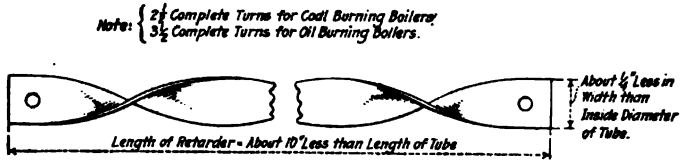

Note: 2½ Complete Turns for Coal Burning Boilers
3½ Complete Turns for Oil Burning Boilers.
About ¼" Less in Width than Inside Diameter of Tube.
Length of Retarder = About 10" Less than Length of Tube

FIG. 21.—Retarder.

increased to four, six or eight convolutions, when oil fuel is used, and where the lesser volume of gas makes a greater retarding effect desirable.

The use of retarders varies greatly in practice, both as regards number of turns and extent to which they are used in the tubes. They may be omitted entirely or they may be inserted in all tubes or in a portion only (in this case usually the upper rows). They are for obvious reasons of greater value in oil-burning boilers than in those using coal; the soot resulting from coal fires is a serious obstacle to their use, on account of the added accumulation of soot and increased difficulty of cleaning. Retarders are usually constructed of No. 14 sheet iron or steel, cut in strips $\frac{3}{16}$ in. or $\frac{1}{4}$ in. less in width than the inside diameter of the tube. After twisting, they nearly fill the tube, as the axis of the twist takes the form of a slight helix. This accounts for the fact that a strip cut to a close fit before twisting will be too large to go into the tube after twisting. The retarders are usually made about the full length of the tube or frequently 6 in. or 8 in. longer, and are provided with holes near the ends to assist in removal for cleaning.

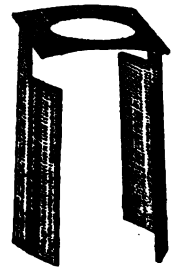


FIG. 22.—Erickson Patent Retarder.

Where a mere reduction in area at the tube ends is desired, resulting in a baffling of the gases throughout the whole or a part of the tube bank, the Erickson patent flue gas retarder shown in Fig. 22 may be used. While undoubtedly less effective than the twisted retarder in promoting the transfer of heat and increasing the boiler efficiency, it is lighter and cheaper and more easily reinserted or removed for cleaning and for these reasons has found considerable usage.

Uptake openings and funnel areas are generally made larger than necessary, on the principle that too much is always enough. Care must of course be taken not to introduce unnecessary resistance, and thus check the draft, especially for natural-draft installations. Gas velocities of 1000 to 2000 feet per minute give satisfactory results. The "rule of thumb"—making the uptake cross-section area equal to $\frac{1}{6}$th of the grate—gives a gas velocity of approximately 1000 feet per minute at 20 lbs. of coal per square foot of grate per hour.

Uptake areas combined should not be less than the funnel area, and usually are made somewhat in excess, particularly in long connections. The lead should always be upward and sharp turns and sudden changes in area should be avoided. This is particularly important in natural-draft installations.

Steam outlets should be calculated on the basis of steam velocity at maximum capacity. Too small an area causes excessive drop in pressure to the engine, while too large an area results in steam pipes being used which give excessive losses through radiation. Usually steam outlets are too large, giving steam velocities of 3000 to 5000 feet per minute. Six thousand to 10,000 feet represents good practice if the steam pipe is comparatively short and free from sharp bends. A little extra pressure drop is preferable to waste of heat through condensation. Smaller pipes and valves also represent a saving in weight and first cost.

It is not infrequently advisable to make the steam outlet the same as the safety valve outlet for structural reasons and flexibility of arrangement. For size of safety valves see page 448.

Size of Feed Connections. Feed connections are frequently made larger than necessary in marine work. The fear that in case of low water and consequent "pumping up" the velocities will become too high has probably lead to this. But it should be remembered that there is no objection to high velocity through the checks except that some increased pressure on the feed line is necessary to force the water into the boiler. On the other hand, low velocity with large checks increases the liability to water-hammer and wear and tear on the valve disc. Feed connections that are too large are therefore objectionable for engineering reasons besides the undesirable factors of increased weights and costs.

The following sizes of boiler feed connections are standard with The Babcock & Wilcox Company, and represent excellent practice. They are based on the average amount of feed delivered to the boiler in pounds avoirdupois per hour: For

Average water per hour up to 13,000 lbs., 1¼-in. valve.
Average water per hour from 13,000 lbs. to 23,000, 2-in. valve.
Average water per hour from 23,000 lbs. to 33,000, 2½-in. valve.
Average water per hour from 33,000 lbs. to 50,000, 3-in. valve.
Average water per hour over 50,000 lbs., one 3½ in. or two smaller size.

The above sizes are ample to provide for fluctuations in demand for feed water, which may at times be double the average rate. The **maximum velocity allowable through a check valve** may be fixed at about 400 feet per minute. The auxiliary connections are made the same diameter as the main feed.

Blow-off connections are often supposed to be capable of keeping a boiler clear of mud, scale and other extraneous substances introduced with the feed. The **surface blow** is, it is true, reasonably effective in removing scum or floating particles which may tend to promote priming, but except in boilers provided with large settling chambers, it is a fallacy to suppose that the **bottom blow** performs any other office effectively than to empty the boiler, or remove any excess of water above the normal steaming level. Surface and bottom blows 1 in. to 1¼ in. for all sizes of boilers are satisfactory. While larger sizes may be used, it must be kept in mind that small valves are more easily kept tight. The waste of fresh water and useful heat through leaky blow valves or excessive use of same is an unjustifiable loss.

Accessibility for cleaning inside and outside is a paramount necessity for well-designed boilers. Handhole and manhole plates should be provided wherever they promote accessibility, even at the expense of extra joints.

Cleaning doors in the casing for the use of **soot hoe** or the steam lance are provided in water-tube boilers, the former for cleaning out soot pockets or spaces over horizontal or semi-horizontal baffles. Doors which automatically close on removal of the **cleaning lance** are advisable, and it is best to make these accessible from side passages or the back of the boiler, so that "dusting" of the heating surfaces can be carried out without interfering with the firing.

The Scotch boiler can only be cleaned exteriorly from the front, from which point the tubes may be cleaned with a scraper or lance, by lifting a door in the uptake opposite the tubes. This interrupts the operation of the boiler, but it has the advantage of permitting the side passages to be greatly reduced in width. In this type of boiler, the "back connection" can be cleaned only when the boiler is out of service entirely, and cooled off.

Soot blowers which operate automatically or partially so, have not been extensively used in marine work, and are not as effective as hand-operated devices

FIG. 23.—B. & W. Soot Door.

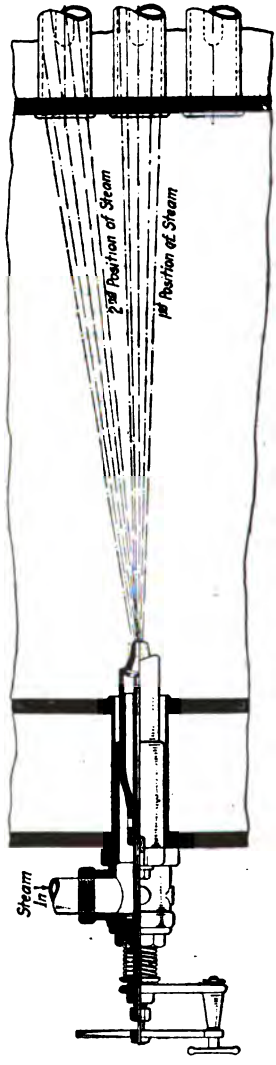


FIG. 24.—Diamond Soot Blower.

if the latter are properly handled. On the contrary, if, as usually happens, the hand-cleaning is indifferently performed, the soot-blowing outfits

have a distinct advantage. They have been proven effective and are widely used in stationary plants on shore. Certain designs of soot blowers are now being perfected which seem to overcome the difficulties of attaching same through the casings of marine boilers, and they are beginning to be introduced in marine work, where they are likely to become popular with the more extensive use of water-tube boilers.

Soot blowers are said to be effective in cleaning soot from the tubes of Scotch boilers which are fitted with superheater loops or retarders. The Diamond soot blower shown in Fig. 24 has been used for this purpose and is also being installed in connection with water-tube boilers for the Emergency Fleet Corporation. One of the features of this apparatus is the employment of metal parts treated with aluminum as a protection against oxidation at high temperatures. The method used is known as **calorising**.

Salinometer. The common, though rather unsatisfactory, method for determining the possible presence and amount of sea water in boilers on board ship, is by use of a special hydrometer, called a salinometer. As the density of the boiler water increases, owing to the presence of sea water, the instrument floats with the stem higher out of the water, enabling the operator to take readings of the salinity. These are designated as 32ds; thus, one-half of one 32d, two and one-quarter 32ds, etc.

The salinometer scale had its origin in the assumption that sea water contains 1 lb. of salt dissolved in 32 lbs. of water; thus, sea water is supposed to be $\frac{1}{32}$ salinity. This is not strictly correct, as sea water varies somewhat in saline content. (See page 1096.) However, the salinometer scale indicates that according to the number of 32ds on the scale there is a corresponding amount of salt in pounds for each 32 lbs. of water.

As the density of water is affected by its temperature, thermometer readings must be taken simultaneously with the salinity determinations. The salinometer is graduated for a certain temperature of water, as for instance, 200 deg. F. or 180 deg. F. Frequently the salinometer is fitted with two scales, as, for example, one for 200 deg. F. and one for 180 deg. F. In any event, the proper scale must be used for the proper temperature, or else proper corrections must be made.

While the use of the salinometer is very prevalent in marine practice, it is in reality a very archaic instrument, and the determinations are extremely crude. On the contrary, very accurate determinations, based on the chemical reaction of chlorine on silver nitrate, can be made which show the exact amount of sea water present in the boiler water. Several ingenius instruments have been devised for this purpose, two of which are described on pp. 469 to 473. (See Silver Nitrate Testing Kit.)

Mechanical Stokers. Increased space and weight, greater initial cost, higher cost of repairs, greater skill required in handling and difficulty of firing by hand in case of breakdown, are some of the factors which account for the almost negligible use of mechanical stokers on shipboard. The best types operating forced draft give better economy than hand firing, and are smokeless, but their greatest attraction on land—saving of labor cost—does not find the same equivalent at sea. In order to fully realize this gain, coal and ash conveyors must be used. It happens also that the semi-bituminous coals used at sea are the most suitable for firing by hand and the least suitable for mechanical firing except in stokers of the underfeed type using closed-ash-pit forced draft. On the other hand, labor troubles and the necessity for employing all possible means for improving fuel economies and lessening operating expenses in order to meet foreign competition, may result in the introduction of stokers on board ship. *Mr. R. Sanford Riley* sums up the situation excellently as follows:

"Mechanical stokers have not been generally applied with marine water-tube boilers because of space limitations aboard ship. The item of weight is also against a heavy stoker, but this is not so serious in view of the greater capacity obtainable. The skill required for operating stokers is much greater than that required for hand firing and of such a different character that a complete change in boiler-room personnel is involved. Reliability of operation has also been in favor of hand firing because the stokers thus far tried under marine conditions were subject to damage or complete shut-downs.

"Most of the stokers that have been tried aboard ship thus far have been of the single horisontal retort underfeed type. These stokers are thoroughly efficient as coal feeders but are sometimes difficult to clean. If not properly worked, clinkers are troublesome and liable to cause damage due to reverberatory action. Other stokers that have been tried aboard ship are of the chain-grate type. These are well suited for certain kinds of coal, but such coal which requires no stirring is not always available for marine use. The brick arches necessary with chain-grates are also objectionable for marine service.

"The prospects for marine stokers in the future seem to depend largely on the space that can be provided by a radical rearrangement of the boiler compartment. The boilers should be raised at least 6 feet above present standard settings and this may encroach on cargo space. However, if such a setting can be provided, there is no reason why a modern multiple retort inclined stoker should not prove just as advantageous afloat as it is ashore. Such a stoker should be arranged in an athwartship fireroom so that the incline of the fuel bed will not be seriously affected by the rolling of the vessel. The combustion chamber should be large enough to allow complete combustion before cooling the gases. Most important of all, from an operating point of view, is a spacious compartment below the stoker into which ashes may be continuously discharged. Preferably, these ashes should fall into an athwartship compartment at the opposite end of the boilers from the stoker hopper. This compartment should be roomy enough to allow a reasonable accumulation of ashes in port and also be convenient for handling the ashes for discharge. Duplicate ash ejectors or other conveyors should be provided to insure easy and continuous operation. The ashes could be withdrawn under the stoker into the fireroom and discharged in the usual way, but it is obviously better to keep the ashes clear of the coal in a separate compartment similar to modern shore practice.

"Automatic coal handling means should be provided to suit the bunker arrangement of the ship and avoid shoveling from floor plates to stoker hoppers.

"With such modern stokers properly arranged, the capacity, economy and reliability should be comparable with that commonly obtained in public utility plants ashore. Reliability and freedom from serious damage is assured in this type of stoker, especially with those which are provided with safety devices which restrict any trouble or interruptions to one retort and do not put the entire stoker out of commission.

"Such an installation of modern stokers would not only increase the efficiency now obtainable from hand firing, but would greatly reduce fireroom labor. The reduction in labor would be especially marked in high-powered passenger vessels.

"With ample space obtained, by an arrangement as above, there seems no valid reason why stokers should not replace hand firing afloat just as they have ashore."

Automatic feed-water regulators of various types have been designed which, when kept in good order, control the admission of feed water to the boiler much better than can usually be done by hand. There is a danger in the employment of this apparatus inherent in all automatic devices, namely, the operating personnel is likely to come to depend entirely upon their continuous and accurate operation without realising the necessity for constant watching and frequent overhaul. This in connection with boiler feed, introduces possibilities which may become disastrous. If carefully watched and cared for, however, automatic feed regulators are an advantage with any type of marine boiler, including Scotch boilers as well as the small-tube express types which contain a very small amount of water per square foot of heating surface.

These regulators are of three classes: first, those controlled by a float in a separate chamber outside the boiler shell; second, those controlled by the expansion of metal alternately heated and cooled as the level of the water in

the boiler fluctuates; and third, those controlled by a float within the boiler shell. While each of these classes possesses individual merit, accessibility for overhaul and repair should be the controlling factor in selecting the design.

By utilising the fact that condensation of steam in a chamber exterior to the boiler will cause a vacuum therein, the rise and fall of the water level in the type of regulators fitted with a chamber exterior to the boiler may be very greatly accentuated, so that the operation of the device is not merely dependent on the rise and fall of water in the boiler proper, the latter being utilised merely as a means for controlling the opening and closing of a valve admitting steam to the external chamber. In apparatus of this character, the multiplying effect results in maintaining a very steady water level. A device of this kind is illustrated in Fig. 25, which operates as follows:

When water in boiler is low, *A* is exposed and fills chamber *C* with steam. Displacement *J* lowers, and closes valve *H* against steam pressure admitted through *D* and opening exhaust *E*, placing piston *G* under atmospheric pressure. This allows feed valve to open. As water level in boiler rises, *A* is submerged, steam is closed off of chamber *C* a partial vacuum is formed by the condensation of steam therein and water rises in the chamber lifting displacement *J* and operating valve *H* to close exhaust *H* to atmosphere and opening steam connection through *D*, placing *G* under steam pressure and closing feed valve.

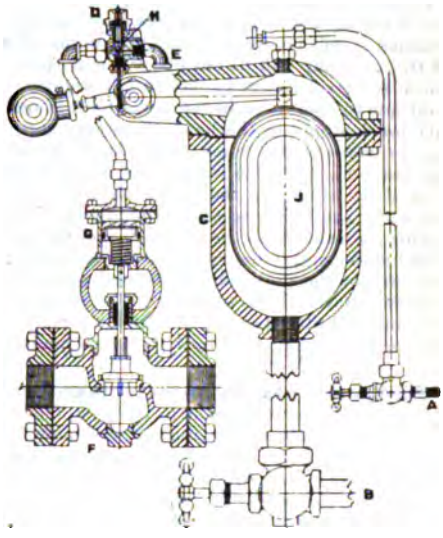


FIG. 25.—Vigilant Feed Water Regulator.

A, connection to boiler at the desired water level;
B, connection to boiler for flooding displacement chamber *C* when *A* is submerged;
D, steam connection to boiler;
E, exhaust to atmosphere;
F, feed control valve, usually consisting of a by-pass around main feed stop valve;
G, piston for operating valve *F*;
H, double seated valve for alternately placing *G* under steam pressure or atmospheric pressure;
J, displacement for operating valve *H*.

The essential requirement in the satisfactory operation of feed-water regulators is a steady water pressure on the feed line, maintained at a satisfactory margin above the boiler pressure, say, 50 lbs. The feed pump, therefore, should be fitted with a controller which will be arranged to maintain a constant pressure on the feed line regardless of the amount of water in the hot-well or filter-box. This reverses the very common practice of controlling the movement of the feed pump in accordance with the water level in the hot-well, the common float arrangement being removed and a suitable pump governor used instead. This involves another modification of usual practice, namely, as the operation of the pump

no longer is made dependent upon the supply of water in the hot-well, the latter must be increased in capacity sufficiently to allow a sufficient supply of feed for the pump at all times and prevent the waste of fresh water by overflowing the hot-well when the feed pump shuts down. With a feed-water regulator and a pump governor controlled by pressure on the feed line, the pump will be moving so long as the level in the boilers is below the normal level, and will be shut down as soon as the level in the boilers reaches that point. On shutting down the main engines, it will be evident that the supply of water in the hot-well by means of the air pump will be cut off; while at the same time checking the fires will cause the level in the boilers to fall slightly. The feed-water regulator will continue to operate until the water level is again pumped up; hence the necessity for a supply in the hot-well to take care of this condition. On starting up, a reverse action takes place; the air pump begins to discharge water into the hot-well, while the water level in the boiler rises slightly under the influence of the firing, but the feed-water regulator does not begin to operate until the water again falls to the normal level. During this time the feed pump is taking no water from the hot-well, and the latter must be sufficiently large to care for the supply delivered by the air pump during this short period. The capacity of the hot-well will depend primarily on the water capacity of the boilers. Barring this one feature of starting up and stopping, the ordinary hot-well and filter-box will be satisfactory; and it is quite possible to shut off the regulators and feed by hand if an increase in size of the hot-well is not permissible.

Feed-water heaters in the uptake, by which the feed water is heated by means of the waste heat in the escaping gases, have long been recognized as adding greatly to the thermal efficiency of the plant. Owing, however, to the excessive corrosion which it has been found takes place in these heaters, they have not been widely used. When composed of steel or iron the metal is rapidly attacked on the water side, while, if brass or other composition is used, the corrosion takes place on the side exposed to the gases. The location in the uptake of necessity renders this type of feed-water heater difficult of access for cleaning purposes and repair, and a leak, once started, soon becomes disastrous in its effect on the uptake plates and the boiler heating surface. A further objection lies in the fact that the cooling of the gases escaping to the funnel reduces the draft suction, and, to be fully effective, feed-water heaters of this character require the use of forced-draft fans. The value of uptake feed-water heaters is also diminished by reason of the fact that in all modern installations the feed water from the hot-well is passed through a heater in which the auxiliary exhaust steam heats the feed water well above 200 deg F., so that the feed water enters the uptake heaters at the latter temperature.

Notwithstanding all the above objections, there is still a considerable inducement on the economic side for the employment of feed-water heaters which return a portion of the waste heat in the gases to the boiler, and it is likely that the difficulties heretofore encountered will eventually be overcome and uptake feed-water heaters will be used on shipboard.

Live steam purifiers and feed-water heaters of the type in which the feed water on its way to the boilers comes in direct contact with live steam, have been found effective in removing scale-making material such as sulphate and carbonate of lime in installations operating on fresh-water lakes and rivers where jet condensers are used and the feed water contains these impurities. Heaters of this type are usually constructed of steel in the form of

a cylindrical shell subject to the full boiler pressure, with steel-plate trays or pans arranged in the interior thereof, in such a way as to permit the water, after having been brought in contact with steam at boiler pressure and thus raised to the boiling temperature, to flow slowly over these flat surfaces and thus deposit the salts which are precipitated at this comparatively high temperature. The water having been thus purified, enters the boiler, leaving the impurities in the trays of the heaters. The heaters are periodically opened and the deposits in the trays removed by mechanical means. A most essential point to observe in selecting a purifier of this type, is to have it of very large capacity; in fact, where these heaters have failed, it has generally been due to their having been designed to handle only the exact amount of water evaporated per hour by the boiler, so that in case it becomes necessary to pump the water through the heaters at a higher rate than this average, as inevitably happens with hand regulation of the feed, the salts which had previously been deposited in the trays are swept through the purifiers into the boiler.

While purifiers of this type have not been favored in salt-water service, there is some reason to believe that the mere heating of the feed water up to the boiling point before it enters the boilers, promotes the circulation in the boiler with resulting improvement in economy, and prevention of strains due to unequal expansion. This field may find advocates in the further development of marine engineering when increasing competition requires greater effort to attain maximum efficiency.

THE DESIGN OF SCOTCH MARINE BOILERS

The following notes on Scotch boiler design and the tables of sizes and weights given on p. 362 and riveted seams, p. 397, were prepared especially for this article by *Henry C. E. Meyer, Assistant to the Chief Surveyor of the American Bureau of Shipping.*

Classification

The many different designs of Scotch boilers can be subdivided into two general classes, which, while varying in minor details, are alike in principle; a great many of the differences in designs are largely due to local conditions of manufacture or the individual opinion of the designer.

The two **main classes** are: (1) **Single ended boilers,** which are fired from one end only, and in which the combustion chamber or chambers are separated from the back head by a water space; (2) **double ended boilers** which consist practically of two single ended boilers placed back to back with the back heads omitted. While for large passenger steamers, where the number of boilers required is great and space limited, the double ended boiler is still constructed, the same has inherent faults that have caused it to fall into desuetude, and the **single ended boiler is the one in most common use** for freight vessels and vessels where weight is not the most important factor affecting design.

Single ended boilers may in general again be subdivided into two general types as follows:

1. **Common Combustion Chamber Type,** where the furnaces all meet in one common back or combustion chamber. This type, on account of its simplicity of construction, has been widely adopted for tugs and coastwise steamers and donkey boilers for large vessels. However, the interior cannot be kept clean very readily and the application of forced draft is difficult and this boiler is therefore seldom used under forced draft conditions. The common combustion chamber type of boiler has apparently found greater favor in the United States than abroad, probably due to the fact that it is simple to

construct and that with proper attention this boiler has given wonderful service in coastwise vessels and small harbor craft. (See Fig. 17.)

2. **Separate Combustion Chamber Type**, where a separate back or combustion chamber is provided for each furnace. This type is more commonly used for ocean-going vessels and is used almost exclusively in European practice. The separate combustion chamber type of boiler has distinct **advantages**. First, its accessibility is the result of the necessity for providing sufficient water space between the sides of the separate combustion chambers. This feature results in leaving wide spaces between the separate nests of tubes and between the nests of tubes and the shell, thereby providing spaces through which access can be had to the tops of the furnaces without difficulty. Second, the separate combustion chamber type of boiler renders itself more readily to the application of forced draft. Its **disadvantage** is the fact that it is considerably more difficult and expensive to construct than the common combustion chamber type. (See Fig. 18.)

Double Ended Boilers are not used very extensively except in the case of vessels of large power. In a vessel which would require six single ended boilers, the substitution of three double ended boilers would mean a loss of power of about 33 per cent in the case of the breakdown of one boiler, whereas only a loss of 15 to 16 per cent would result if one of the six single ended boilers were lost. Furthermore, by having a comparatively large number of single ended boilers one can be put out of commission en route for cleaning without seriously affecting the speed of the vessel; also one of the main boilers can generally be used in port as a donkey boiler.

Double ended boilers, where **forced draft** is used, are always constructed with separate combustion chambers for each furnace. With **natural draft** it is common practice to have one combustion chamber for each two opposing furnaces. This construction is not feasible for forced draft systems since when a furnace door is opened flame would be forced into the fireroom from the pressure in the opposite furnace.

Relations between Heating and Grate Surface. In the best modern practice ratio of heating to grate surface for cylindrical marine boilers lies between the limits of 30 and 40 to 1 for natural draft and between 40 and 48 to 1 for forced draft.

Diameter and Length of Boilers. Having determined upon the grate surface and heating surface, the diameter and length of boiler may be obtained by reference to the tables on p. 362. These tables are based upon the proportions of boilers which are actually in successful operation, and are of value in determining a new design.

The **length of the boiler** may range from 10 ft. to 12 ft. 6 in. and while, up to quite recently, the most serious limitation to the length of a cylindrical boiler was the difficulty of obtaining plates wide enough to permit of making the shell in one course, another factor tending to restrict the boiler length is the fact that while increased length may mean increased heating surface it does not increase the area through the tubes, and for this reason boilers within the above mentioned dimensions will give a better balanced design for natural draft purposes.

Now that it has become possible to obtain plates of much greater width in the United States, there may be a tendency to increase the length of the boiler, and while this may result in some economy, if not carried to excess, the necessity of maintaining proper areas through the tubes for obtaining satisfactory draft should not be overlooked.

The area through the tubes per square foot of grate surface should not be less than 24 sq. in.

The **diameter of the boiler** may range from as small as 8 ft. to as large as 18 ft., and will depend to a large extent upon the grate surface required since for particular types of boilers the diameter of the furnace should not exceed given proportions.

The **length of the grate bars** is limited to that length which can be properly fired. For **natural draft** this should not exceed 6 ft. 6 in.; 6 ft. 3 in. is still better. For **forced draft** the customary length of grate bars is 5 ft. 6 in.

Number of furnaces to each boiler; 2 for diameters up to 12 ft. 6 in.; 3 for diameters from 12 ft. to 16 ft. 3 in. incl.; 4 for diameters from 16 ft. 0 in. up.

The **diameter of furnaces** (maximum outside) in inches may be obtained from Table 16.

Table 16. Diameter of Furnaces

Number of furnaces	Common or separate combustion chambers	Diam. of furnace	Number of furnaces	Common or separate combustion chambers	Diam. of furnace
2	Common	$\frac{D}{3} + 3$	3	Separate	$\frac{D}{3} - 13$
3	Common	$\frac{D}{3} - 8$	4	Separate	$\frac{D}{3} - 22$

Tubes. The outside **diameter** of tubes may vary from 2½ in. to 3¼ in. It is, however, general practice to use 2½-in. tubes for **forced draft** boilers and 3-in. tubes for **natural draft** boilers.

Whereas the greater part of the heating surface of the boiler consists of the tubes, this constitutes the least efficient part. Although ample heating surface is necessary for the economical performance of the boiler, nevertheless, many boilers have been ruined by attempting to crowd too much heating surface into a limited space with the result that the boilers primed, were difficult to clean, and generally did not perform as well as they might have done had some of the so-called heating surface been sacrificed for better distribution of tubes, etc.

Tubes should never be **spaced** closer than 1¼ in. between the outside of tubes horizontally or closer than 1 in. between the outside of tubes vertically. The first figure is sometimes decreased to 1¼ in., but the resulting advantages are extremely doubtful and, provided the boilers are designed with a reasonable margin of heating surface, the additional tubes which may be placed in a boiler by spacing them ¼ of an inch closer together are negligible.

The amount of steam space required in Scotch boilers varies somewhat with conditions but here again the best practice follows along fairly definitely established lines. The following formulæ will give locations for the upper row of tubes, which if adhered to will positively assure sufficient steam room: Distance in inches from centre line of boiler to centre line of upper row of tubes: For boilers with common combustion chambers = 0.2 × diam. of boiler; for boilers with separate combustion chambers = 0.166 × diam. of boiler. For tugs and coastwise vessels which are not subject to quite as

heavy weather as ocean-going vessels, the steam space may be somewhat smaller.

Distance between Furnaces. Where the front ends of the furnaces are left cylindrical the minimum distance between two furnaces on the front head should not be less than 4½ in. and in general the distance between centres of furnaces, should be:

Outside diameter in inches + 5 in. for common combustion chambers.
Outside diameter in inches + 7 in. for separate combustion chambers.

Spaces between tube nests should in no case be less than 10 in. In designing boilers it is well to keep in mind that a boiler which can be easily inspected will be kept in better condition than one which requires the removal of stays for cleaning purposes.

Shell Plates. The thickness of the shell plates is determined from the formulæ of the U. S. Steamboat Inspection Service and the American Bureau of Shipping which are given on pp. 409 and 410 (unless the rules of another classification society are specified).

In this connection it will be noted that the efficiency of the riveted joints is of great importance in determining the thickness of the shell. Tables of riveted joints which will meet the requirements of both the above agencies are given on pp. 397 to 402 and while the efficiency of the rivets as calculated by the rules of the American Bureau of Shipping would be considerably higher than indicated, these riveted joints are in accordance with the average best practice both here and abroad. Generally speaking it is an advantage to have the efficiency of the rivets somewhat higher than that of the plate, and it is not advisable to either increase the pitch of the rivets greatly beyond the pitches indicated nor decrease the diameter of the rivets, since the economy resulting from same would be negligible and the proportions indicated have given satisfactory results in the past.

The **circumferential seams** of boilers should never be less than double riveted. While on the Great Lakes it is quite common to make the seams connecting the shell to the heads single riveted, this practice is unknown in boilers for vessels trading in salt water and is not to be recommended. While single riveting can be made sufficiently strong for the purpose it does not leave any margin for deterioration, chipping and recaulking, and double riveting these seams for ⅔ of the circumference and treble riveting the lower third of the circumference represents the best existing practice.

Circumferential seams **in the middle length of the boiler** when the shell consists of two or more rims should always be double riveted and in the case of double ended boilers treble riveted. In single ended boilers it is only rarely that the shell need be made in two courses, and a single course is always to be preferred. There is always more or less trouble experienced with the middle seams owing to unequal expansion of the top and bottom of the boiler and this trouble can be eliminated by the use of a single course for the shell.

Flat Surfaces subject to pressure must be supported against collapse by means of stays. These surfaces include heads, combustion chamber backs, tube sheets, wraps and tops.

The thicknesses of plates forming flat surfaces are determined from the rules on pp. 410 to 413, and are dependent on the methods of staying and the pitch of the stays.

Each flat surface should be treated independently. In staying surfaces care must be exercised to so arrange the stays that local severe bending stresses are not set up with consequent damage to the boiler. This condition

sometimes results from staying flat surfaces without regard to conditions of expansion and contraction.

Stays. The diameter of stays can be determined from the rules on p. 414. In general there are three principal types of stays as follows: (1) Steam space stays; (2) screwed stays; and (3) portable stays. (See Fig. 54, p. 430.) **Steam space stays** are generally made of tested steel bars. They extend from end to end of the boiler, passing through the heads and are fitted with nuts and washers on both sides of the plate. **Screwed stays** are used for the support of the flat surfaces of combustion chambers and are tapped through both the plate supported and the supporting plate with a continuous thread. In some cases nuts are fitted at both ends, in others the ends are riveted over. The latter method has the advantage of cheapness and furthermore is not so liable to deterioration from the effect of the fire in the combustion chamber. For supporting flat surfaces between the furnaces and tubes, **portable stays** of varying types are used.

Combustion chamber tops are generally supported by means of **girders** which transmit the load to the vertical tube sheet and back plate of the combustion chamber. The dimensions of these girders are covered by the rules on p. 420, as are the permissible compressive stresses on the small ligaments of the tube sheets between adjacent rows of tubes.

Mr. Charles F. Bailey gives the following notes on Scotch boilers in the paper on Marine Boilers, presented by him at the *International Engineering Congress*, San Francisco, in 1915:

"The changes during the past decade in the Scotch type of boiler and other fire-tube boilers have been less than in most forms of water-tube boilers.

"We note simplification in design due to the modern rolling-mill facilities for supplying larger plates, thus reducing the number of seams. It is usual now to construct single-end Scotch boilers up to 12 feet (3660 mm.) in length and nearly 18 feet (5486 mm.) in diameter with no middle circumferential seam and with only one seam across the back head. Separate combustion chambers are usually fitted in single-end boilers and also in double-end boilers in connection with forced draft. With natural draft or assisted draft, in double-end boilers, it is not unusual to connect the corresponding furnaces in opposite ends of the boiler to straight through combustion chambers; this considerably reduces the length and weight and simplifies the construction. When such combustion chambers are fitted, firebrick walls are sometimes built up in the middle between the furnaces.

"The combustion chambers for double-end boilers are usually tied to the bottom of the boilers to prevent working when steaming, although with single-end boilers it is not essential to tie the chambers down.

"The lower wrapper plates of the combustion chambers are sometimes made in one plate of greater thickness at the bottom part where unsupported by stays, thus saving the side riveted laps.

"The question of material of boiler tubes is one on which there is a considerable difference of opinion. The best charcoal iron tubes and seamless drawn steel tubes seem under favorable conditions to give equally satisfactory results. The use of Serve tubes in Scotch boilers has practically been discontinued. With water-tube boilers it is common to use seamless drawn steel tubes.

"It is now usual to fit the back ends of the corrugated furnaces either of the horse-collar type, the straight type or of some form of flanged type which permits of the furnaces being renewed at a minimum expense.

"It is not uncommon to fit Scotch boilers of very large dimensions. The

boilers of the S.S. 'Aquitania' are 17 ft. 8 in. (5385 mm.) mean diameter by 22 ft. 0 in. (6706 mm.) mean length.

"Some owners take precautions to prevent air drafts from circulating at the bottoms of their boilers, and lag the bottoms with asbestos or magnesia blankets or pads which are easily removed for examination. This is an excellent practice, promoting economy in operation and upkeep.

"The ordinary plate saddle or support is largely used for Scotch boilers in conjunction with fore and aft chocks. A form of Scotch boiler support which is frequently adopted by American builders is illustrated in Fig. 14, page 349. These supports consist of heavy bent steel lug plates riveted to the lower shell of the boiler, each lug being fitted with a ¼-in. (6.3 mm.) calking plate adjacent to the shell. These plates can be hydraulically riveted when the shell riveting is being done. Girders are fitted in the ship, as indicated in Fig. 14, to which the lug plates are bolted. Provision is made at one end of the boiler for expansion. All boiler supports should provide the greatest facility possible for care of the lower parts of the boiler. It may be noted that the heating surface per cubic foot volume of boilers has not altered materially, but the tendency is to allow more steam space, which gives better access for cleaning and care."

RULES AND REGULATIONS GOVERNING BOILER CONSTRUCTION

Classification Societies

As a safeguard against accident on board ship due to faulty boiler construction, the United States government and the governments of Great Britain and other countries have established rules for the inspection of material and construction of marine boilers. These are supplemented by the Rules of the so-called Classification Societies, early established for the purpose of making surveys and issuing certificates of classification for vessels for standardising and regulating marine insurance. The rules of these societies while originally applying to the construction of the hulls of vessels only have, since steam propulsion came into vogue and working pressures increased, been extended to govern the construction of the propelling machinery, including the boilers.

In the United States, the boilers for all steam-driven craft which are self-propelled, except those of the U. S. Navy and other government departments, are required to comply with the Rules of the **U. S. Board of Supervising Inspectors of Steam Vessels.** Copies of these Rules may be obtained on application to the Local Inspectors of Steam Vessels in the various Districts or to the Department of Commerce, Washington, D. C. All merchant vessels flying the flag of the United States, whether of foreign or domestic construction, are required to comply with these Rules.

The only American Classification Society is the **American Bureau of Shipping.** The standards of this Society are accepted by the United States government, as indicated by the following extract from the Rules of the U. S. Board of Supervising Inspectors of Steam Vessels:

"In the inspection of hulls, boilers, and machinery of vessels, the rules promulgated by the American Bureau of Shipping respecting material and construction of hulls, boilers, and machinery, and the certificate of classification referring thereto, except when otherwise provided for by these rules and regulations, shall be accepted as standard by the Inspectors of this service."

In Great Britain, the government rules are represented by "Instructions as to the survey of passenger steamships" issued by the **British Board of Trade.** These rules, while primarily intended for the regulation of passenger

vessels, may be applied as regards safe working pressures, etc., in cases of accident to cargo ships or possible conversion of cargo craft into passenger-carrying vessels.

The two Classification Societies in Great Britain are **Lloyd's Register of Shipping** and the **British Corporation** for the Survey and Registry of Shipping. The rules for boilers of the British Corporation are in general identical with those of the American Bureau of Shipping; and the boiler rules of Lloyd's Register of Shipping do not differ to a great extent.

In Canada, the **Department of Marine** issues "Rules for the Inspection of Boilers and Machinery of Steamboats." These are based on the rules of the British Board of Trade.

Other Classification Societies in foreign countries are as follows: France, **Bureau Veritas**; Italy, **Registro Navale Italiano**; Norway, **Norske Veritas.** Most of the Classification Societies have representatives in other countries. In the United States, however, the American Bureau of Shipping represents the British Corporation and the Italian society.

While amendments have been made to the boiler rules of the various governments from time to time, it is extremely desirable that a general revision should be undertaken and a strong effort made to reconcile the comparatively slight difference now existing. This applies with even greater force to the Classification Societies, where the actual passage of new laws is not necessary. It is to be noted in this connection that the boiler rules of the American Bureau of Shipping, the British Corporation and the Registro Navale Italiano are identical. It not infrequently happens that two or more agencies have the supervision of the construction of the machinery for the same vessel. It appears, therefore highly ridiculous that boilers built under rules which have proven, by long years of satisfactory service, in all respects satisfactory, should be unacceptable to some Classification Society, or acceptable only for lower steam pressures, on account of the fact that the rules of said Society require, for example, another 32d of an inch in the thickness of a drum head, or that boilers satisfactory and in common use on one side of a lake or river are not acceptable on the other.

The necessity for uniform rules for the construction of stationary boilers has already been recognized by the American Society of Mechanical Engineers. A committee of representative engineers and boilermakers of the country has been appointed which has, after thorough study of the subject, agreed upon certain rules embodied in the **A.S.M.E. Boiler Code.** This has been approved by that Society and adopted by many of our State governments, and is recognised as the standard authority in stationary boiler practice. Already, boilers built of material specified under the A.S.M.E. Boiler Code have been accepted by the United States Navy.

It is believed that marine engineers will recognise the necessity and value of similar action, and that steps will be taken toward the desired end. However, it must be emphasized that in order to be effective, not only must all the Classification Societies agree among themselves, but their rules, and the rules of the various government agencies, which, while different in form do not differ very greatly in effective results, must be brought into harmony. It is believed that the U. S. Supervising Inspectors will co-operate to accomplish this result and that uniform rules for the construction of marine boilers will be endorsed by government action.

FACTOR OF SAFETY

The safe working pressure allowable on a boiler may be arrived at in two different ways:

The first is by assuming a factor of safety deemed sufficiently high to cover such contingencies as poor design, poor workmanship and subsequent thinning of the plate by corrosion. This method is followed by the **U. S. Supervising Inspectors of Steam Vessels** in their rule for cylindrical shells. A factor of safety of 6 is used for shells having longitudinal seams single riveted, while a factor of safety of 5 is allowed if the longitudinal seams are double riveted, and provided that the rivet holes have been fairly drilled and not

punched. The feature of design of seams is sufficiently well covered by specific rules for size, number and pitch of rivets; and deterioration is further allowed for by the requirement that when a boiler is ten years old, and whenever thereafter the Local Inspector deems necessary, the thickness of the plate shall be determined by drilling, and the actual thickness used in calculating the working pressure, which may be lowered accordingly. It will be observed that the efficiency of the riveted joint is not considered, other than that it is assumed that the double-riveted joint is 20 per cent more efficient than the single-riveted joint.

The second method is exemplified by the rule of the **British Board of Trade**, wherein a basic factor of safety of 4½ is prescribed for boilers which have the rivet holes drilled in place, and all longitudinal seams fitted with double-riveted double buttstraps, and provided that the boilers have been open to inspection during the period of construction. Any departure from this standard as applying to the riveting, requires a substantial increase in the factor of safety. These additions may increase the factor of safety to 7 or more, according to the conditions. The factor of safety so found is applied in a separate formula for the working pressure, which allows for the efficiency of the riveted joint; and it is further required that the efficiency of the joint shall be worked out assuming that the plate may tear or the rivets may shear, the lowest percentage of efficiency to be used in the calculation.

The Classification Societies, as will be noted from the formulæ for cylindrical shells, p. 408, not only make allowance for the character of the riveted seam and the efficiency of the joint, but go even further, and make an allowance in the calculations for new boilers for a subsequent thinning of the plates by corrosion. Thus, instead of basing the calculation for strength on the actual thickness of the plate, the American Bureau of Shipping subtracts $\frac{1}{16}$ in. from the thickness of the plate, Lloyd's rule makes a deduction of $\frac{1}{8}$ in., while the Bureau Veritas rule reduces the thickness of the plate by 0.04 in.

As an offset to this arbitrary allowance for deterioration, the factor of safety itself in these formulæ is reduced to about 4. It would seem, in view of the fact that the percentage efficiency of the joint is used in the original formula (amounting usually to about 80 per cent), the main shell plates might be reduced by a corresponding amount before the strength (as figured by the formula) would be affected. However, the principle of making a stated allowance in thickness for subsequent corrosion is a logical one, and is looked upon with favor by those who are studying these problems.

It is to be noted that in all calculations of the strength of boilers and the efficiency of riveted joints, the lowest tensile strength of any of the plates used in the construction, as determined by mill tests shall be employed.

HYDRAULIC TESTS

The **U. S. Supervising Inspectors** require that a hydraulic test pressure shall be applied to all boilers, the test pressure to be 1.5 times the working pressure for cylindrical fire-tube boilers and 2 times the working pressure for water-tube boilers. The **Board of Trade** requires 2 times the working pressure on new boilers and 1.5 to 1.75 times the working pressure on old boilers on which repairs have been made. The **American Bureau of Shipping** requires a hydraulic test pressure 1.5 times the intended working pressure for all boilers. **Lloyd's** requires the test pressure to be 2 times the working pressure for all boilers. The **Bureau Veritas** requires the test pressure to be 2 times the working pressure for all boilers, provided the test pressure does not exceed the working pressure by more than 142 pounds.

There appears to be no good reason why a thing which is inherently strong should be tested more severely than a weaker structure; and if, as sometimes happens, test pressures must be repeatedly applied on account of small leaks, a very high test pressure may approach the character of a menace. The successful application of the Supervising Inspectors' rule for cylindrical boilers would indicate that pressures of more than 50 per cent above the working pressure are unnecessary, if not unwarranted.

RIVETED JOINTS

Two things must be provided for in the riveted joint, both of equal importance: (1) **Strength against rupture**; (2) **Tightness against leakage.**

The **strength of the joint is expressed as the percentage of strength of the solid plate.** It is well known, however, that a joint may be designed which when analyzed will show a very high percentage of strength, but which, owing to the spaces between rivets in the outer row, can not be made tight. This condition is exaggerated with thin plates, which obviously require a closer spacing of rivets than thick plates, to prevent distortion of the plate between the rivet holes under the effects of the internal pressure.

In calculating the strength of a riveted joint, it is assumed that every rivet completely and exactly fits the rivet hole, and that the holes in the plates are exactly in line so that there is no offset or reduction in the stock of the rivet subject to shear. While the best workmanship, coupled with the requirement that rivet holes must be drilled after the plates are formed, does insure this condition, it doubtless often happens that some deviation exists, resulting in an impairment of the designed strength. On the other hand, the resistance due to friction between the plates and under the rivet heads is in this country ignored in the design, and this factor is a very important one in adding to the strength and tightness of the joint. It doubtless offsets any usual losses due to faulty riveting.

In **estimating the strength of the joint,** a section through the plate and rivets on the pitch line of each row of rivets, and also various combinations, is calculated and the weakest section taken as the strength of the seam. It is of course the object to approximately balance the aggregate tensile strength of the ligaments of the plate between rivet holes with the aggregate shearing strength of the rivets; however, the efficiency of the rivets in the joint is usually made slightly higher than that of the plate as the shearing strength of the driven rivet involves more elements of uncertainty than the tensile strength of the plate ligaments. (See pp. 397 to 402.)

Where **rivets are in double shear,** it is customary to assume the shearing strength as equal to 1¾ or 1⅞ times the strength of the rivet in single shear. There seems to be no good reason why the strength of a rivet in double shear should not be twice the strength of a rivet in single shear; and it is probable that destructive tests of longitudinal lap seams which have shown the ultimate strength of rivets in single shear to be greater than half those in double shear (double buttstrap seams) have, through bending of the plate at the joint, put some degree of tension on the rivets, thus resulting in increasing resistance to rupture. It is possible, however, that rivets in double shear may be to some extent affected by the crushing action of the plate on the side of the rivet; and the custom of estimating the double-shear strength as less than twice that of single shear tends to offset the effect of this crushing action.

The **U. S. Supervising Inspectors** give the following **rules for riveted joints,** in which

p = The greatest pitch of rivets in inches.
n = Number of rivets in one pitch.
p_d = Diagonal pitch in inches.
d = Diam. of rivet holes in inches.
T = Thickness of plate in inches.
V = Distance between rows of rivets in inches.
E = Distance from edge of plate to centre of rivet in inches.

Iron plates and iron rivets: $p = \frac{d^2 \times 0.7854 \times n}{T} + d.$

Steel plates and steel rivets: $p = \frac{23 \times d^2 \times 0.7854 \times n}{28 \times T} + d.$

This follows the English practice of making the ratio of tensile strength of steel plates to the shearing strength of the rivets as 28 long tons to 23 long tons per square inch

Distance from centre of rivet to edge of lap: $E = \frac{3 \times d}{2}$.

Distance between rows of rivets for chain riveted joints, that is, where the rivet holes in adjacent rows come opposite to each other (not greatly used in modern design);

V not less than 2 × d; preferably not less than $\frac{4d + 1}{2}$.

Diagonal pitch, for double zigzag riveted lap joint. This is the common design where the rivet holes in one row come opposite the spaces in the adjacent row: $p_d = \frac{6p + 4d}{10}$.

Maximum pitch for riveted **lap joints:**

Single riveted, $p = (1.31 \times T) + 1\frac{1}{8}$.
Double riveted, $p = (2.62 \times T) \times 1\frac{1}{8}$.

Diameter of Rivets. The **Board of Trade** requires that the diameter of rivets shall not be less than the thickness of the plates; and that with lap joints or single buttstraps they shall exceed the thickness of the plates. This requirement is not insisted upon by the **Supervising Inspectors,** but it is found in practice that the joints generally work out to better advantage when this condition exists. Usually the diameter of the rivet is made considerably greater than the thickness of the plate, when relatively thin plates are used, but in plates over 1¼ inches in thickness the rivet diameter may be about equal or even slightly less than the thickness of the plate without departing from good practice.

Circumferential Seams. Single and double-riveted lap joints are universally employed for circumferential seams. The stress per sq. in. for the same working pressure is one-half that of the longitudinal seams.

Buttstraps. All first-class marine work calls for **double buttstrap joints** on the longitudinal seams. The **U. S. Supervising Inspectors'** rule requires that the **thickness** of each of the **double buttstraps** shall be **not less than 5/8 the thickness of the shell-plates.** This is the same as the rule of the Board of Trade. Where **single buttstraps** are used, the **Supervising Inspectors** require that the **buttstraps** shall be **at least equal to the thickness of the shell-plate;** while the Board of Trade requires the thickness to be 1¼ times the thickness of the plate. It is to be noted that in double buttstrap joints, those which have **buttstraps of equal width** show usually a higher efficiency than those having buttstraps of unequal width.

It will be observed in this connection that the Classification Societies allow a higher working pressure on boilers fitted with double buttstraps of equal width. However, this all refers to strength of joint, and takes no cognisance of the fact that the joint must also be made tight against leakage. Usually no trouble is experienced with

thick plates but it frequently happens, with thin plates, that joints designed with the inside buttstrap wider than the outside strap will insure the possibility of making a tight joint by calking; whereas if buttstraps of equal width were used with wide pitch of rivets in the outer row, the stiffness of the plates between the rivets would not be sufficient to withstand the pressure of the calking, and the joint could not be made tight.

Examples of Riveted Joints. Fig. 26 shows an example of a seam used in Babcock & Wilcox boilers where the diameter of the drum does not exceed 42 in. or 48 in.

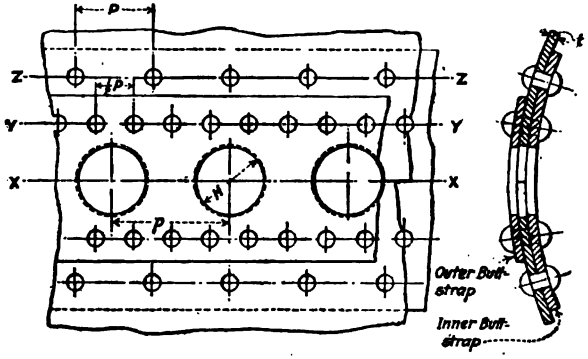


FIG. 26.—Example of Riveted Seam, The B. & W. Co.

In the following analysis of this seam the tensile strength of plate and shearing strength of rivets is taken in conformity with the A.S.M.E. Boiler Code recently approved by the U. S. Supervising Inspectors of Steam Vessels during the World War. The diameter of drum, thickness of plate, pitch, etc., are typical values. It will be noted that this joint not only fills the requirements of the longitudinal seam of the drum but also provides for tube holes on 7-inch centres. We have:

$T.S.$ = Tensile strength of plates (min.) lbs. per sq. in. = 55,000 (A.S.M.E.)
s = Shearing strength of rivet in single shear, lbs. per sq. in. = 44,000 (A.S.M.E.).
S = Shearing strength of rivet in double shear, lbs. per sq. in. = 88,000 (A.S.M.E.).
t = Thickness of plate = $\frac{1}{2}$ inch.
b = Thickness of buttstrap = $\frac{9}{16}$ inch.
P = Pitch of rivet, on row having greatest pitch = $4\frac{9}{16}$ inches.
d = Diameter of rivet after driving = diameter of rivet hole = $\frac{29}{32}$ inch.
a = Cross-sectional area of rivet after driving = 0.645 sq. inch.
n = Number of rivets in single shear in a unit length of joint.
N = Number of rivets in double shear in a unit length of joint.
H = Diameter of tube hole = $4\frac{1}{32}$ inches.
p = Pitch of tube holes = 7 inches.
D = Diameter (inside) of drum = 42 inches.

Possible Modes of Failure of Joint:

I. Plate fractures at ZZ.
II. Rivets shear at ZZ (single) and at YY (double).
III. Plate fractures at YY and rivets shear at ZZ (single).
IV. Inner buttstrap fractures at XX and rivets shear at YY (single).
V. Inner and outer buttstraps fracture at XX.

There are other possibilities of failure but these give high efficiency of joint and are very remote.

Solid plate (on row having greatest pitch) $= P \times t \times TS = 4.5625 \times 0.5 \times 55{,}000 = 125{,}468$.

	Per cent of solid plate
I. $(P - d) \times t \times T.S.$ = $(4\ 9/16 - 2\ 9/32) \times 0.5 \times 55{,}000 = 100{,}547$ =	80.1 per cent.
II. $N \times S \times a + n \times s \times a$ = $2 \times 88{,}000 \times 0.645 + 1 \times 44{,}000 \times 0.645 = 141{,}900$ =	113.0 per cent.
III. $(P - 2d) \times b \times T.S. + n \times s \times a$ = $(4\ 9/16 - 2 \times 2\ 9/32) \times 0.5625 \times 55{,}000 + 1 \times 44{,}000 \times 0.645 = 113{,}420$ =	90.4 per cent.
Solid plate (at tube holes) 7 in. pitch = $p \times t \times 55{,}000 = 7 \times 0.5 \times 55{,}000 = 192{,}500$.	
IV. $(p - H) \times b \times T.S. + n \times s \times a$ = $(7 - 4.03125) \times 0.5625 \times 55{,}000 + 3 \times 44{,}000 \times 0.645 = 176.840$ =	81.6. per cent.
V. $(p - H) \times b \times TS \times 2 = (7 - 4.03125) \times 0.5625 \times 55{,}000 \times 2 = 183{,}691$ =	95.4 per cent.

NOTE.—With reference to IV, the pitch of rivets not being an exact multiple of the pitch of tubes the calculation is not precisely correct. However, the error is slight and may be neglected.

The **least percentage** is that of $I = 80.1$, which is considered as the **efficiency of this joint.**

The analyses given in tables 17 to 22 inclusive, of riveted joints shown in Figs. 27 to 32 inclusive, as used in Scotch boilers are submitted by Mr. Henry C. E. Meyer, of the American Bureau of Shipping. The illustrations marked "Type A" to "Type F" show typical constructions for Scotch boilers.

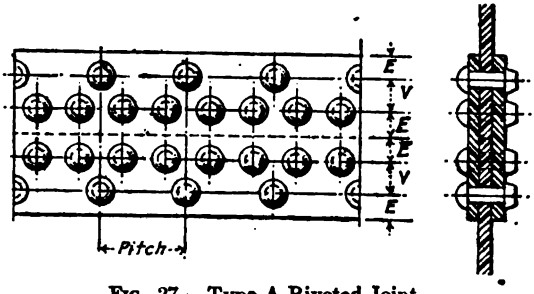


FIG. 27.—Type A Riveted Joint.

Table 17.
(Fig. 27, Type A Joint)

Shell plate, in.	Diam. of rivet hole, in.	Pitch, in.	Percentage		E, in.	V, in.
			Plate	Rivet		
½	¾	4¾	81.5	109.1	1⅝	1⅞
17/32	¾	5	82.5	97.6	1⅝	1⅞
9/16	¾	5	82.5	92.2	1⅝	1⅞
19/32	¾	5	82.5	87.3	1⅝	1⅞
⅝	¾	5	82.5	82.9	1⅝	1⅞
21/32	15/16	5¼	82.1	86.4	1⅞	2
11/16	15/16	5¼	82.1	82.4	1⅞	2
23/32	1	5¼	80.9	89.7	1½	2⅛
¾	1	5¼	80.9	86.0	1½	2⅛
25/32	1 1/16	5½	80.6	88.9	1⅝	2¼
13/16	1 1/16	5½	80.6	85.5	1⅝	2¼
27/32	1 1/16	5½	80.6	82.3	1⅝	2¼
⅞	1⅛	5¾	80.4	85.2	1 11/16	2⅜
29/32	1⅛	5¾	80.4	82.2	1 11/16	2⅜

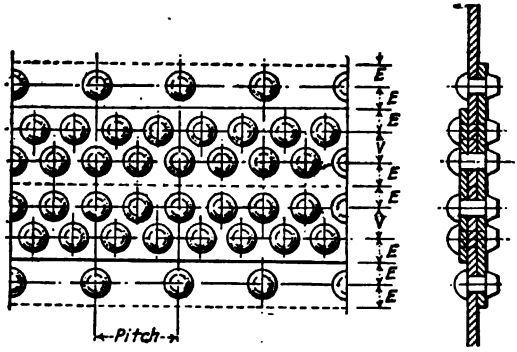


FIG. 28.—Type B Riveted Joint.

Table 18.
(Fig. 28, Type B Joint)

Shell plate, in.	Diam. of rivet hole, in.	Pitch, in.	Percentage		E, in.	V, in.
			Plate	Rivet		
15/16	1	6	83.3	92.8	1½	2⅛
27/32	1 1/16	6¼	83.0	96.6	1⅝	2¼
⅞	1 1/16	6¼	83.0	93.2	1⅝	2¼
29/32	1 1/16	6¼	83.0	90.0	1⅝	2¼
15/16	1⅛	6½	82.6	93.7	1 11/16	2⅜
31/32	1⅛	6½	82.6	90.7	1 11/16	2⅜
1	1⅛	6½	82.6	87.9	1 11/16	2⅜
1 1/32	1 3/16	6¾	82.4	91.4	1 13/16	2½
1 1/16	1 3/16	6¾	82.4	88.7	1 13/16	2½
1 3/32	1¼	7	82.1	92.1	1¾	2⅝
1⅛	1¼	7	82.1	89.6	1¾	2⅝
1 5/32	1¼	7	82.1	87.1	1⅞	2⅝
1 3/16	1 5/16	7¼	81.8	90.3	2	2¾
1 7/32	1 5/16	7¼	81.8	88.0	2	2¾
1¼	1 5/16	7¼	81.8	85.8	2	2¾
1 9/32	1⅜	7½	81.6	88.8	2 1/16	2⅞
1 5/16	1⅜	7½	81.6	86.7	2 1/16	2⅞
1 11/32	1⅜	7½	81.6	84.7	2 1/16	2⅞
1⅜	1 7/16	7¾	81.4	87.5	2 3/16	3
1 13/32	1 7/16	7¾	81.4	85.4	2 3/16	3
1 7/16	1 7/16	7¾	81.4	83.7	2 3/16	3
1 15/32	1½	8	81.2	86.4	2¼	3⅛
1½	1½	8	81.2	84.6	2¼	3⅛

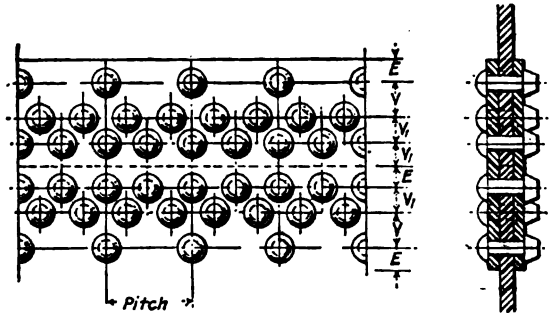


FIG. 29.—Type C Riveted Joint.

Table 19.
(Fig. 29, Type C Joint)

Plate, in.	Diam. of rivet hole, in.	Pitch, in.	Percentage		*E*, in.	*V*, in.	V_1, in.	*W* = width of butt-strap, in.
			Plate	Rivet				
1¼	1¼	7¾	83.8	91.0	1⅞	3 1/16	2 5/16	17¾
1 9/32	1 5/16	8	83.5	94.8	2	3 3/16	2 3/16	18¾
1 5/16	1 5/16	8	83.5	92.6	2	3 3/16	2 3/16	18¾
1 11/32	1⅜	8¼	83.3	96.2	2 1/16	3¼	2¼	19¼
1⅜	1⅜	8½	83.7	91.3	2 1/16	3 5/16	2 5/16	19½
1 13/32	1 7/16	8¾	83.5	94.8	2 1/16	3½	2⅜	20½
1 7/16	1 7/16	9	84.0	90.1	2 1/16	3½	2⅜	20½
1 15/32	1½	9¼	83.7	93.4	2¼	3⅝	2½	21¼
1½	1½	9½	84.2	89.1	2¼	3 11/16	2 9/16	21½
1 17/32	1 9/16	9½	83.5	94.7	2⅜	3¾	2⅝	22¼
1 9/16	1 9/16	9¾	83.9	90.4	2⅜	3 11/16	2 11/16	22½
1 19/32	1⅝	10	83.7	93.5	2 7/16	3 11/16	2 11/16	23
1⅝	1⅝	10¼	84.1	89.4	2 7/16	4	2¾	23¼
1 21/32	1 11/16	10½	83.9	92.4	2 9/16	4 1/16	2 11/16	24
1 11/16	1 11/16	10½	83.9	90.7	2 9/16	4 1/16	2 11/16	24
1 23/32	1¾	10¾	83.7	93.5	2⅝	4 1/16	2 11/16	24¾
1¾	1¾	11	84.0	89.8	2⅝	4¼	3	25
1 25/32	1 13/16	11¼	83.8	92.5	2¾	4⅜	3	25¾
1 13/16	1 13/16	11¼	83.8	90.9	2¾	4⅜	3	25¾
1 27/32	1⅞	11½	83.6	93.6	2 13/16	4½	3⅛	26½
1⅞	1⅞	11¾	84.0	90.0	2 13/16	4 9/16	3 3/16	26¾
1 29/32	1 15/16	12	83.8	92.6	2 15/16	4 11/16	3 3/16	27½
1 15/16	1 15/16	12	83.8	91.1	2 15/16	4 11/16	3 3/16	27½
1 31/32	2	12¼	83.6	93.6	3	4 13/16	3 5/16	28¼
2	2	12½	84.0	90.3	3	4⅞	3⅜	28½

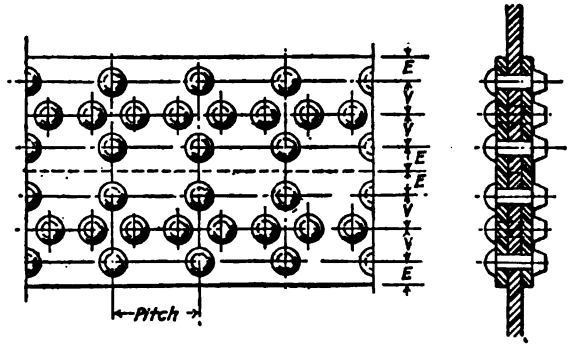


Fig. 30.—Type D Riveted Joint.

Table 20.

(Fig. 30, Type D Joint)

Shell plate, in.	Diam. of rivet hole, in.	Pitch, in.	Percentage		Combined %			*E*, in.	*V*, in.
			Plate	Rivet	Comb.	Plate	Rivet		
19/32	15/16	6¼	85.0	95.4	85.2	70.0	15.2	1 7/16	2
5/8	1	6½	84.6	99.2	85.0	69.2	15.8	1½	2⅛
21/32	1	6½	84.6	94.5	84.3	69.2	15.1	1½	2⅛
11/16	1	6¾	85.1	86.8	84.1	70.2	13.9	1½	2⅛
23/32	1	6¾	85.1	83.1	83.5	70.2	13.3	1½	2⅛
¾	1 1/16	7	84.8	86.6	83.4	69.6	13.8	1⅝	2¼
25/32	1 1/16	7	84.8	83.4	82.9	69.6	13.3	1⅝	2¼
13/16	1⅛	7¼	84.4	86.6	82.7	68.9	13.8	1 11/16	2⅜
27/32	1⅛	7¼	84.4	83.4	82.2	68.9	13.3	1 11/16	2⅜

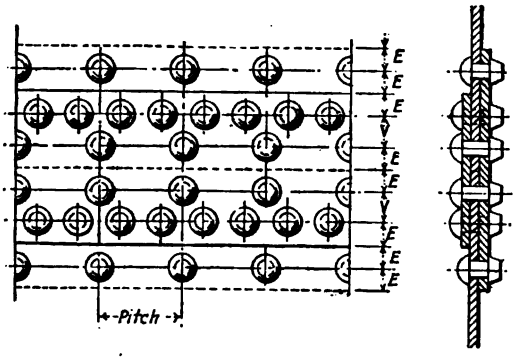


FIG. 31.—Type E Riveted Joint.

Table 21.

(Fig. 31, Type E Joint)

Shell plate, in.	Diam. of rivet hole, in.	Pitch, in.	Percentage		Combined %			*E*, in.	*V*, in.
			Plate	Rivet	Comb.	Plate	Rivet		
27/32	1 1/16	8 1/2	86.7	91.0	84.8	73.5	11.3	1 11/16	2 5/8
7/8	1 1/16	8 1/2	86.7	87.8	84.4	73.5	10.9	1 11/16	2 5/8
29/32	1 1/16	8 1/2	86.7	84.7	84.0	73.5	10.5	1 11/16	2 5/8
15/16	1 3/16	8 5/8	86.2	90.0	83.6	72.4	11.2	1 13/16	2 1/2
31/32	1 3/16	8 5/8	86.2	87.1	83.2	72.4	10.8	1 13/16	2 1/2
1	1 3/16	8 5/8	86.2	84.3	82.9	72.4	10.5	1 13/16	2 1/2
1 1/32	1 3/16	8 5/8	86.2	81.8	82.6	72.4	10.2	1 13/16	2 1/2
1 1/16	1 1/4	8 3/4	85.7	86.7	82.2	71.4	10.8	1 7/8	2 3/4
1 3/32	1 1/4	8 3/4	85.7	84.2	81.9	71.4	10.5	1 7/8	2 3/4
1 1/8	1 1/4	8 3/4	85.7	81.9	81.6	71.4	10.2	1 7/8	2 3/4
1 5/32	1 5/16	8 7/8	85.2	86.6	81.2	70.4	10.8	2	2 3/4
1 3/16	1 5/16	8 7/8	85.2	84.3	80.9	70.4	10.5	2	2 3/4
1 7/32	1 5/16	8 7/8	85.2	82.1	80.6	70.4	10.2	2	2 3/4
1 1/4	1 5/16	8 7/8	85.2	80.1	80.4	70.4	10.0	2	2 3/4

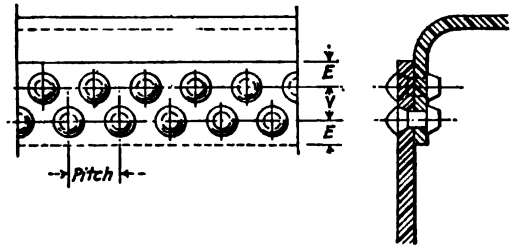


FIG. 32.—Type F Riveted Joint.

Table 22.
(Fig. 32, Type F Joint)

Diam. of rivet hole, in.	Pitch, in.	*E*, in.	*V*, in.	Width of lap, in.
7/8	3 1/4	1 5/16	1 7/8	4 1/2
15/16	3 1/2	1 7/16	2	4 7/8
1 1/16	3 1/2	1 5/8	2 1/4	5 1/2
1 1/8	3 3/4	1 11/16	2 3/8	5 3/4
1 3/16	3 3/4	1 13/16	2 1/2	6 1/8
1 1/4	4	1 7/8	2 5/8	6 3/8
1 5/16	4 1/4	2	2 3/4	6 3/4
1 3/8	4 1/2	2 1/16	2 7/8	7
1 7/16	4 3/4	2 3/16	3	7 3/8
1 1/2	4 3/4	2 1/4	3 1/8	7 5/8

Rivet Holes. The **U. S. Supervising Inspectors' Rules** require that all rivet holes shall be drilled in place, with the plates assembled, and without any previous punching. A few "**tack**" **holes** are drilled and the plates fastened together with turned bolts to insure accuracy of fit. After drilling, it is essential that the plates be taken apart and all burrs carefully removed. The plates are then re-assembled, metal to metal, and held in position by turned bolts, preparatory to riveting.

It has been well established that with properly constructed dies, open-hearth steel plates may be punched without distortion of the metal around the periphery of the hole to a depth from same greater than 1/8 in. If, therefore, holes are punched 1/4 in. less in diameter than the finished hole, and then reamed or drilled to size, the effect will be equivalent to a hole drilled without punching. If the dies, however, are not properly designed, or if they become greatly worn, there is a possibility of injuring the metal to a greater depth than 1/8 in., and this is accentuated on the less ductile plates.

Rivets. The **diameter of rivet** used in the analyses of riveted seams and in all calculations of riveted joints, is that of the driven rivet, and is the same as the diameter of the rivet hole, it being assumed that after driving, the rivet will occupy the full volume of the hole. This result is practically assured by proper hydraulic riveting. The diameter of the rivet selected before riveting is usually about 1/16 in. less than the diameter of the drilled hole. With accurate alignment of rivet holes, and particularly with plates not over 3/4 in. or 1 in. in thickness, the clearance allowance between diameter of rivet and rivet hole may be reduced to 1/32 in.

The length of rivet stock under the head must be correctly proportioned to fill the full volume of the rivet hole and to give the proper thickness and form of head after driving. The diagram, Fig. 33, submitted by Mr. Philip

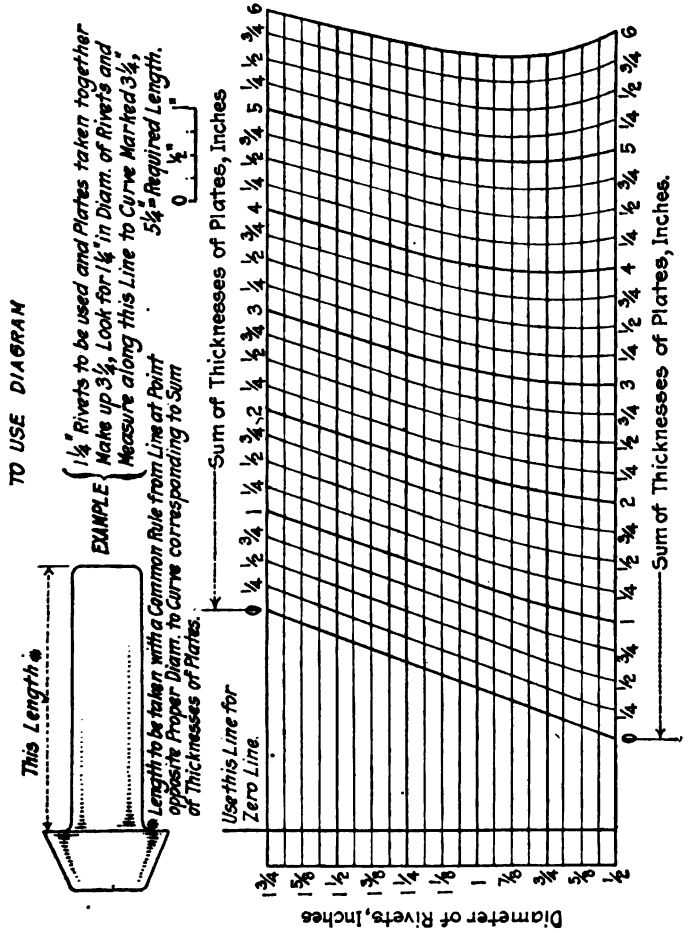

TO USE DIAGRAM
This Length
EXAMPLE
1¼" Rivets to be used and Plates taken together
Make up 3¼", Look for ⅛" in Diam. of Rivets and
Measure along this Line to Curve Marked 3¼",
5¼"=Required Length.
Length to be taken with a Common Rule from Line at Point
opposite Proper Diam. to Curve corresponding to Sum
of Thicknesses of Plates.
0 ½"
Use this Line for
Zero Line.
Sum of Thicknesses of Plates, Inches
Diameter of Rivets, Inches
Sum of Thicknesses of Plates, Inches.

Fig. 33.—Rivet Length Diagram (Diagram to ½ Scale, Multiply Actual Measurement by 2.)

M. Young, Chief Engineer of the New York Shipbuilding Corporation, provides a ready means for calculating the proper length of rivets. The form and proportions of **rivet heads** may vary considerably, the essential point being to secure proper holding power with the minimum amount of metal. **Representative practice** is indicated by Fig. 34a also furnished by Mr. Young.

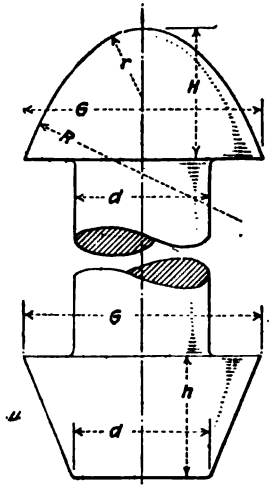


FIG. 34a, See Table 23.

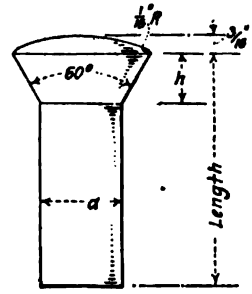


FIG. 34b, See Table 24.

Table 23. Standard Boiler Rivet Dimensions

New York Shipbuilding Corporation

(Fig. 34a)

d	G	h	H	r	R
15/8	2 13/16	1 27/64	1 33/64	13/16	3 1/16
1 1/2	2 9/16	1 5/16	1 13/32	13/16	2 15/32
1 3/8	2 3/8	1 9/64	1 9/32	3/4	2 1/4
1 5/16	2 5/16	1 9/64	1 7/32	21/32	2 5/32
1 1/4	2 3/16	1 3/32	1 5/32	5/8	2 1/16
1 3/16	2 1/16	1 1/32	1 3/32	19/32	1 31/32
1 1/8	1 15/16	63/64	1 1/16	9/16	1 7/8
1 1/16	1 13/16	59/64	1	17/32	1 25/32
1	1 3/4	7/8	15/16	1/2	1 11/16
7/8	1 1/2	49/64	13/16	7/16	1 1/2
3/4	1 5/16	21/32	11/16	3/8	1 5/16
5/8	1 1/16	35/64	9/16	5/16	1 1/8
1/2	7/8	7/16	15/32	1/4	15/16
3/8	5/8	21/64	11/32	3/16	3/4
5/16	17/32	17/64	9/32	5/32	21/32
1/4	7/16	7/32	7/32	1/8	9/16

Table 24. Dimensions of Countersunk Rivets

(Fig. 34b)

d, inches	3/4	13/16	7/8	15/16	1
h, inches	1/2	17/32	9/16	19/32	5/8

In seams not exposed to contact with the hot gases, **countersunk** rivets are no longer considered good practice in boiler work, as they weaken the plate without adequate compensation. **In the combustion chambers of Scotch boilers,** however, the button-head rivet shown above is not suitable, on account of the liability to overheat the rivet head, and the countersunk rivet is resorted to in this case. A typical design is shown in Fig. 34b. Representative values of the depth of the countersink may be taken from Table 24.

CALKING

It is customary to calk the edge of the buttstrap on the **outside** of the shell and the edge of the shell on the outside of the drumhead, etc., to make the seam tight against leakage. It is also very common practice with Scotch boilers to calk the seams on the inside as well. Some manufacturers calk the seams thoroughly both outside and inside and rely on light outside calking to stop any leaks which may subsequently develop. The value of **inside calking** in combination with outside calking is disputed, although it is to be noted that with inside calking there is less liability of the rivets leaking.

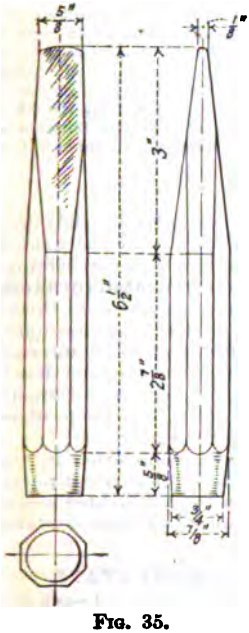


FIG. 35.

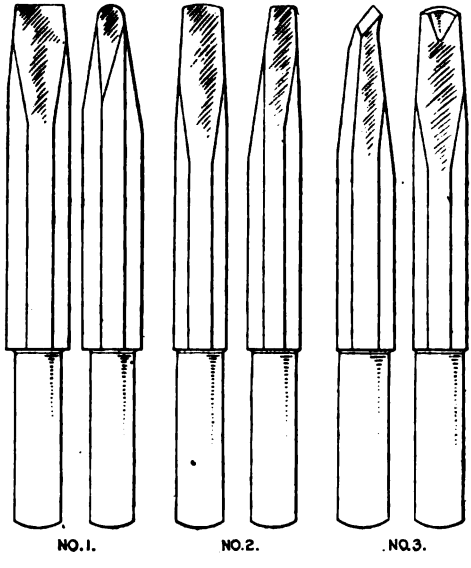


FIG. 36.

The Navy Department has recently specified in certain cases that the calking shall be done on the inside of the shell only, the object being to prevent boiler water which may have soda in solution from getting in between the shell plates and the buttstrap. The use of soda in the boiler water in considerable amount, while effective in preventing corrosion, may result in a concentrated soda solution collecting between the plates and the buttstraps; and it is now a well-recognised fact that this condition may result in a chemical reaction resulting in the liberation of hydrogen which on being absorbed by the metal causes the same to become brittle. It was to prevent the possibility of this action that the Navy Department adopted the above-mentioned ruling. The boilers

covered by this rule were built with buttstraps of unequal width, and in order to give a properly stiffened calking edge, the narrow buttstrap was placed on the inside of the shell.

It is obvious that calking on the inside of the shell only is a much more difficult and expensive job than the usual method of depending on the outside calking for a tight joint. In the first place, it is difficult to find the point where the leakage exists; and in the second place, in case of leakage it requires emptying the boiler, re-calking the seam, filling the boiler and again putting on the test pressure. The same difficulty occurs in case a leak subsequently develops in the seam and additional calking is required. This method, however, is effective in preventing all water from getting in between the plate and the buttstrap.

The Navy Department has been the first to use soda in considerable amounts in boiler water, and has found it effective in preventing corrosion; and as boiler pressures increase, with corresponding increases in temperature, with the result that the magnesium chloride in the sea water becomes more corrosive in its action, it is not impossible that the use of soda will become greatly extended in merchant work. If it does, it is likely that inside calking, with no outside calking, may be resorted to. This will inevitably result in a greater effort on the part of the boiler manufacturers to make a riveted joint which will be tight without calking.

Already considerable progress in this respect has been made by The Babcock & Wilcox Company, who use a **calking tool** shown in Fig. 35. Obviously, the less calking required, the smaller the tool which will be used. It is now not infrequent, in Scotch boilers using thick plates, to use three calking tools, one for the rivets and two for the plates; one heavy tool being used for setting down the edge of the buttstrap, and a finer tool for giving the final touch. (See Fig. 36.) Even this has not always been effective, and cases have occurred where a tight joint could only be secured by welding the edge of the buttstrap to the shell plate. Greatly improved methods of fitting the buttstraps and bending the shell plates will be necessary if inside calking instead of outside calking becomes the vogue.

WELDED DRUMS

Welded seams are used successfully in corrugated furnaces for Scotch boilers and **for small drums for water-tube boilers.** Theoretically the welded joint is fully as strong as the original plate and it of course requires no calking. This joint has been used to the exclusion of all others **in corrugated furnaces** where a riveted joint cannot be employed to advantage and where the thickness of the plate is limited and where the material is in compression only. That it has not been used extensively **for boiler drums** subjected to internal pressure, notwithstanding its apparent advantages, is due to the fact that a weld is always a questionable quantity. Imperfections which cause tremendous weakness in the joint cannot always be detected and it lacks the positive assurance of permanency which is contained in the riveted joint.

It sometimes happens also that welded drums become distorted in the welding process and sufficiently out of true to affect the fit of adjacent parts, if not to weaken the drum. However, cylindrical shells with welded seams may be constructed from 20 in. diameter up to diameters requiring plates 1 in. in thickness and even to 1¼ in. in extreme cases.

DISHED OR BUMPED HEADS—HEADS WITHOUT STAYS

The U. S. Supervising Inspectors allow flat heads to be used without stays, provided the diameter of the drum head does not exceed 20 in. and the flanges are made with an inside radius of at least 1½ in. In this case the allowable steam pressure is determined by the formula:

Flat heads less than 20 in. in diam.: $P = \frac{C \times T^2}{A}$; where

P = Steam press. allowable in lbs.
T = Thickness of material in sixteenths of an in. (numerator only).

A = One-half the area of head in in.
C = 112 for plates $\frac{7}{16}$ in. and under.
C = 120 for plates over $\frac{7}{16}$ in.

Flat heads greater than 20 in. in diameter are considered as "flat surfaces," and require staying. (See Flat Surfaces, p. 410.)

Dished or bumped heads are not required to be stayed. To avoid grooving, it is required that the flanging shall be well rounded at the bend. Bumped heads may contain a manhole opening flanged inwardly when such flange is turned to a depth of three times the thickness of the material in the head. When these heads are "convex" (that is, having the pressure on the concave side) they are allowed by the Supervising Inspectors' Rules a working pressure in accordance with the formula:

$$\textbf{Convex Heads: } P = \frac{T \times S}{R}\text{; where}$$

P = Steam press. allowable in lbs.
T = Thickness of plate in in.
S = One-fifth of the tensile strength in lbs. per sq. in.
R = One-half of the radius to which the head is bumped in in.

"Concave" heads (that is, those having the pressure on the convex side) are allowed 80 per cent of the pressure allowable for convex heads as per the above formula.

Bumped heads are required to be formed to a true spherical surface. While not specifically so stated in the **Supervising Inspectors' Rules**, it is understood—and it is the general practice—that the heads will be bumped to a radius not greater than the diameter of the drum. This is in accordance with the rules of the **British Board of Trade**, which specifically require the following:

"Ends of steam receivers which are dished and flanged in a hydraulic press, the whole end being operated upon at each heat, may be passed without stays, provided that the radius of the end is not greater than the diameter of the shell to which it is attached and does not exceed 4 ft., that the outer radius of the flange at the root is not less than 3 in., that the end is made of mild steel of the usual quality, and that it is efficiently annealed after the completion of the setting and flanging; but the maximum pressure allowed without stays should not exceed that found by the following formula:

$$\frac{90{,}000 \times T \times h}{D^2} = \textbf{Working pressure; where}$$

T = Thickness of end, in in.
D = Diam. of shell, inside, in in.
h = Distance, in in., from the centre of the end to the inner side to a straight line passing through the points where the curve of the end due to the inner radius, and produced, would intersect the sides of the cylindrical shell, produced if necessary, at the diameter D, and

$$= R - \sqrt{R^2 - \frac{D^2}{4}}\text{, in which } R = \begin{cases}\text{inner radius}\\ \text{of end.}\\ \text{in in.}\end{cases}$$

"If there is a manhole in the end, T should exceed that found by the formula by $\frac{1}{8}$ in."

The printed rules of **Lloyd's Register of Shipping** do not include any formula for bumped heads, presumably because the rules were originally made to cover shell boilers with flat stayed heads. Being called upon to examine water-tube boilers with bumped heads, it became necessary for Lloyd's surveyors to devise a rule for safe working pressures. The resulting formula is given below as submitted through the courtesy of the

Secretary of Lloyd's Register of Shipping, New York, and is the formula used by Lloyd's surveyors in this country:

$$\text{Working Press., Dished Heads} = \frac{C \times (T - 2) \times B}{R}; \text{ where}$$

C = 19.05.
T = Thickness of plate in sixteenths of an in. (numerator only).
B = Percentage of strength of joint.
R = Radius of dishing in in.

The **American Bureau of Shipping** also fails to include a formula for bumped heads in their printed rules. The rule used by their surveyors is given by Mr. Meyer, as follows, this being based on the rule of the A. S. M. E., boiler code:

$$\text{Working press.} = \frac{S\,(T - 4)}{R \times 88}; \text{ where}$$

S = Tensile strength of plate in lbs. per sq. in.
T = Thickness of plate in thirty-seconds of an inch (numerator only).
R = Radius of bump in in. Where two radii are used the larger shall be taken as the value of R.

MANHOLES, HANDHOLES, ETC.

The U. S. Supervising Inspectors specify that all boilers shall have a manhole opening above the flues or tubes, not less than 10 in. × 16 in. or 11 in. × 15 in., or equal, and such other openings as may be required when considering the design submitted for approval. Openings exceeding 6 in. in diameter shall be reinforced either on the inside or outside with reinforcing wrought-iron or steel rings, which shall be securely riveted or properly fastened to the boiler, such rings to be of sufficient width and thickness of material to fully compensate for the amount of material cut from the boilers. Where the opening is made in the circumferential plates, the reinforcing ring shall have a sectional area equal to at least ½ of the sectional area of the opening parallel to the longitudinal seams of such boiler. On boilers carrying 75 lbs. or less, a stop valve properly flanged may be used as a reinforcement to the opening. When holes are cut in flat surfaces and are flanged inwardly to a depth of not less than 1½ in., measuring from the outside surface, reinforcing rings may be dispensed with.

Bumped heads may contain a manhole opening, flanged inwardly, when such flange is turned to a depth of three times the thickness of material in the head.

The American Bureau of Shipping recommends that manholes be not less than 16 in. × 12 in., and requires that the openings "are to be efficiently compensated."

CYLINDRICAL SHELLS, RIVETED

The U. S. Supervising Inspectors require that:

"The working steam pressure allowable on cylindrical shells of boilers, constructed of plates inspected as required by these rules, when single riveted, shall not produce a strain to exceed one-sixth of the tensile strength of the iron or steel plates of which such boilers are constructed; but where the longitudinal laps of the cylindrical parts of such boilers are double riveted, and the rivet holes for such boilers have been fairly drilled instead of punched, an addition of 20 per cent to the working pressure provided for single riveting shall be allowed.

"The pressure for any dimension of boilers shall be ascertained by the following rule, viz.:

"Multiply one-sixth of the lowest tensile strength found stamped on the plates in the cylindrical shell by the thickness—expressed in inches or part of an inch—and divide by the radius or half diameter, also expressed in inches, and the result will be the pressure allowable per square inch of surface for single riveting, to which add 20 per cent where the longitudinal laps of the cylindrical parts of such boiler are double riveted, when all the rivet holes of such boiler, including steam and mud drums, have been fairly drilled and no part of such holes has been punched. The pressure allowed shall be based on the plate whose tensile strength multiplied by its thickness gives the lowest product."

American Bureau of Shipping requires that:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times (T - 1) \times E \times S}{D \times 28}$$

where D = greatest internal diam. of shell in in.

C = 20.4 when longitudinal seams are fitted with double butt straps of equal width, and having thickness at least equal to that obtained from the formula for butt straps.

C = 19.7 when the double butt straps are of unequal width.

C = 19.0 when the longitudinal seams are lap joints.

S = minimum tensile strength of material, in long tons per sq. in.

E = Least percentage of strength of longitudinal joints, found as follows:

For the plate at the joint, $E = \frac{p - d}{p} \times 100$

For the rivets at the joint, $E = \frac{n \times a}{p \times t} \times \frac{r}{s} \times 100$

Where p = pitch of rivets, in outer rows, in in.

d = diam. of rivet holes, in in.

n = number of rivets used per pitch.

a = sectional area of one rivet, in sq. in., when in single shear.

a = sectional area of one rivet in sq. in., multiplied by 1.875 when in double shear.

r = 23 for steel and 19 for iron rivets.

s = minimum tensile strength of shell plates as specified.

t = thickness of plate, in in.

Lloyd's Register of Shipping requires that the strength of cylindrical steel boiler shells be calculated from:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times (T - 2) \times B}{D}$$

where D = mean diam. of shell in in.

T = thickness of plate in sixteenths of an in.

C = 22 when the longitudinal seams are fitted with double straps of equal width.

C = 21.25 when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets.

C = 20.5 when the longitudinal seams are lap joints. If the minimum tensile strength of shell plates is other than 28 tons per sq. in., these values of C should be correspondingly modified.

B = the least percentage of strength of longitudinal joint (found as follows):

For plate at joint $B = \frac{p - d}{p} \times 100$

For rivets at joint $B = \frac{n \times a}{p \times t} \times 85$ for steel rivets

$B = \frac{n \times a}{p \times t} \times 70$ for iron rivets

where p, t, d, n and a have the same meaning as given by preceding Bureau of Shipping rules.

In case of rivets in double shear $1.75a$ is to be used instead of a in the above formula. The inside butt strap to be at least ¾ of the strength of the longitudinal joint. For the **shell plates of superheaters** or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the coefficients should be ⅔ of those given in the preceding tables. Proper deductions are to be made for openings in shell. All manholes in circular shells to be stiffened with compensating rings. The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

Bureau Veritas requires that:

$$\text{Working pressure, lbs. per sq. in.} = \frac{2\alpha R(t - 0.04)}{D}$$

where D = greatest inside diam. of shell in in.

t = thickness of shell plate in in.; $t - 0.04$ represents thickness left after reduction of 0.04 in. by corrosion.

R = Tensile stress in lb. per sq. in. allowed in plate. The value of R will be the breaking strength divided by 4, the latter figure representing factor of safety for plate after it has corroded away by 0.04 in. If the actual breaking strength be known by tests carried out to the administration's satisfaction it may be applied to find R, otherwise the value of R will be.

For steel, the fourth part of the lower limit of tensile strength chosen by the designer, which in such case is to be stated when a boiler design is submitted for approval.

For iron, 11,200 lbs. per sq. in., corresponding with a tensile strength of about 20 tons.

α is the ratio of the resistance of the plate left between the holes, to that of the full plates. It will be determined from the expression:

$\alpha = \frac{p - d}{p}$ where p and d have same meaning as in Bureau Shipping rules.

FLAT SURFACES, STAYED

U. S. Supervising Inspectors require that:

"The area of the segment of a head to be stayed shall be that surface contained within a line drawn 3 in. from the inner circle of the head and 2 in. from the tubes or flues. All stayed surfaces formed to a curve the radius of which is over 21 in., excepting surfaces otherwise provided for, shall be deemed flat surfaces.

The maximum stress allowable on flat plates supported by stays shall be determined by the following formula:

$$\text{Working pressure} = \frac{C \times T^2}{P^2}$$

Where T = thickness of plates in sixteenths of an in.

P = greatest pitch of stays in in.

C = 112 for screw stays with riveted heads, plates $\frac{7}{16}$ in. and under.

C = 120 for screw stays with riveted heads, plates above $\frac{7}{16}$ in.

C = 120 for screw stays with nuts, plates $\frac{7}{16}$ in. and under.

C = 125 for screw stays with nuts, plates above $\frac{7}{16}$ in. and under $\frac{9}{16}$ in.

C = 135 for screw stays with nuts, plates $\frac{9}{16}$ in. and above.

C = 175 for stays with double nuts, having one nut on the inside and one nut on the outside of plate, without washers or doubling plates.

C = 160 for stays fitted with washers or doubling strips which have a thickness of at least 0.5 of the thickness of the plate and a diam. of at least 0.5 of the greatest pitch of the stay, riveted to the outside of the plates, and stays having one nut inside of the plate and one nut outside of the washer or doubling strip. For T take 72 per cent of the combined thickness of the plate and washer or plate and doubling strip.

C = 200 for stays fitted with doubling plates which have a thickness equal to at least 0.5 of the thickness of the plate reinforced, and covering the full area braced (up to the curvature of the flange, if any), riveted to either the inside or outside of the plate, and stays having one nut outside and one inside of the plates. Washers or doubling plates to be substantially riveted. For T take 72 per cent of the combined thickness of the two plates.

C = 200 for stays with plates stiffened with tees or angle bars having a thickness of at least $\frac{3}{8}$ the thickness of plate and depth of webs at least $\frac{1}{4}$ of the greatest pitch of the stays, and substantially riveted on the inside of the plates, and stays having one nut inside bearing on washers fitted to the edges of the webs that are at right angles to the plate. For T take 72 per cent of the combined thickness of web and plate.

No such flat plates or surfaces shall be unsupported at a greater distance than 18 in.

American Bureau of Shipping reqûire that for flat surfaces:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times T^2}{P^2 + p^2}$$

Flat Surfaces, **fitted with doublings,** the doublings being at least $\frac{3}{8}$ of the thickness of the plates to which they are riveted.

$$\text{Working pressure, in lbs. per sq. in.} = \frac{C \times \left(T + \frac{t}{2}\right)^2}{P^2 + p^2}$$

Where T = thickness of plate, in sixteenths of an inch.

t = thickness of doubling plates, in sixteenths of an inch.

P = distance between the rows of stays, in in.

p = pitch of stays in the rows, in in.

C = 200 for plates fitted with screwed stays having riveted heads.

C = 270 for plates fitted with screwed stays and nuts.

C = 350 for plates fitted with stays and double nuts.

C = 370 for plates fitted with stays, double nuts, and washers outside. The washers to be at least $\frac{3}{8}$ of the thickness of the plate and in diam. equal to $\frac{1}{5}$ the pitch of the stays, but in no case are they to be less than 3 times the diam. of the stays.

C = 400 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be at least $\frac{3}{8}$ of the thickness of the plate, and in diam. equal to $\frac{2}{5}$ the pitch of the stays.

C = 450 for plates fitted with stays, double nuts, and washers outside riveted to the plate. The washers to be the same thickness as the plate, and in diam. equal to $\frac{3}{5}$ the pitch of the stays.

C = 480 for plates fitted with stays, double nuts, and doubling strips outside riveted to the plates. The strips to be the same thickness as the plate, and in width equal to ⅔ the pitch of the stays.

For front plates in the steam space, which are not protected against the direct action of the flame, the values of C given above are to be reduced 20 per cent. Stays which are screwed into plates of less thickness than 7/16 in. are not to have riveted heads, but are to be fitted with nuts.

Lloyd's Register requires that the **strength of flat plates** supported by stays is to be taken from the formula:

$$\text{Working pressure in lbs. per sq. in.} = \frac{C \times T^2}{P^2}$$

Where T = thickness of plate in sixteenths of an inch.
P = square of pitch in in. If the pitch in rows is not equal to that between the rows, then the means of the squares of the two pitches is to be taken.
C = 90 for iron or steel plates 7/16 thick and under, fitted with screw stays with riveted heads.
C = 110 for iron or steel plates 7/16 thick and under, fitted with stays and nuts.
C = 120 for iron plates above 7/16 thick, and for steel plates above 7/16 and under 9/16 thick fitted with screw and nuts.
C = 135 for steel plate 9/16 thick and above fitted with screw stays and nuts.
C = 140 for iron plates fitted with stays with double nuts.
C = 150 for iron plates fitted with stays with double of at least ½ of the pitch in diam. and ½ the thickness of the plates.
C = 160 for iron plates with stays with double nuts and washers riveted to the outside of the plates, of at least ⅔ of the pitch in diam. and ½ the thickness of the plates.
C = 175 for iron plates with stays with double nuts and washers when the washers are at least ⅔ of the pitch in diam. and of the same thickness as the plates.

For **iron plates** fitted with stays with double nuts and double strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to ⅔ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows and 190 when P is taken to be the pitch between the stays in the rows.

For **steel plates**, other than those for Combustion chambers, the values of C may be increased as follows:

C =	140	150	160	175	190
Increased to.....	175	185	200	220	240

If flat plates are **strengthened with doubling plates** securely riveted to them having a thickness of not less than ⅔ of that of the plates, the strength to be taken from the following formula:

$$\frac{C \times \left(T + \frac{t}{2}\right)^2}{P} = \text{working pressure in lbs. per sq. in.}$$

Where t = thickness of doubling plates in sixteenths, and C, T, and P are as above. In the case of **front plates of boilers in the steam space**, these numbers should be reduced 20 per cent, unless the plates are guarded from the direct action of the heat.

Bureau Veritas requires that:

$$\text{Working pressure in lbs. per sq. in.} = \frac{(t-1)^2}{a^2 + b^2} \times \frac{T}{C}$$

Where t = thickness of plate in sixteenths of an in.
a = pitch of stays, in in., in one row.
b = distance, in in., between two rows of stays.

When plates are effectively stiffened by doubling plates well riveted thereto, and having a thickness t in sixteenths, the value $(t + \frac{1}{2}t)$ may be substituted for t in the formula:

In case of irregular staying, $\frac{1}{4}(a + b)^2$ shall be used instead of $(a^2 + b^2)$; where a and b denote distance between centres of diagonally opposed stays.

T = tensile strength of the material in tons per sq. in. of the original section: For **steel** it will be equal to the lower limit of the tensile stress, which is to be stated on the drawing.

For **iron** 21 tons per square inch.

C = a constant, the value of which depends, upon the mode of staying as follows:

C = 0.104 when plates are supported by stays screwed through them and riveted over.

C = 0.079 when plates are supported by stays screwed through them and fitted with outside nuts.

C = 0.062 when there are nuts on both sides of plate and the outer nut has under it a washer whose diameter is not less than 0.4 of the distance between rows of stays with thickness not less than $\frac{2}{3}$ that of the plate.

C = 0.055 when the outside washers as above have a diam. at least 0.6 of the distance between rows of stays with a thickness not less than $\frac{3}{4}$ that of plate, the washers being well riveted to the plate.

C = 0.050 when outside washers are replaced by continuous strips of plate having a width of at least 0.6 of the distance between the rows of stays, with a thickness not less than $\frac{3}{4}$ that of the plate; the strips being well riveted to the plate.

For the values of $T \div C$ see Table 25.

When the **plates are in contact with steam on one side and flame or hot gases on the other**, the thickness is to be increased. For instance when in return tube boilers, the top front plates are in no way protected from the hot gases, the working pressure or thickness will in such case, be determined by the formulæ.

$$\text{Working pressure in lbs. per sq. in.} = \frac{(t - 2)^2}{a^2 + b^2} \times 0.9 \frac{T}{C}$$

When the said front plates are protected by a flame plate, no increase of thickness will be required. When front plates are in two pieces, the lap should be double riveted if the thicker plate is ½ in. or above.

Table 25. Values of $\frac{T}{C}$ in the Formula for Flat Plates

Tensile strength of plate in tons per sq. in.	$C = 0.104$ $\frac{T}{C} =$	$C = 0.079$* $\frac{T}{C} =$	$C = 0.062$ $\frac{T}{C} =$	$C = 0.055$ $\frac{T}{C} =$	$C = 0.050$ $\frac{T}{C} =$
20	192.3	253.1	322.6	363.6	400
21	202.0	265.8	338.6	381.8	420
22	211.5	278.4	354.8	400.0	440
23	221.1	291.1	371.0	418.0	460
24	230.8	303.8	387.0	436.2	480
25	240.4	316.4	403.2	454.4	500
26	250.5	329.1	419.4	472.6	520
27	259.6	341.8	435.4	490.8	540
28	269.2	354.4	451.6	509.0	560
29	278.8	367.1	467.8	527.2	580
30	288.4	379.7	483.8	545.4	600

* When plates are not subjected to flame or hot gases, C may be reduced from 0.079 to 0.066, or the values of $\frac{T}{C}$ in this column may be multiplied by 1.2.

STAYS

U. S. Steamboat Inspection Service requires that all holes for stays shall be drilled and no part punched. The **maximum working pressure** in lbs. allowable per sq. in. of cross-sectional area for stays used in the construc-

tion of marine boilers where same are accurately fitted normal to supported surfaces and properly secured, shall be ascertained by the following formula:

$$\text{Working pressure, lbs. per sq. in.} = \frac{A \times C}{a}$$

where A = least cross-sectional area of stay in in.
a = area of surface supported by one stay in in .
C = 9000 for tested steel stays 1 in. and upward in diam. when such stays are not forged or welded. The ends may be upset to a sufficient diam. to allow for depth of thread. The diam. shall be taken at the bottom of the thread, provided it is the least diam. of stay. All such stays after being upset shall be thoroughly annealed.
C = 8000 for a tested Huston or similar type of brace, the cross-sectional area of which exceeds 5 sq. in.
C = 7000 for such tested braces when the cross-sectional area is not less than 1.227 and not more than 5 sq. in., provided such braces are prepared at one heat from a solid piece of plate without welds.
C = 7500 for wrought-iron stays 1 in. and upward in diam. when made of the best quality of refined iron. The ends may be upset to allow for depth of thread. The diam. shall be taken at bottom of thread, provided it is the least diam. of stay. Such stays may be welded.
C = 6000 for welded crowfoot stays when made of best quality of refined wrought iron, and for all stays not otherwise provided for when made of the best quality of refined iron or of steel without welds.

American Bureau of Shipping requires that:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C(D - \frac{1}{8})^2}{S} \text{ or } D = \sqrt{\frac{S \times W}{C}} + \frac{1}{8}$$

where D = effective diam. of stay, in in.
S = flat surface, in sq. in., supported by the stay.
C = 8900 for steel longitudinal stays made from tested bars.
C = 8000 for steel screwed stays, and for iron screwed and longitudinal stays made from tested iron bars.
C = 7000 for untested iron longitudinal stays.
C = 6500 for iron screwed stays.
C = 5000 for welded iron stays.

The stress in **stay tubes** is not to exceed 7500 lbs. per sq. in.

Lloyd's Register of Shipping requires that the strength of stays supporting flat surfaces be calculated from the smallest part of the stay or fastening, and that the strain per sq. in. upon them is not to exceed the following limits:

For **Iron Stays** not exceeding 1½ in. smallest diam. and for all stays which are welded, 6000 lbs.; for unwelded stays above 1½ in. smallest diam., 7500 lbs.

For **Steel Stays**, if **screw stays** not exceeding 1½ in. smallest diam., 8000 lbs.; for screw stays above 1½ in. smallest diam., and for plain stays not exceeding 1½ in. smallest diam., 9000 lbs.; for stays exceeding 1½ in. smallest diam., except screw stays, 10,400 lbs. No steel stays are to be welded.

The stress in **stay tubes** is not to exceed 7500 lbs. per sq. in.

Bureau Veritas requires that diameter of **stays supporting flat surfaces** is to be determined by the formula:

$$D = 0.125 + \sqrt{\frac{Q}{300T}}$$

where D = effective diam. in in.
Q = total load on stay in lbs.
T = tensile strength of material in tons per sq. in.

For steel this tensile strength will be the lower limit chosen by the boiler designer: For iron, use 22 tons if the actual strength is not known from tests. If stays are not round, their cross-section must be such that stress per sq. in. caused by the load Q nowhere exceeds one 5.75th part of the tensile strength, after deducting $\frac{1}{16}$ in. all round as an allowance for corrosion or wear. For high working pressures, such as used in triple expansion engines, it is recommended to screw all stays into the plates they support in addition to fitting them with nuts. **Welded stays and stay tubes** are not allowed, with the exception that it is recommended that nuts should not be fitted in combustion chambers.

The diameter of **fire-box stays** is determined by:

$$D = 0.177 + \sqrt{\frac{Q}{300T}}$$

The diameter of **margin stays**:

$$D = 0.177 + \sqrt{\frac{Q}{260T}},$$

where D, Q and T have same meaning as above.

FURNACES

Oliver Evans, a progressive American engineer, and a great advocate of high steam pressures, seems to have recognized the inherent strength of the cylindrical furnace. About 1800, or earlier, he suggested a design of boiler to be good for 120 lbs. steam pressure, consisting of "Two cylinders, one inside the other: the space between them contains water, and the inner one contains fire which is surrounded by water." This appears to be the first conception of the cylindrical furnace, which, as will be seen on reference to Fig. 5, p. 336 (Smith's boiler), was early recognized as a desirable feature in internally fired boilers of the Scotch type. With the low pressures in vogue on the introduction of the first designs, there was no difficulty in constructing these furnaces of riveted metal sheets. It is to be noted, however, that plates exposed to the direct action of the flame cannot be made thicker than about $\frac{3}{4}$ in., on account of the danger of overheating. Plates as thick as $\frac{7}{8}$ in. have been used in the past, but it seems that there is a strong tendency for the fire side of the plate to burn off until the final thickness is $\frac{3}{4}$ in. or less. As steam pressures increased, it therefore became necessary to stay the sheets of cylindrical furnaces against collapsing pressure. This was at first done by means of metal rings fitted around the outer circumference (water side) of the furnace, with rivets connecting the rings with the plates and supporting them at certain specified distances. Direct contact of the ring with the metal of the plate was prohibitive on account of the inability to transmit the heat rapidly enough, so clearance was provided between the ring and the plates, ferrules being used to secure alignment.

FIG. 37.

FIG. 38.

Another method of securing the desired stiffness, as well as a certain amount of needed longitudinal elasticity, was by the use of the "Bowling hoop," riveted to several short cylindrical sections, Fig. 37. This had the disadvantage of introducing rivets and a double thickness of metal in contact with the fires. The Adamson ring, Fig. 38, was patented in 1851. This proved to be a very satisfactory method of staying the furnace sheets, as the rivets are well removed from contact with the fire. Longitudinal elasticity is secured by making the flanges of a sufficiently large radius, and the use of the ring between the plates also provides a ready means for calking the

joint. The Adamson ring is the most satisfactory device for constructing cylindrical furnaces by the use of riveted joints.

The corrugated furnace was patented in this country by Montgomery, in 1852, but was not generally introduced into marine service until some twenty-five years later, when welding processes had been perfected, and the Fox, Fig. 39, and Purves, Fig. 40, furnaces were introduced in England.

Various forms of corrugations have been brought forward from time to time, the objects of which have been generally to facilitate cleaning when in service, to simplify manufacturing processes or strengthen the construction. The principal types are the Holmes Furnace shown in Fig. 41; the Farnley furnace, Fig. 42, similar to the Fox furnace, except that the corrugations take the form of a helix around the cylinder; Brown's cambered section, Fig. 43,

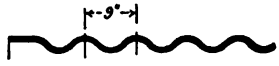


FIG.39. FOX TYPE

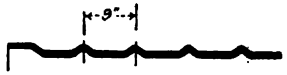


FIG.40. PURVES TYPE

FIG.41. HOLMES TYPE

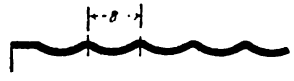


FIG.45. MORISON TYPE.

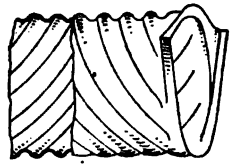


FIG.42. FARNLEY FURNACE

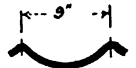


FIG.43. BROWN'S CAMBERED SECTION

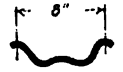


FIG.44. DEIGHTON TYPE.

the Deighton furnace, Fig. 44, Leeds Suspension Bulb, and the Morison suspension furnace, Fig. 45. The Morison furnace, which is very extensively used in this country, takes its name from the inventor and from the fact that the metal "suspended" between the adjacent corrugations takes the form of a catenary. Various claims are put forward for the various types, but as they all present effective means for increasing the strength of the furnace against collapsing pressure, and also provide adequate means for allowing for longitudinal expansion when heated, the questions of material and workmanship and facility for cleaning are the salient features to consider.

The corrugated furnace as used in marine work may be fitted with either plain ends, as shown in Fig. 59, Type *D*, or with the inner end flanged and of the so-called **"horse-collar" form**, the object of which is to facilitate removal through the front of the boiler shell, as shown in Fig. 59, Type *C*.

The U. S. Supervising Inspectors require for furnaces that the tensile strength of steel used in the construction of corrugated or ribbed furnaces shall not exceed 67,000 and be not less than 54,000 lbs.; and in all other furnaces the minimum tensile strength shall be not less than 58,000 and the maximum not more than 67,000 lbs. The minimum elongation in 8 in. shall be 20 per cent.

All corrugated furnaces having plain parts at the ends not exceeding 9 in. in length (except flues especially provided for), when new, and made to practically true circles, shall be allowed a steam pressure in accordance with the following formula:

$$\text{Pressure in lbs. per sq. in.} = \frac{C \times T}{D}$$

In which T = Thickness in in. not less than $\frac{5}{16}$ in., except that in the Purves type and type having sections 18 in. long, T shall not be less than $\frac{7}{16}$ in.

D = Mean diam. in in. except that in the Purves type and the Brown type D = least outside diam. in in.

C = A factor, depending upon the type of furnace and character and dimensions of the corrugations, as follows:

Leeds Suspension Bulb Furnace. C = 17,300 when corrugations are not more than 8 in. from centre to centre and not less than $2\frac{1}{4}$ in. deep.

Morison Corrugated Type. C = 15,600 when corrugations are not more than 8 in. from centre to centre and the radius of outer corrugations is not more than $\frac{1}{2}$ of the suspension curve. In calculating the mean diam. of the Morison furnace; Mean diam. = least inside diam. + 2 in.

Fox Type. C = 14,800 when corrugations are not more than 8 in. from centre to centre and not less than $1\frac{1}{2}$ in. deep.

Purves Type. C = 14,000 when rib projections are not more than 9 in. from centre to centre and not less than $1\frac{3}{8}$ in. deep.

Brown Type. C = 14,000 when corrugations are not more than 9 in. from centre to centre and not less than $1\frac{5}{8}$ in. deep.

The thickness of corrugated and ribbed furnaces shall be ascertained by actual measurement. The manufacturer shall have said furnace drilled for a $\frac{1}{4}$ in pipe tap and fitted with a screw plug that can be removed by the inspector when taking this measurement. For the Brown and Purves furnaces the holes shall be in the centre of the second flat; for the Morison, Fox, and other similar types in the centre of the top corrugation, at least as far in as the fourth corrugation from the end of the furnace.

Type having sections 18 in. long. C = 10,000, when corrugated by sections not more than 18 in. from centre to centre and not less than $2\frac{1}{2}$ in. deep, measuring from the least inside to the greatest outside diam. of the corrugations and having the ends fitted one into the other and substantially riveted together, provided that the plain parts at the ends do not exceed 12 in. in length.

Adamson Type. When plain horizontal flues are made in sections not less than 18 in. in length and not less than $\frac{5}{16}$ in. thick, and flanged to a depth of not less than 3 times the diam. of rivet holes plus the radius at furnace wall (inside diam. of furnace), the thickness of the flanges shall be as near the thickness of the body of the plate as practicable.

The radii of the flanges on the fire side shall be not less than 3 times the thickness of plate.

The distance from the edge of the rivet hole to the edge of the flange shall be not less than the diam. of the rivet hole, and the diam. of the rivets before driven shall be at least $\frac{1}{4}$ in. larger than the thickness of the plate.

The depth of the ring between the flanges shall be not less than 3 times the diam. of the rivet holes, and the ring shall be substantially riveted to the flanges. The fire edge of the ring shall terminate at or about the point of

tangency to the curve of the flange, and the thickness of the ring shall be not less than ½ in.

The pressure allowed shall be determined by the following formula:

$$\text{Working pressure, lbs. per sq. in.} = \frac{87.6}{D}[(18.75 \times T) - (1.03 \times L)]$$

where D = outside diam. of furnaces in in., or outside *mean* diam. of cone top in in.
L = length of section or flue in in.
T = thickness of furnace or cone top in sixteenths of an in., not to be less than $\frac{5}{16}$ in.

Cylindrical riveted flues and furnaces made in sections of not less than 18 in. in length fitted one into the other and substantially riveted, combustion chambers for vertical submerged tubular boilers in the shape of a frustum of a cone, constructed to a practically true circle, shall be allowed a steam pressure according to the following formula:

$$\text{Working press. in lbs. per sq. in.} = \frac{51.5}{D}[(18.75 \times T) - (1.03 \times L)]$$

where D, T and L had same meaning as above.

When the diam. of plain furnaces and flues used in vertical type of boilers or mean diam. of cone tops exceeds 42 in., they shall be deemed a flat surface and must be stayed in accordance with rules governing flat surfaces. If a greater working press. than given by formula is desired for mean diam. under 42 in., the flues or cone tops shall be substantially stayed for such additional pressure.

American Bureau of Shipping requires for **plain circular furnaces and furnaces with Adamson rings more than 20 in. apart:**

$$\text{Working pressure, lbs. per sq. in.} = \frac{51.5}{D}[(18.75 \times T) - (L \times 1.03)]$$

for **Suspension bulb furnaces:**

$$\text{Working pressure, lbs. per sq. in.} = \frac{1250 \times (T - 2)}{D}$$

for **furnaces with Adamson rings pitched not more than 20 in. apart;** or corrugated, ribbed and suspension furnaces:

$$\text{Working pressure, lbs. per sq. in.} = \frac{1160 \times (T - 2)}{D}$$

where T = thickness of plate in sixteenths of an in.
L = length of furnace, or pitch of stiffening rings in in.
D = smallest outside diam. of furnace in in.

Lloyd's Register of Shipping requires that the strength of **plain furnaces** be calculated as follows:

Where the length of plain cylindrical part of furnace exceeds 120 times thickness of plate:

$$\text{Working pressure, lbs. per sq. in.} \frac{1,075,200 \times T}{L \times D}$$

Where length of plain cylindrical part is less than 120 times thickness of plate:

$$\text{Working pressure, lbs. per sq. in.} \frac{50 \times (300T - L)}{D}$$

where D = outside diam. of furnace in in.
T = thickness of plates in in.
L = length of plain cylindrical part in in. measured from the centres of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the commencement of the curvature of the flanges of the furnace where it is flanged or fitted with Adamson rings.

In the furnaces referred to below the formulæ given are applicable if the steel used has a tensile strength of not less than 26 nor more than 30 tons per sq. in. If the material of furnaces has a less tensile strength than 26 tons per sq. in., then for each ton per sq. in. which the minimum tensile strength falls below 26, the coefficient is to be correspondingly decreased by ½₀th part.

The strength of corrugated furnaces made on **Fox's, Morison's, Deighton's,** or Beardmore's plan, to be calculated from

$$\text{Working press. in lbs. per sq. in.} = \frac{1259 \times (T - 2)}{D}$$

The strength of **spirally corrugated furnaces** is to be calculated from the formula:

$$\text{Working press. in lbs. per sq. in.} = \frac{912 \times (T - 2)}{D}$$

Where T = thickness of plate in sixteenths of an in.
and D = outside diam. of corrugated furnaces, in in.

The strength of the **Improved Purves'** furnaces with ribs 9 in. apart, and of **Brown's Cambered furnaces** with ribs either 8 in. or 9 in. apart, is to be calculated from the formula:

$$\text{Working press. in lbs. per sq. in.} = \frac{1160 \times (T - 2)}{D}$$

Where T = thickness of plate in sixteenths of an in.
and D = smallest outside diam. of furnaces, in in.

The strength of the **Leeds Forge bulb** furnace is to be calculated from the formula:

$$\text{Working press. in lbs. per sq. in.} = \frac{1259 \times (T - 2)}{D}$$

Where T = thickness of plate in sixteenths of an inch.
and D = smallest outside diam. in in.

The strength of **Holmes' patent** furnaces, in which the corrugations are not more than 16 in. apart from centre to centre, and not less than 2 in. high, to be calculated from the formula:

$$\text{Working press. in lbs. per sq. in.} = \frac{945 \times (T - 2)}{D}$$

Where T = thickness of plain portions of furnace in sixteenths of an inch.
and D = outside diam. of plain parts of the furnace in in.

Bureau Veritas rules for furnaces require that when **plain furnaces** are made as truly circular as practicable, and of steel having a tensile strength of not less than 26 tons per sq. in., the working pressure P and thickness of plate T may be calculated by the following formulæ:

$$\text{Working press. in lbs. per sq. in.} = \frac{16{,}000T - 60L}{D}$$

where T is the thickness of plate in in.
D is the external diam. in in.
L is the length in in.

When the furnace is in one length, L is measured from the centre of rivets at furnace mouth to those connecting back end to the tube plate, or to the commencement of flanging where back end of furnace is flanged. When the furnace is divided into two or more by Adamson joints, L is measured to the centre of joint.

For Iron or low tensile steel the constant C to be reduced to 14,400.

Furnace plates should not exceed ¹³/₁₆ in. in thickness.

The plate thickness is to be found by the following formulæ.

For **Corrugated furnaces:** $T = \frac{PD}{1260} + 2.$

Where P is the working pressure in lbs. per sq. in.
T is the thickness of plate in sixteenths of an inch.
D is the outside diameter in in. measured on the top of the corrugations.

The formula applied to Fox's, Morison's, or Deighton's furnace of standard type, for **ribbed furnaces**, when manufactured to the satisfaction of the administration: $T = \frac{PD}{1160} + 2$. Where P and T are as above.

D is the greatest outside diam. between the ribs in in. The formula applies to ribs spaced 9 in. and projecting 1⅜ in., the difference between the greatest and the smallest diam. in any part of the furnace not exceeding 0.003.

For Bulb furnaces: $T = \frac{DP}{1260} + 2$.

Where D is the outside diam. in in. measured between the bulb.

The coefficients in the above formulæ apply to the case where the tensile strength of the material is 26 tons per sq. in. or above. When it is below 26 tons, the coefficient is to be reduced by ½₆th for each ton below 26.

COMBUSTION CHAMBER GIRDERS

Notation:

W = extreme width of combustion box in in.
P = pitch of supporting bolts in in.
D = distance between girders from centre to centre in in.
d = depth of girder at the centre in in.
T = thickness of girder at centre in in. (sum of both plates).
n = number of stays fitted with each girder.
L = length from tube plate to tube plate, or from tube plate to back of combustion chamber, in in.

U. S. Supervising Inspectors require that for girders over back connection and other flat surfaces:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times d^2 \times T}{(W - P) \times D \times L}.$$

C = 550 when the girder is fitted with 1 supporting bolt.
C = 825 when the girder is fitted with 2 or 3 supporting bolts.
C = 917 when the girder is fitted with 4 or 5 supporting bolts.
C = 963 when the girder is fitted with 6 or 7 supporting bolts.
C = 990 when the girder is fitted with 8 or more supporting bolts.

American Bureau of Shipping requires for girders for combustion chamber tops:

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times d^2 T}{D \times L^2} \times \frac{(n+1)^2}{n(n+2)},$$

When n = 2, 4, or 6 and

$$\text{Working pressure, lbs. per sq. in.} = \frac{C \times d^2 T}{D \times L^2}, \text{ when } n = 1, 3, \text{ or } 5.$$

Where C = 14,500 for steel and 13,180 for iron.

Lloyd's Rules require that the strength of girders supporting the tops of combustion chambers and other flat surfaces be taken from the formula:

$$\text{Working pressure in lbs. per sq. in.} = \frac{C \times d^2 \times T}{(L - P) \times D \times L}$$

C for wrought iron = 6,000, if there is one stay to each girder.
9,000, if there are two or three stays to each girder.
10,000, if there are four or five stays to each girder.
10,500, if there are six or seven stays to each girder.
10,800, if there are eight stays or above to each girder.

C for wrought steel = 7,110, if there is one stay to each girder.
10,660, if there are two or three stays to each girder.
11,850, if there are four or five stays to each girder.
12,440, if there are six or seven stays to each girder.
12,800, if there are eight stays or above to each girder.

If the minimum tensile strength of girder plates is other than 28 tons per sq. in., these values of C should be correspondingly modified.

Bureau Veritas rules require that the strength of girders on the tops of combustion chambers shall be determined as follows:

$$\text{Working pressure, lbs. per sq. in.} = \frac{Cd^2t}{(W - P)DL}$$

Where $C = \frac{13{,}200n}{n+1}$ for odd numbers of bolts.

$C = \frac{13{,}200(n+1)}{n+2}$ for even numbers.

COMBUSTION CHAMBER TUBE SHEET

U. S. Supervising Inspectors rules require that pressure allowable on tube sheets where combustion chambers are not suspended from the shell of the boiler shall be determined by the following formula:

$$\text{Working pressure, lbs. per sq. in.} = \frac{(D - d)T \times 27{,}000}{W \times D}$$

The compressive stress on tube plates, as determined by the following formula, shall not exceed 13,500 lbs. per sq. in., when pressure on top of combustion chamber is supported by vertical plates of such chamber:

$$C = \frac{P \times D \times W}{2(D - d)T}$$

Where C = stress on tube sheet; D = least horizontal distance between tube centres in in.; d = inside diam. of tube in in.; W = extreme width of combustion chamber in in.; T = thickness of tube sheet in in.

American Bureau of Shipping rules require that tube plates in way of nests of tubes are to be governed by the formula:

$$\text{Working pressure in lbs. per sq. in.} = \frac{280 \times T^2}{P^2 + p^2}$$

Where T = thickness of plate, in sixteenths of an in.; P = mean length of area supported, measured between centres of tubes, in in.; p = mean breadth of area supported, measured between centres of tubes, in in.

For wide spaces between nests of tubes the formulæ for flat surfaces are to be used.

When girders are fitted to the tops of combustion chambers, the thickness of the tube plates is not to be less than given by the preceding formula nor than is found from the formula:

$$T = \frac{W \times L \times P}{(P - d) \times 1800}$$

Where W = working pressure, in lbs. per sq. in.; L = width of combustion chamber over the plates in in.; P = horizontal pitch of tubes, in in.; d = inside diameter of plain tube, in in.

Lloyd's rules require that for steel tube plates in the nest of tubes the strength is to be taken from

$$\frac{140 \times T}{P} = \text{working pressure in lbs. per sq. in.}$$

Where T = the thickness of the plates in sixteenths of an in.

P = the mean pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes the strength to be taken from

$$\frac{C \times T}{P} = \text{working pressure in lbs. per sq. in.}$$

Where P = the horizontal distance from centre to centre of the bounding rows of tubes, and
C = 120 where the stay tubes are pitched with two plain tubes between them and are not fitted with nuts outside the plates.
C = 130 if they are fitted with nuts outside the plates.
C = 140 if each alternate tube is a stay not fitted with nuts.
C = 150 if they are fitted with nuts outside the plates.
C = 160 if every tube in these rows is a stay tube and not fitted with nuts.
C = 170 if every tube in these rows is a stay tube and each alternate stay tube is fitted with nuts outside the plates.

The thickness of tube plates of Combustion Chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule:

$$T = \frac{P \times W \times D}{1750 \times (D - d)}$$

Where P = working pressure in lbs. per sq. in.
W = width of Combustion Chamber between plates in in.
D = horizontal pitch of tubes in in.
d = inside diameter of plain tubes in in.
T = thickness of tube plates in sixteenths of an inch.

CYLINDRICAL TUBE-SHEETS—WATER-TUBE BOILER SHELLS

The Supervising Inspectors' rule for the pressure allowable on **cylindrical tube-sheets**, or the shells of water-tube boilers when such shells have a row or rows of tubes inserted therein, is expressed as follows:

$$P = \frac{(D - d) \times T \times S}{D \times R}, \text{ in which}$$

P = Working press. allowable in lbs. per sq. in.
D = Distance in in. between the tube centres in a line from head to head.
d = Diam. of tube hole in in.
T = Thickness of plate in in.
S = One-sixth of the tensile strength of the plate in lb. per sq. in.
R = Radius of shell in in.

When **tubes** on any one row are **pitched unequally** nd must be substituted for d in the formula, where n = number of tube holes in a pitch.

Where **rows of tubes** are **pitched diagonally**, each diagonal ligament shall be not less than three-fifths of each longitudinal ligament.

The Supervising Inspectors also give a rule for **"porcupine" type boilers** having tube holes in vertical cylindrical shells, in which the strength of the **diagonal ligament** is treated directly as follows:

CYLINDRICAL SHELLS—DIAGONAL LIGAMENTS

$$P = \frac{E \times T \times S}{C \times R}, \text{ in which}$$

C = Vertical distance in in. between centres of rows of tubes, and
E = Width of diagonal ligament in in.

An **analysis of the stresses on a cylindrical tube-sheet having diagonal ligaments between tube holes** shows that there is a shearing action

on the metal in addition to that of tension. (See Fig. 46 which shows a section of a tube-sheet of this character which has been strained above the elastic limit.) This subject has been carefully studied by *Mr. J. E. Black*, under the supervision of *Mr. W. A. Jones*, both of The Babcock & Wilcox Company. Mr. Black's curves for calculating the strength of diagonal ligaments are given in Fig. 47, and his analysis of the problem follows:

"These curves apply only to a cylindrical surface subject to pressure, in which case the circumferential tension has twice the intensity of the longitudinal tension. With this condition, Rankine's Ellipse of Stress enables us to determine the intensity of tension and of shear on any plane. These

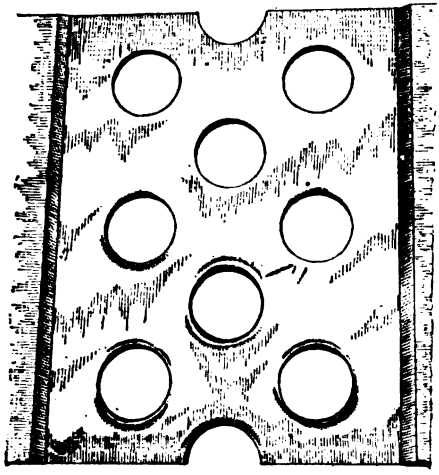


FIG. 46.

intensities multiplied by the distance between the centers of the holes which form the diagonal ligament gives total tension and total shear on the ligament.

"The total tension on the ligament divided by the cross-sectional area at any point of the ligament gives the stress in the ligament due to tension.

"The plane, passing through the centers of the two holes which form the ligament, divides the ligament into two cantilevers with a point of contra-flexure at the plane, and evidently the amount of bending moment on each of these cantilevers at any section is the product of the total shear times the distance of the particular section from the dividing plane.

"For any section this bending moment divided by the modulus of the section gives stress due to bending. The total stress on that section is the sum of the stress due to direct tension on that section plus the stress due to bending on that section, having in mind that both the area and the modulus of any section are dependent upon its width and that the bending moment on the section increases with the distance of the section from the point of contra-flexure.

"The maximum total stress is found by trial by figuring the stresses due to tension and due to bending on each of several sections at one side of the

point of contra-flexures and placing the results in the form of a table which will show a gradual increase of total stress up to a maximum and then a decrease.

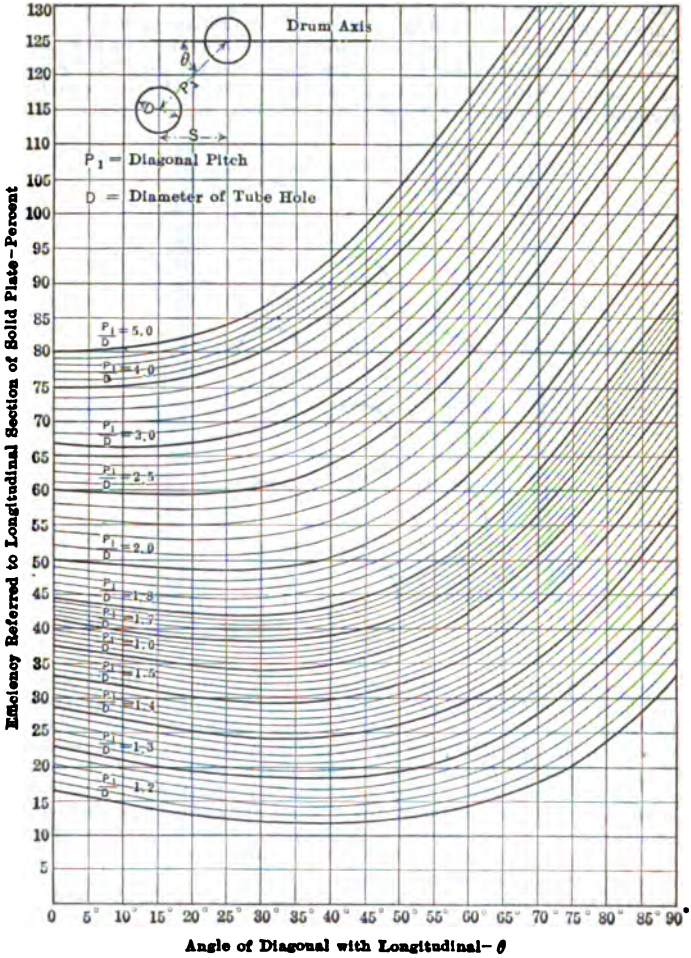


FIG. 47.—Diagonal Ligament Chart.

"The relation between the maximum stress in the solid plate on a longitudinal section and the maximum stress in the diagonal ligament as determined in this way gives the efficiency as shown by the curve.

"The center-to-center distance between any two holes which form a diagonal ligament multiplied by the cosine of the angle which their common centre line makes with the longitudinal, multiplied by the efficiency of the ligament, gives the equivalent width of solid plate on a longitudinal section.

"With these curves, for instance, it is possible to figure the equivalent width of solid plate on a longitudinal section for each ligament at one side of the longitudinal centre line for a circular nozzle riveted to a drum shell, and the sum of all these equivalent widths is the equivalent remaining metal available to transmit stress."

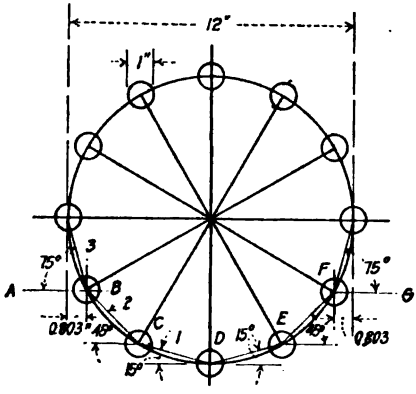


FIG. 48.

To Determine a Point on Curve of Efficiency of a Diagonal Ligament: Let D = diameter of holes; P_1 = distance between centre line of holes; θ = angle which the normal plane through centres of holes makes with the plane through the drum axis; Y = distance of any plane parallel to the normal plane passing the centres of holes from that plane; T = thickness of plate. From Rankine's Ellipse of Stress, we have:

Intensity of tension on any plane $= \dfrac{\cos^2\theta + 1}{2}$

Intensity of shear on any plane $= \dfrac{\cos\theta \sin\theta}{2}$

Total tension $= \dfrac{\cos^2\theta + 1 \times P_1 T}{2}$, Total shear $= \dfrac{\sin\theta \cos\theta \times P_1 T}{2}$

Width of ligament $= P_1 - \sqrt{D^2 - 4Y^2}$

Bending moment $= S \times Y$; Section Modulus $= \dfrac{(P_1 - \sqrt{D^2 - 4Y^2})^2 \times T}{6}$

Stress due to tension $= \dfrac{\cos^2\theta + 1}{2} P_1 T + (P_1 - \sqrt{D^2 - 4y^2}) \times T$

Stress due to bending $= \dfrac{\sin\theta \cos\theta}{2} \times P_1 T Y + \dfrac{(P_1 - \sqrt{D^2 - 4Y^2})^2 \times T}{6}$

Total stress = Stress due to tension plus stress due to bending.

$$\text{Efficiency} = \frac{\text{Maximum stress in solid plate}}{\text{Maximum stress in diagonal ligament}}.$$

Example: Assume $\theta = 45$ *deg.;* $D = 1$ *in.;* $P_1 = 2$ *in.;* $T = 1$ *in.;*

$$\text{Total tension} = \frac{\left(\frac{1}{\sqrt{2}}\right)^2 + 1}{2} \times 2 = 1.5$$

$$\text{Total shear} = \frac{\frac{1}{\sqrt{2}} \times \frac{1}{\sqrt{2}}}{2} \times 2 = 0.5$$

Y	Length of ligament	Section modulus of ligament	Moment	Total tension	Stress due to bending	Stress due to tension	Total stress
0	1.000	0.1667	0	1.500	0	1.500	1.500
0.05	1.005	0.1683	0.025	1.500	0.148	1.492	1.650
0.10	1.020	0.1734	0.050	1.500	0.288	1.471	1.759
0.12	1.029	0.1765	0.060	1.500	0.340	1.454	1.794
0.14	1.040	0.1803	0.070	1.500	0.388	1.442	1.830
0.16	1.053	0.1848	0.080	1.500	0.433	1.424	1.857
0.18	1.067	0.1897	0.090	1.500	0.474	1.406	1.880
0.20	1.084	0.1958	0.100	1.500	0.511	1.384	1.895
0.22	1.102	0.2024	0.110	1.500	0.543	1.360	1.903
0.24	1.123	0.2102	0.120	1.500	0.570	1.335	1.905
0.26	1.146	0.2189	0.130	1.500	0.594	1.309	1.903
0.5	2.000	0.6667	0.25	1.500	0.375	0.75	1.125

$$\text{Efficiency} = \frac{1}{1.905} = 0.525$$

$P_1 = 6 \times \sin 15 \text{ deg.} \times 2 = 3.10,$

$\frac{P_1}{D} = 3.10$

From curves—

Efficiency of ligament No. 1 = 0.672
Efficiency of ligament No. 2 = 0.783
Efficiency of ligament No. 3 = 1.133

Equivalent solid plate—

(1) $3.10 \times \cos 15 \text{ deg.} \times 0.672 = 2.01$
(2) $3.10 \times \cos 45 \text{ deg.} \times 0.783 = 1.72$
(3) $3.10 \times \cos 75 \text{ deg.} \times 1.133 = 0.909$

Upon examination of the foregoing, it is evident that the plate will break on the line *A, B, C, D, E, F, G.*

To find the efficiency of the drum plate: If two similar nozzles are placed 20 in. centre to centre—

Equivalent solid plate $= 20 - 13 + 2 \times 0.803 + 2 \times (2.01 + 1.72) = 16.066.$

$$\text{Efficiency} = \frac{16.066}{20} = 0.8033.$$

NOTES ON CONSTRUCTION OF SCOTCH BOILERS

The following notes were prepared by *Mr. Ernest L. Woolley, Asst. Superintendent, Bethlehem Shipbuilding Corporation, Ltd., Field's Point Works:*

Shell. In boilers above the smallest sizes it will be found necessary to make the shell plate of two strakes or lengths due to the difficulty of securing wide plates. Very few mills can roll wide boiler plates due to the spring

of the rolls and when it is remembered that boilers have been constructed with shell plates 1¾ in. thick the difficulty in rolling wide plates will be appreciated. In order to keep this thickness as low as possible for considerations of weight and ease of working material in the shop, the efficiency of the longitudinal seams should be kept as high as possible and this necessitates a butt joint with double buttstraps treble riveted giving an efficiency of from 80 per cent to 85 per cent. As a boiler is fundamentally twice as strong circumferentially as longitudinally the circumferential seams must have an efficiency at least equal to one-half the efficiency of the longitudinal seams and it is usual to somewhat better this figure and give the circumferential seams an efficiency of about 50 per cent. From these considerations it is necessary to have the circumferential seams double riveted lap joints and

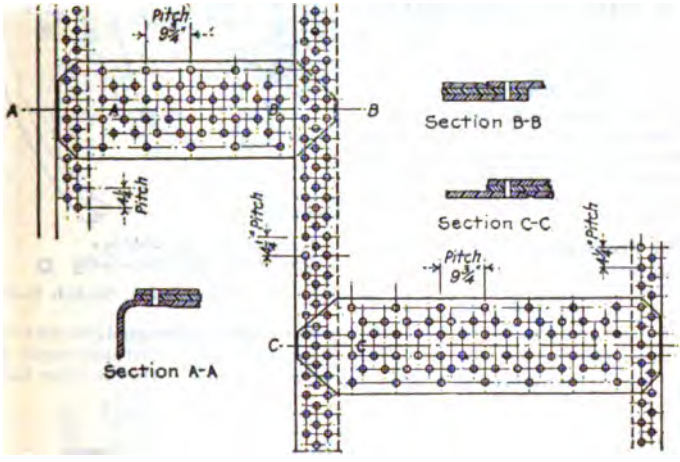


FIG. 49.—Riveted Seams, Scotch Boilers.

some designers even make the mid circumferential seam treble riveted lap (if shell is in two strakes) with an efficiency of about 65 per cent due to the unsupported nature of the shell at this point. Fig. 49 shows a development of the longitudinal and circumferential seams of a boiler embodying these features.

The thickness of the shell, portions of the riveted joints such as pitch, distance between rows of rivets, thickness and width of buttstraps, etc., will be fixed by the rules on pp. 393 to 410.

The Front Tube Plate of the boiler is a flat surface and as such must be efficiently supported. This is accomplished by fitting stay tubes among the ordinary expanded tubes, these stay tubes being extra thick boiler tubes with screwed ends, the ends being screwed into the front tube-plate and combustion chamber tube-plate respectively. Fig. 50 shows a typical arrangement of tubes in tube-plate and the position of stay tubes will be noted. The area supported by each stay tube is indicated by dotted lines. At positions marked "A" the stay tubes are called upon to support an extra large surface and this

is accomplished by fitting in these positions "**extra strong stay**" **tubes** which are simply stay tubes of greater thickness than the regular stay tubes. They are frequently fitted with nuts on the outside of the tube-plates in addition to being screwed through the sheets.

Fig. 51 shows these various types of boiler tubes. The ordinary boiler tubes are expanded into the tube-sheets and the ends beaded over, but no supporting effect on the tube-sheets is figured on the ordinary tubes the assumption being that this load is carried entirely by the stay tubes.

The staying of the tube-plate will fix the thickness of this plate, as obviously the alternative exists of fitting a thick tube-sheet and few stays or a thin sheet well supported. The rule governing this situation will be found on p. 410. The cross-sectional area of the stay tube should be figured at the least cross-section of the tube, usually the bottom of the thread.

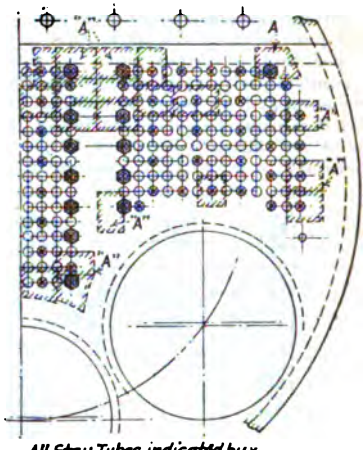


FIG. 50.—Tube Sheet, Scotch Boiler.

The Front Head of a Scotch boiler is made in one, two or three plates, the number depending on the opinion of the designer and on the equipment of the shop making the boiler. Generally speaking, the fewer riveted joints a boiler has the better the boiler is, as such joints are the places corrosion starts and rivets and riveted joints have always a tendency to leak. On the other hand,

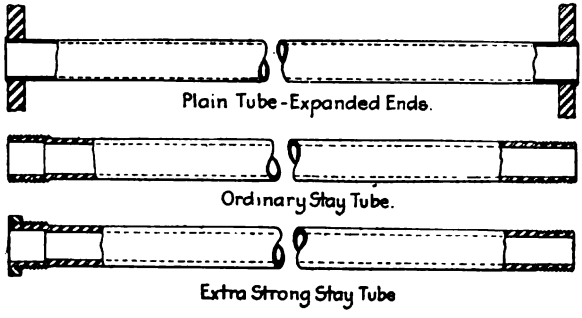
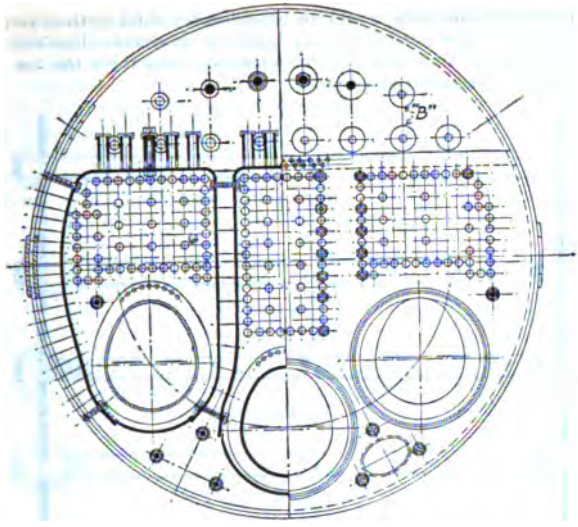


FIG. 51.—Types of Scotch Boiler Tubes.

should one plate only be used in making a head, the thickness would have to be determined by the part of the head which was least efficiently stayed and this in all cases will be the upper part of the head in way of the steam space. Should weight of boiler be of no importance, it would be desirable to use one



Half Sectional Elevation. Half Front Elevation.

FIG. 52.

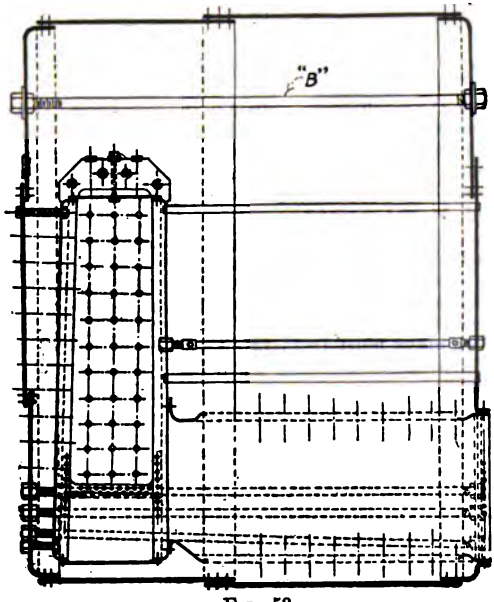


FIG. 53.

plate, even though this plate would be considerably thicker than required in certain portions of the head, but only heads of moderate diameter can be obtained from the makers, and this consideration may force the use of more than one plate in constructing a head.

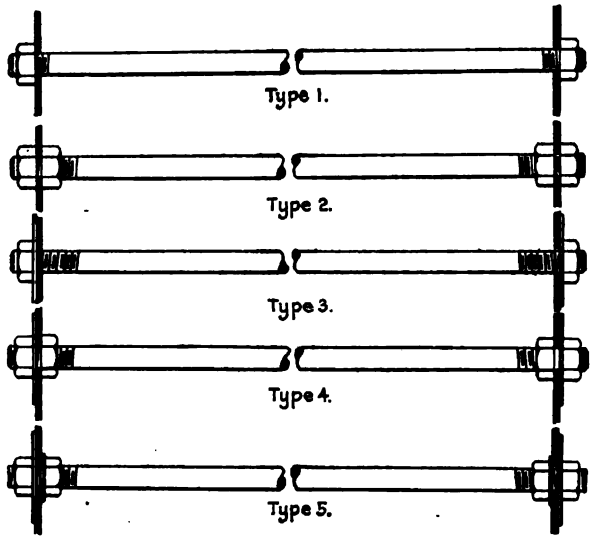


FIG. 54.—Methods of Securing Scotch Boiler Stays.

Examining then the conditions involved in the upper front and rear heads of a boiler and referring to Figs. 52 and 53 it will be seen that these parts of the boiler are supported by horisontal stay bars "B." These are not screwed through the heads but are passed through clearance holes and secured by nuts and washers. Various methods of securing these stay bars to the heads are shown in Fig. 54, the method adopted varying with individual opinion. Fig. 55 shows the setting out of these stay bars on the heads and also shows, by dotted lines, the surface supported by each stay. These stays must be widely spaced in order to allow freedom of inspection and ample room must be provided between the stays for a man to pass through. The pitch of the stays must also be arranged to clear the girder stays on the combustion chamber tops. In figuring the area supported by a stay the heel of the small radius of the head is considered as a row of stays and the area supported by a stay bar will be divided between this imaginary row of stays and the nearest

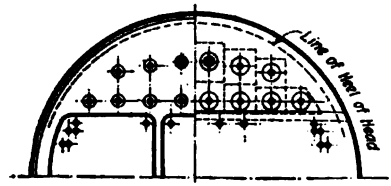


FIG. 55.—Laying out Stayed Heads.

stay bar. On Fig. 55 the line of the heel of the head radius appears as a dotted line and the method of determining the area supported by each stay bar with reference to this line can be followed. The thickness of the upper front and rear heads would be fixed by the rules on p. 410. The cross-sectional area of the stays will be determined by the rules on p. 414. The ends of the stay bars may be upset to a sufficient diameter for the depth of thread. Steel stays must not be forged or welded but iron stays may be welded.

Lower Back Head. A typical arrangement of the lower back head plate is shown in Fig. 56 from which it will be seen that this plate is supported chiefly by the stays connecting this plate to the combustion chamber. The area supported by these stays is shown in dotted lines and the thickness of the plate will be fixed by the rule on p. 410. The cross-sectional area of the stays will be determined by the rule on p. 414.

Stays 1 in. diam. and above can be welded when made of iron but may not be welded when made of steel. The ends may be upset to allow for the depth of the thread and the least cross-sectional area of the stay should be taken at the threaded part should this be the minimum diameter.

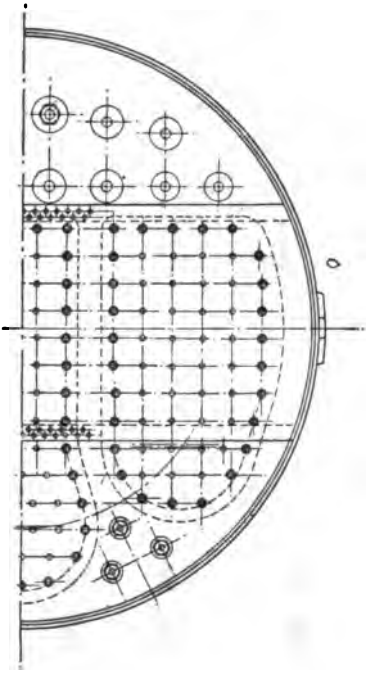


FIG. 56.—Lower Back Head Plate.

Combustion Chambers. The number of combustion chambers is generally the same as the number of furnaces but in boilers containing two furnaces only it is quite common to find one combustion chamber serving both furnaces. The combustion chamber consists of the tube sheet, wrapper sheet, back sheet and bottom plate. Fig. 57 shows wing and center combustion chambers and, referring also to Fig. 52, the method of securing the combustion chambers to the shell is seen. The tube-sheet and back sheet are both flanged inward around the rim to make the joints with the combustion chamber wrapper sheet. These joints are single riveted with rivets close enough together to allow efficient calking. The thickness of the front tube-sheet is determined by the pitch of the stays by the method already explained on p. 410.

The thickness of the back plate of the combustion chamber will be determined by the pitch of the stays in the lower back head of shell, as explained under "Lower Back Head," above. The pitch and area of these stays being already fixed by considerations involved in the shell, it only remains to make the combustion chamber plates of the necessary thickness to suit. The

thickness of the combustion chamber wrapper sheet will be fixed by the arrangement of the stays attaching the combustion chambers to shell and also to adjacent combustion chambers. For rule, see p. 410.

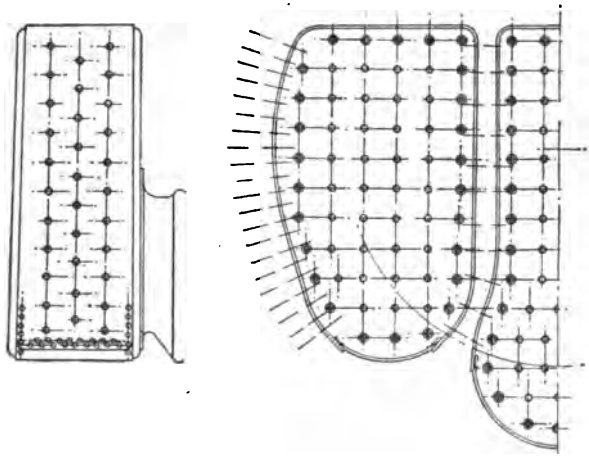


FIG. 57.—Combustion Chambers, Scotch Boilers.

Fig. 58 shows typical methods of securing **stay bolts** to plates and also illustrates various types of stays in common use. It should be noted that all stays have a hole $\frac{3}{16}$ in. in diameter drilled from the outside. In the event of a stay breaking it will be immediately noticed by the leakage of water

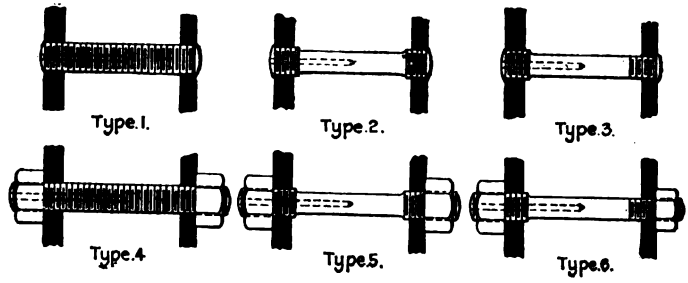


FIG. 58.—Types of Stay Bolts and Methods of Fastening Same.

from the hole. Referring again to Fig. 52 and Fig. 57 it will be seen that the bottom of the combustion chamber is an unsupported area and two methods are used to overcome this difficulty. An extra thick plate can be fitted in this position sufficiently strong to resist the collapsing pressure.

Alternatively the round bottom of the combustion chamber may be stiffened by angles.

Where rounded bottoms of combustion chambers are stiffened with single angle-iron stiffeners, such angles shall have a thickness of leaf 0.8 that of the plate and a depth of at least ½ pitch. Where stiffened with double angle irons or tee bars, such angles or tee bars shall have a thickness of leaf at least ⅝ that of plate and a depth of at least ¼ of pitch. Said angles or tee bars shall be substantially riveted to the plate supported. Where the bottoms of combustion chambers are strengthened by angles or tee irons, the same shall be on the water side of the combustion chambers.

The combustion chamber roofs are supported either by girder stays or by sling stays from the shell. The former method is most commonly used and consists of girders placed across the combustion chambers in the longitudinal

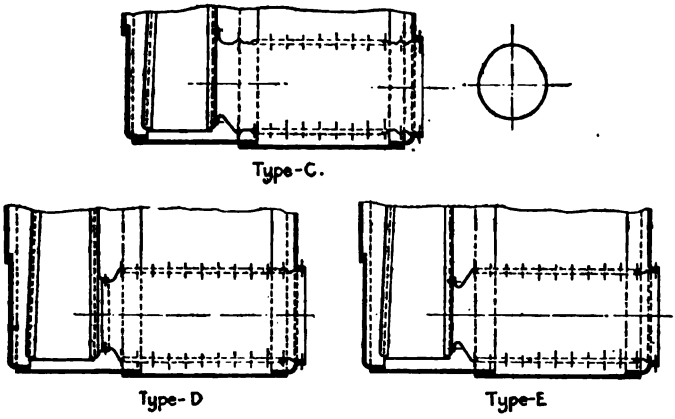


FIG 59.—Types of Scotch Boiler Back Connections.

direction, these girders resting on the top radius of the combustion chamber tube plate and back plate. Between these girders are stay bolts screwed through the combustion chamber roof and the top end of the stay being supported by washers bridging over two adjacent girders. (See Figs. 52 and 53.) The proportions of these girders are settled by the rule on p. 420.

Furnaces are almost exclusively of the corrugated type, this type being both strong and flexible and allowing freedom of expansion. Due to the fact that furnaces are peculiarly liable to accident due to the natural wear and tear caused by the nature of their work, the furnaces should be designed in such manner that they can be removed without disturbing any other part of the boiler. This means that the back connection must be able to pass through the opening in the front head of the boiler and several such back connections are shown in Fig. 59. The most common type is the horse-collar type, Type C in Fig. 59. The type of corrugations varies but all embody the same principle. Figs. 39 to 45 show types of corrugations in common use. The rules for corrugated furnaces are given on pages 415 to 420.

TUBES

Welded-iron pipes of small diameters are employed in certain classes of "water-tube boilers" in which threaded joints are used. Strictly speaking, these are the true "pipe boilers."

Seamless-drawn-steel and **charcoal-iron** tubes are used in standard designs of water-tube and fire-tube boilers. Charcoal-iron tubes may be used in fire-tube boilers where the metal is in compression, and they are frequently advocated for water-tube boilers on account of their ability to withstand corrosion better than the steel tube. Charcoal-iron tubes, however, cannot be made seamless, and objection to the use of a welded tube probably accounts for the trend toward the exclusive use of the seamless-steel tube in water-tube boilers. Improvement in material and a better understanding of the causes and means of prevention of corrosion have rendered the seamless open-hearth steel tube practically as durable as the charcoal-iron tube.

Cold-drawn-seamless-steel tubes have in the past usually been specified for the best classes of work, on account of their superior finish as compared with the **hot-finished** tube. However, a great improvement has recently been made, and hot-finished tubes of 2 inches in diameter and upward are now produced almost equal in finish to the cold-drawn tube, and, on account of the thoroughly homogeneous structure of the hot-finished tube, and the fact that the cold-drawn tube has to be very carefully **annealed** after drawing, and is more expensive in manufacture, the hot-finished tube is being very largely used to the exclusion of the cold-drawn tube. See Table 26,

The U. S. Supervising Inspectors of Steam Vessels permit the use of both lap-welded and seamless tubes of any length, and specify maximum working pressures allowable as per the following table:

Outside diam., in.	Thickness of material, in.	Thickness of material, B.W.G., number.	Maximum pressure allowed, lbs.
2	0.095	13	427
2¼	0.095	13	380
2½	0.109	12	392
2¾	0.109	12	356
3	0.109	12	327
3¼	0.120	11	332
3½	0.120	11	308
3¾	0.120	11	282
4	0.134	10	303
4½	0.134	10	238
5	0.148	9	235
6	0.165	8	199

Where heavier material is used, increased working pressure may be allowed as determined by a formula for lap-welded flues, as follows:

$$T = \frac{[(F \times P) + 1386] \times D}{86{,}670}$$, in which D = outside diam. of flue in in.; T = thickness of wall in in.; P = working pressure in lbs. per sq. in.; F = factor of safety = 5.

The Supervising Inspectors require that all tubes shall be carefully gaged with a Birmingham wire gage and shall not be less than the gage specified. As some tolerance is required in the manufacture of tubes, the Supervising Inspectors' rule probably results in tubes heavier than the specified gage being used in many cases.

Table 26. Tube Sizes and Weights of Open-hearth-steel Tubes

Diam., external, in.	B.W. gage	Minimum thickness, in.	Maximum internal diam., in.	Sq. ft. external surface per linear foot	Length in feet per sq. ft. external surface	Manufacturer's weight per linear foot, lbs.	Maximum internal pressure allowed, A.S.M.E. code, lbs. per sq. in.
1	14	0.083	0.834	0.2618	3.8197	0.9182	542
1	13	0.095	0.810	0.2618	3.8197	1.037	758
1	12	0.109	0.782	0.2618	3.8197	1.138	1010
1⅛	14	0.083	0.959	0.2945	3.3955	1.045	454
1⅛	13	0.095	0.935	0.2945	3.3955	1.183	646
1⅛	12	0.109	0.907	0.2945	3.3955	1.288	870
1¼	14	0.083	1.084	0.3272	3.0568	1.172	383
1¼	13	0.095	1.060	0.3272	3.0562	1.328	557
1¼	12	0.109	1.032	0.3272	3.0562	1.448	785
1½	13	0.095	1.310	0.3927	2.5463	1.619	422
1½	12	0.109	1.282	0.3927	2.5463	1.769	590
1½	11	0.120	1.260	0.3927	2.5463	1.955	722
2	12	0.109	1.782	0.5236	1.9098	2.409	380
2	11	0.120	1.760	0.5236	1.9098	2.671	479
2	10	0.134	1.732	0.5236	1.9098	2.927	605
2	9	0.148	1.704	0.5236	1.9098	3.234	731
2	8	0.165	1.670	0.5236	1.9098	3.499	884
2¼	12	0.109	2.032	0.5890	1.6978	2.730	310
2¼	11	0.120	2.010	0.5890	1.6978	3.028	398
2¼	10	0.134	1.982	0.5890	1.6978	3.323	510
2¼	9	0.148	1.954	0.5890	1.6978	3.674	622
2¼	8	0.165	1.920	0.5890	1.6978	3.979	758
2½	12	0.109	2.282	0.6545	1.5278	3.050	254
2½	11	0.120	2.260	0.6545	1.5278	3.386	333
2½	10	0.134	2.232	0.6545	1.5278	3.718	434
2½	9	0.148	2.204	0.6545	1.5278	4.115	535
2½	8	0.165	2.170	0.6545	1.5278	4.460	657
2¾	12	0.109	2.532	0.7199	1.3890	3.371	208
2¾	11	0.120	2.510	0.7199	1.3890	3.744	280
2¾	10	0.134	2.482	0.7199	1.3890	4.113	372
2¾	9	0.148	2.454	0.7199	1.3890	4.555	464
2¾	8	0.165	2.420	0.7199	1.3890	4.941	575
3	12	0.109	2.782	0.7854	1.2732	3.691	170
3	11	0.120	2.760	0.7854	1.2732	4.102	236
3	10	0.134	2.732	0.7854	1.2732	4.508	320
3	9	0.148	2.704	0.7854	1.2732	4.996	404
3	8	0.165	2.670	0.7854	1.2732	5.421	506
3¼	11	0.120	3.010	0.8508	1.1753	4.459	199
3¼	10	0.134	2.982	0.8508	1.1753	4.903	276
3¼	9	0.148	2.954	0.8508	1.1753	5.436	354
3¼	8	0.165	2.920	0.8508	1.1753	5.902	448
3½	11	0.120	3.260	0.9163	1.0913	4.817	167
3½	10	0.134	3.232	0.9163	1.0913	5.298	238
3½	9	0.148	3.204	0.9163	1.0913	5.877	310
3½	8	0.165	3.170	0.9163	1.0913	6.382	398
3¾	11	0.120	3.510	0.9817	1.0186	5.175	139
3¾	10	0.134	3.482	0.9817	1.0186	5.694	206
3¾	9	0.148	3.454	0.9817	1.0186	6.318	273
3¾	8	0.165	3.420	0.9817	1.0186	6.863	355
4	10	0.134	3.732	1.047	0.9551	6.089	178
4	9	0.148	3.704	1.047	0.9551	6.758	240
4	8	0.165	3.670	1.047	0.9551	7.344	317
4½	9	0.148	4.204	1.178	0.8489	7.639	186
4½	8	0.165	4.170	1.178	0.8489	8.305	254
5	8	0.165	4.670	1.309	0.7639	9.266	204

The weight given in Table 26 is known as the "manufacturer's weight," being the actual weight of tubes one gage heavier than listed. This is in accordance with the usual practice of rolling tubes nominally one gage heavier than specified, in order to insure all tubes being at least equal to the specified gage. The cold drawn tubes will run a little lighter than the hot-finished tube, for the reason that the drawing process brings the tube nearer to the actual gage.

While not so stated in the Supervising Inspectors' rules, it is evident that the data as given originally, refer to tubes for fire-tube boilers in which the tube is in compression. There is a tendency among manufacturers of marine water-tube boilers to allow internal working pressures on tubes in accordance with the rules of the A.S.M.E. Boiler Code. (See last column in Table 26.)

Tube Holes may be drilled full size from the solid plate, or, if guide-holes are used, the Supervising Inspectors specify that guide-holes for drills and cutters may be punched not to exceed 75 per cent of the diam. of the full-sized finished hole. The remainder shall be cleanly cut, drilled or reamed full size. Holes not to exceed 4 in. in diam. in plates $\frac{5}{8}$ in. thick and over may be punched to within $\frac{1}{4}$ in. of the full diam., the remainder to be cleanly cut, drilled or reamed to full size.

All burrs on both sides of the plate should be removed, and the edges of the finished hole slightly chamfered to receive the tube.

Tube Joints. Threaded tube joints have been used with more or less success, and also occasionally special machined or ground joints. Nothing, however, has proved so simple and reliable as the **expanded joint**, formed preferably by the action of the roller-expander on the inside surface of the tube. These joints are easily made up, and can be made tight against almost any pressure; while by the use of a half-round chisel, supplemented by a ripping-chisel, the tube can be easily loosened and removed without injury to the tube-seat. In all the most durable types of water-tube boilers, and in Scotch boilers, this joint is invariably employed. In Scotch boilers the heavy gage stay-tubes are threaded and screwed into the tube sheet and then expanded, while the ordinary tubes are expanded only. The tube should project through the tube-sheet $\frac{1}{4}$ in. to $\frac{1}{2}$ in. before expanding, and the rolls of the expander set to overlap the seat about $\frac{1}{8}$ in. on each side.

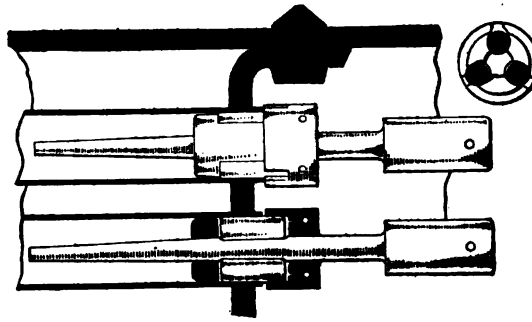


FIG. 60.—Method of Expanding Tubes.

A little roughness of the tube-seat or circumferential grooving is not objectionable, but is rather an advantage in making up this type of joint, though, obviously, grooving which runs entirely across the seat is disastrous, and must be removed by reaming, or, in some cases, by dovetailing or welding in a piece. An easy sliding fit between the tube and tube-sheet before expanding is desirable.

The amount of enlargement of the hole which is permissible depends upon the character of the tube and how much it may be swelled by the rollers of the expander without injury. It is customary to make the tube holes for a 4-in. tube about $4\frac{1}{16}$ in. diam. in new work. If the hole becomes enlarged by reaming or repeated expanding to $4\frac{3}{16}$ in., the joint can usually be made without trouble. If the diam. of the tube hole becomes

larger than that, it will be necessary to enlarge the tube by heating, swaging and annealing before making up the joint, or a soft iron Dutchman overlapping the tube-seat ½ in. on each side, may be inserted in the space between the tube and the tube-seat before expanding.

The perfection of electric welding has made possible another means of repair of enlarged tube holes. The new metal is built up on the old enlarged seat and bored ou to size, thus restoring the tube hole to its original proportions.

Beading and Flaring. In boilers of the fire-tube type in which the tube ends project into the space occupied by the hot gases, they must be beaded over into close contact with the tube-sheet. This not only protects the metal of the tube end from overheating, but also increases the resistance of the tube to pulling through the tube-sheet. Beading is not necessary in water-tube boilers where, in fact, the shoulder formed by the rollers of the expander if properly set and the rolling properly performed, answers every requirement as to strength, and the ends of the tubes being protected by water, there is of course no danger of overheating.

It is not uncommon in water-tube boilers to flare the end of the tube slightly by means of a tapered mandrel driven into the tube, this being a precaution against the tube pulling through the tube-seat in case of overheating due to low water. The diam. of the extreme end of the tube is usually increased about ½ in. in the flaring process.

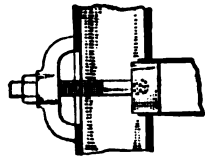


FIG. 61.—Plug Extractor.

Tube Plugs or stoppers fitted over the tube ends are used for cutting out of service a defective tube in both fire-tube and water-tube boilers. These plugs are usually of cast-iron, and designed to make a joint against the tube-sheet or tapered slightly to fit into the inside of the tube. In the case of fire-tube boilers, the plugs are held in place by a rod passing through the entire length of the tube and through holes in the plugs. The rod is threaded at the ends and suitable nuts screwed up to place the rod in tension and securely fasten the plugs. In the case of water-tube boilers, the boiler is

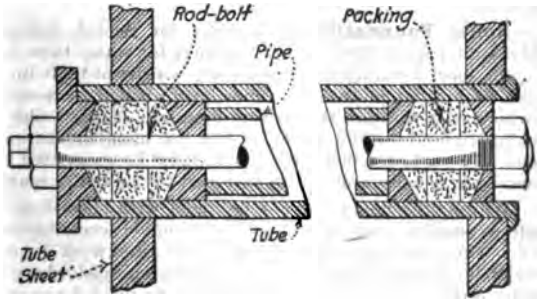


FIG. 62.—Tube Plug, Fire-tube Boiler.

opened and the plugs merely driven into the ends of the tubes and held in place by the steam pressure when the boiler is again put in service. The plugs in this case are drilled on the larger ends with holes passing only part way through the plug, and these holes are tapped for the reception of a bolt used for the subsequent extraction of the plugs, a plug extractor being furnished for this purpose.

To avoid the delay of hauling fires and cooling off fire-tube boilers to enable a man to enter the back connection, a tube-plug device may be used consisting of a rod passing through a pipe a few inches shorter than the tube, with washers so arranged as to produce a compressive action on special packing rings when tension is put on the rod by means of a nut on the front end. The entire device will enter the tube, and may be installed from the front so that the packing will be located near the tube ends and nearly fill same. The compression of the packing expands it and makes a joint at each end sufficiently tight for temporary repairs.

Tube plugs when designed to be driven into the ends of the tube should be at least 2 in. long and made to suit the gage and diam. of the tube. The taper should be slight, about ⅛ in. reduction in diam. per in. of length of plug for 4-in. tubes, and about half this amount for 1-in. tubes—other sizes in proportion. The smaller end of the plug should be $\frac{1}{32}$ in. to $\frac{1}{16}$ in. greater in diam. than the inside diam. of the tube before expanding. This will insure, when the plug is driven home, a bearing of the plug on the flexible end of the tube projecting through the tube-seat instead of on the seat itself, the end of the tube having been already enlarged slightly by the expander or flaring tool.

BOILER SETTING

Space Around Boilers. **The U. S. Supervising Inspectors** require that externally heated boilers shall have a clear space between the boiler and the woodwork of not less than 6 in. at the sides and 4 in. at the top. Internally heated boilers shall have a clear space between the boiler and the woodwork of not less than 4 in. at the sides and 4 in. at the top. All boilers in wooden steamers shall have a clear space of at least 8 in. between the under side of the cylindrical shell and the floor or keelson; and on all other steamers the boilers shall be so placed as to permit of proper inspection of the under side thereof. All water-tube boilers shall have a clear space of not less than 4 in. between the back end of boiler and the bulkhead. All other boilers shall have a clear space of not less than 2 ft. between the back end of boiler and the bulkhead.

Seatings—Boiler Foundations. Seatings for Scotch boilers are described by Mr. Bailey on p. 390. Boiler seatings for water-tube boilers may be altered to suit special conditions, but usually consist of built-up structural work, finishing with a flat plate at the floor level, drilled to receive holding-down bolts which pass through the bottom members of the boiler structure, the latter being riveted to the boiler casing, or members supporting the pressure parts. The ashpans or furnace floors must be supported by suitable structural members, usually angle-irons fastened to the boiler seatings.

Casings. Water-tube boilers should be fitted with sheet-iron casings as nearly air tight as possible, and the same should be made accessible for overhaul and properly backed by insulating material and brickwork to protect the casing. Special tile fitting the tubes are usually employed where the casing covers the tube surfaces. Ample means in the form of doors or manholes should be provided to give access to the furnaces and tubes for cleaning purposes.

Brickwork. The introduction of oil fuel has greatly emphasized the importance of the use of firebrick of very high refractory qualities. These are made necessary, not only on account of the higher furnace temperatures, but also, in many instances, boiler weights have to be reduced and the firebrick walls are consequently made as thin as possible. At least 9 in. of firebrick

should be used in commercial designs, although this may be reduced to 2 in. in extreme cases for Destroyer boilers. In any case, all brickwork in walls which are vertical, or nearly vertical, should be bolted to the casing, or otherwise secured. This does not refer to Scotch boilers, where the use of firebrick is practically limited to the bridge-wall. All joints should be made of fireclay of a refractory quality at least equal to the brick. The so-called high temperature cements are very effective for repairing furnace brickwork, but are not necessary in laying up new walls. The time-honored custom of mixing the fireclay with salt water produces a mixture which is lower in refractory qualities; consequently, the melting of this mixture brings about a cementing action which is rather to be desired in coal-burning furnaces which do not reach very high temperatures. Salt water, however, must never be used in mixing fireclay for use in oil-burning furnaces, as in this case the high temperature causes the cement to run out of the joints and flux with the firebrick, greatly lessening its life.

The practice of painting firebrick walls with a wash of high-temperature cement or fireclay is of no advantage for new work. If the walls have been in service, however, and have developed surface cracks, this wash constitutes an effective method of filling up these cracks, and undoubtedly adds to the life of the brick.

Non-conducting Material—Lagging. Exposed portions of the boiler shell or pressure parts should be covered with at least 2 in. of suitable non-conducting material protected on the outside by a substantial covering, preferably thin sheet-steel riveted in place. The parts of water-tube boilers requiring to be lagged are usually of small area, as this type of boiler is almost invariably largely protected by a sheet-iron casing lined with non-conductor. In the case of Scotch boilers, the entire outside surface of the shell requires covering, as does also the rear head. To allow inspection of the boiler the lagging should be made removable.

Table 27. Insulating Materials

Trade name	Manufacturers	Bulk sp. gr.	Wt. per cu. ft.	Remarks
Nonpareil	Armstrong Cork & Insulation Co.	0.36	22.45	The main insulating ingredient is diatomaceous earth as mined, bonded with asbestos fibre and carbonates or silicates, and then pressed into block or slab form.
Fire backing	National Magnesia Co.	0.41	25.57	
Celite blocks	Celite Products Co.	0.35	21.83	
Zenithern high Temp. block	The Zenithern Co.	0.35	21.83	
J-M high temp. block	J. W. Johns-Manville Co.	0.39	24.32	
Zenithern-U. S. Navy block	The Zenithern Co.	0.87	54.26	Calcined diatomaceous earth, bonded with asbestos fibre and silicates; then pressed into block form.
Insulbrix	Quigley Furnace Specialty Co., Inc.	1.14	71.10	Calcined diatomaceous earth bonded and made into compressed brick.
Sil-O-Cel brick	Celite Products Co	0.56	34.93	Diatomaceous earth as mined, cut into brick shapes.
Calcined Sil-O-Cel	Celite Products Co.	.....	.....	Calcined diatomaceous earth in powder or lump form. Calcined at temperatures between 1800° to 2500° F. to suit purpose. Wt. per cu. ft. depends on degree of packing down.

The Babcock & Wilcox Company recently conducted elaborate experiments which showed that 85 per cent magnesia covering, while entirely satisfactory for lagging steam drums, pipes, etc., is quite unsuitable as a material for insulating furnace walls, but that on the contrary, non-conducting materials in which diatomaceous earth is used as a basis, are satisfactory both as to lightness and non-heat-conducting qualities, as well as ability to withstand the necessary temperatures. Several forms of this material are now on the market, see Table 27

BOILER MOUNTINGS

Internal Fittings

Dry Pipes. The main purpose of the dry pipe is to collect the steam over an extended area near the top of the shell farthest removed from the surface of the water, thus preventing a sudden local pull, which would promote priming. These pipes should be of the same diameter as the steam outlet, and are usually perforated with a series of $\frac{5}{16}$-in. or $\frac{1}{4}$-in. holes on the upper half of the pipe, or furnished with a series of narrow slots. The aggregate area of the perforations, or the slots, should equal 1.25 or 1.50 times the area of the pipe. If a boiler is to be subjected to sudden, intermittent demands for steam, as would be the case, for example, in operating a very long-stroke, slow-speed engine, the aggregate area of the openings in the dry pipe may be reduced to 75 per cent or 80 per cent of the area of the dry pipe, thus creating a wire-drawing action and effectively preventing priming, although somewhat increasing the drop in steam pressure from the boiler drum to the throttle.

The Supervising Inspectors state that safety valves when fitted either to the shell of boiler or steam drum may be fitted with internal dry pipes when made of standard steam pipe or of riveted material equal in thickness and when the combined openings in the dry pipe equal in area at least 1.5 times the opening of the valve

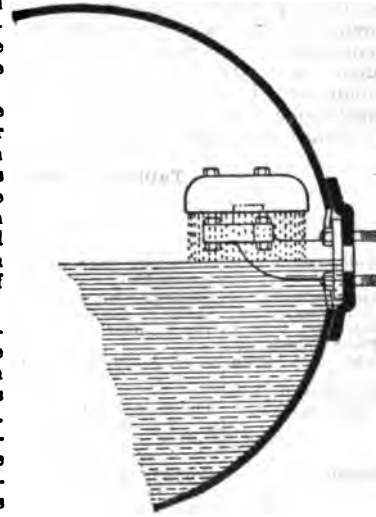


FIG. 63.—B. & W. Feed Nozzle.

Internal Feed Pipes. The customary method of introducing the feed water into the boiler is through a perforated pipe which distributes the cooler water over a wide area. This is an important feature in connection with the Scotch boiler where local cooling of any part of the system is to be avoided as much as possible. The perforations on the pipe should be on its upper side only, so that, as the water is ejected from the pipe, it will flow upward in small jets and facilitate the liberation of air. The practice, sometimes followed, of discharging the feed downward between the nests of tubes, while aiding circulation, tends to increase corrosion.

In the water-tube boiler, a wide distribution of the incoming feed is not

essential on account of the rapid circulation constantly going on, though perforated feed pipes are notwithstanding frequently employed with satisfactory results. The Babcock & Wilcox Company has introduced a form of internal feed connection which has the advantage of greatly facilitating the liberation of air from the feed water. This consists of an up-turned nozzle of reduced area, situated above the normal water level and forcing the water in the form of a jet upward against an umbrella-shaped cover, which deflects the water downward in a circular spray, or thin film, from which the air is easily liberated, see Fig. 63.

Internal pipes for bottom blows are not ordinarily used in water-tube boilers, the bottom blow valve being connected to a manifold in the lower portion of the boiler. In Scotch boilers, internal pipes are used to connect the outlet wherever located with the lowest point in the interior of the shell, so that by a siphoning action or by connection with the suction of a pump the boiler may be entirely emptied of water. These pipes are not perforated, but have open ends located at the desired points.

Scum Pans. The shallow pan installed just below the normal water level, and open on the side opposite to the surface blow connection, remains as the favorite type of fitting. The United States Navy, however, has recently replaced this form of pan with a perforated pipe, running along the side of the boiler-shell just below the normal water level, the theory being that the ebullition of the water causes the scum to find its way to the sides, instead of remaining near the centre.

Zinc Baskets. Zinc plates on corroding in the boiler, flake off in large pieces, which, if allowed to fall on furnace crowns or tops of combustion chambers or into the boiler tubes or headers, are likely to cause local overheating and serious danger to the boiler. The supports for the zincs, therefore, while of open construction, to give free access of water to the zincs, must be so arranged as to catch any falling pieces and hold them securely till the boiler is opened and the baskets cleaned. The bottoms of the baskets may be drilled with ¼-in. holes on about 1-in. centres. The zincs should be suspended in the boiler water near the incoming feed.

Swash Plates. In all boilers having a large water surface in a single compartment, swash plates should be installed to prevent excessive movement of the water due to rolling and pitching of the vessel. These plates should be of substantial construction, not less than No. 10 B.W.G., and easily detachable. They should extend from about the normal water level, well below the surface, at least 12 in. or 15 in. They should not divide the boiler drum into separate compartments, but should be open at the bottom to allow free passage of the water to other parts of the boiler. Swash plates are rarely fitted in Scotch boilers but it is questionable if their use would not be advantageous.

Internal Baffles. To prevent splashing of water onto the dry pipe, due to ebullition of steam or other causes, suitable baffle plates should be installed. They have been used effectively in Scotch boilers over the top of the combustion chambers, and are generally necessary in water-tube boilers. Where used, they should be easily detachable or hinged, to give free access to the boiler for overhaul, and should be of substantial construction to offset corrosion and possible displacement.

Circulators and Hydrokineters. The defective circulation in Scotch boilers may be greatly improved by internal arrangements of pipes designed to take the cold water from the lower part of the boiler and discharge it near

the water level. The **Eckliffe Circulator** operates automatically in the interior of the boiler, the motive power being derived from the heat imparted to the water in pipes laid in close contact with the top of the furnaces, with suction connections extending to the bottom of the shell.

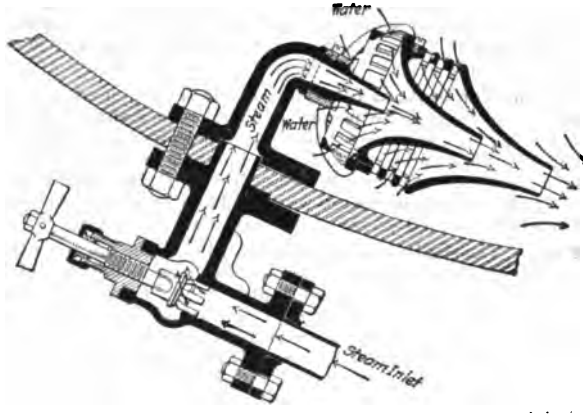


FIG. 64.—Hydrokineter.

The **Hydrokineter** shown in Fig. 64 operates by means of steam admitted to a fitting in the lower portion of the shell, so designed as to drive hot water into the cool space, and thus heat or displace the cold water. The same purpose is accomplished by pumps externally connected which circulate the water mechanically.

External Fittings

The **U. S. Supervising Inspectors' rules** require that a stop cock or valve shall be placed between all check valves and the boiler and between all steam and water pipes and the boiler. All valves shall bear the trade-mark of the manufacturer, which shall guarantee uniform thickness of the walls of the valve chamber. All such valves of 3 in. and over shall also be stamped with the number of pounds pressure of steam the manufacturer guarantees them to stand. The manufacturer of such valves and fittings shall file with the Supervising Inspector General a sworn certificate that all the fittings furnished by him for boilers of steam vessels shall comply with these requirements. Screwed bonnets on cast-iron valves are prohibited. All valves over 2 in. in diam. shall have bolted bonnets. Valves of 2 in. in diam. or less, connected directly to the boiler, shall be of cast steel, hard brass or bronze. All pipe fittings of more than 3 in. diam. shall be subjected by the manufacturer to a hydrostatic test of 3.5 times the working pressure.

Material. The Supervising Inspectors approve **cast-steel** fittings of any size or character and for any pressure and for any temperature for steam-pipe and feed-pipe connections and for valves, cocks and all appliances subject to pressure, when made by regular processes and by manufacturers who stamp such fittings with their identifying stamp and guarantee the castings to

possess the following physical characteristics: Tensile strength, minimum 50,000, maximum 70,000 lbs. per sq. in.; elastic limit, not less than 45 per cent of T.S.; elongation in 2 in., minimum 25 per cent; coupons from each heat shall be tested or furnished to the Local Inspectors for test when required; all steel castings shall be thoroughly annealed.

Cast iron, semi-steel and ferro-steel having a tensile strength of not less than 22,000 lbs. per sq. in., may be used in the construction of valves and fittings for pressures not exceeding 300 lbs. per sq. in. or a temperature of 450 deg. F. Such fittings if 3 in. diam. or over must be stamped with the identifying stamp of the manufacturer. **Malleable iron**, possessing a tensile strength of not less than 30,000 lbs. per sq. in. may be used in the construction of valves and fittings up to and including 6 in. in diam., and for pressures not exceeding 300 lbs. and a temperature not exceeding 450 deg. F. Such valves and fittings of 3 in. diam. or over shall be extra heavy, the fittings beaded or banded, and the valves and fittings stamped by the manufacturer.

Hard brass, bronze or other composition of which 95 per cent is copper, tin and zinc, having a tensile strength of not less than 30,000 lbs. per sq. in., may be used for valves and fittings up to and including 12 in. in diam., and for pressures not exceeding 300 lbs. and for temperatures not exceeding 470 deg. F.; but if the manufacturer stamps such fittings with the temperature they are guaranteed to stand without disintegration, they may be used for temperatures not exceeding 550 deg. F. This rule also applies to any composition meeting the above physical requirements, and stamped in accordance therewith. No materials shall be used for pressures exceeding 300 lbs. per sq. in. or temperatures exceeding 550 deg. F. other than steel or a composition whose tensile strength exceeds 50,000 lbs. per sq. in., with an elongation of 10 per cent in 2 in., and such fittings must be stamped as above.

Thickness of Fittings. The Supervising Inspectors' rule for thickness of fittings may be expressed by the following formula: $T = \frac{P \times D}{H} + C$, where T = thickness of wall of fittings in in.; P = working pressure in lbs. per sq. in.; and D = diam. in in.; H and C are constants depending upon the material used in the fitting.

For cast steel, $H = 7000$; $C = 0.188$[1].

For cast-iron as above, malleable iron, semi-steel and ferro-steel, $H = 3000$; $C = 0.250$.

Method of Fastening to Boiler. The Supervising Inspectors require that all boiler connections over 2 in. in diam., except connections for safety valves, shall be flanged and bolted directly to the boiler. When the point of connection on the boiler is of circular form, distance pieces shall be allowed for squaring the point of attachment to the flange, but no distance piece shall exceed 8 in. in length on its shortest side.

Cast-steel flanged fittings may be used for the purpose of connecting main and auxiliary steam stop valves, safety valves and other steam outlets, the distance from axis of outlets of such fittings to point of connection to boiler to be as short as practicable.

Provided the thickness of material of the boiler is not less than ½ in., or where the material of said boiler has been reinforced by a pad or flange riveted on same, to make the thickness not less than ½ in., pipes or fittings of 2 in. in diam. or under may be screwed directly into same. Pipe connections shall be extra heavy, and as short as possible.

Valves and fittings of 3 in. and under may be connected by screw-threads at intermediate joints in pipe lines, but at point of connection with boiler they shall be flanged if over 2 in. in diam., and properly secured by bolts, studs or rivets. No fitting shall be of greater length than specified by the "Manufacturers' Standard." Cast nozzles shall not be used when exposed to the direct action of the fire.

Feed Connections. The Supervising Inspectors require an auxiliary feed connection, in addition to the main feed, which shall enter the boiler through

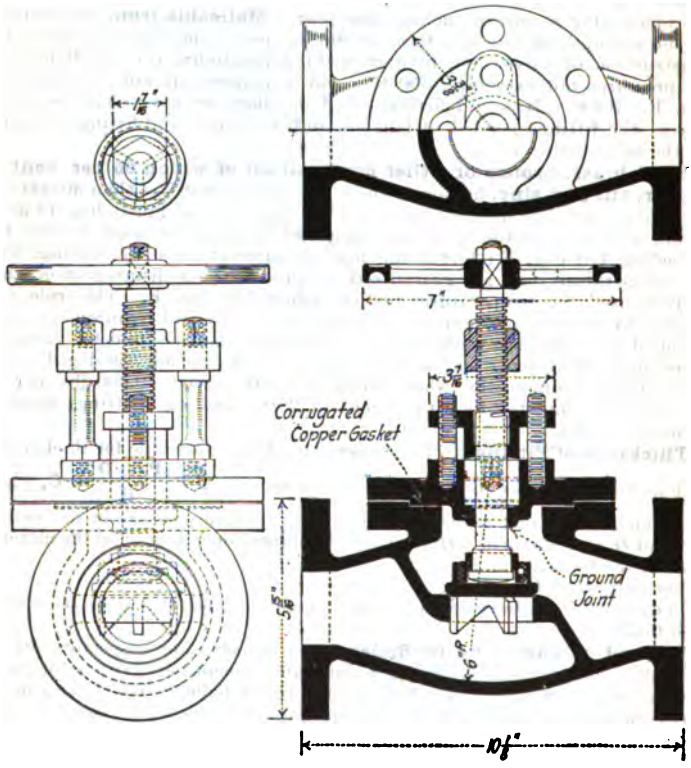


FIG. 65.—Feed Stop Valve.

an opening and fitting entirely independent of the main feed. As it is required that on both main and auxiliary feeds the feed stop valves shall be placed between the check and the boiler, and as the check should be as near the boiler as possible, it is usual to bolt the two valves together and bolt the flange of the stop valve to the boiler shell or pad riveted thereon. The Navy Department has a design in which the two valves are arranged in one body, thus eliminating one joint and reducing weight.

The feed supply may be controlled either by regulating the lift of the check or adjusting the stop, but the check is usually of such design that the control of same from the fireroom floor by means of an extension handle requires the use of gearing. The simpler method is, therefore, to set the stop valve with the stem downward and attach the extension handle to same. Feed valves should be of heavy substantial construction to withstand shock of water hammer. Typical designs are shown in Figs. 65 and 66. For sizes, see page 379.

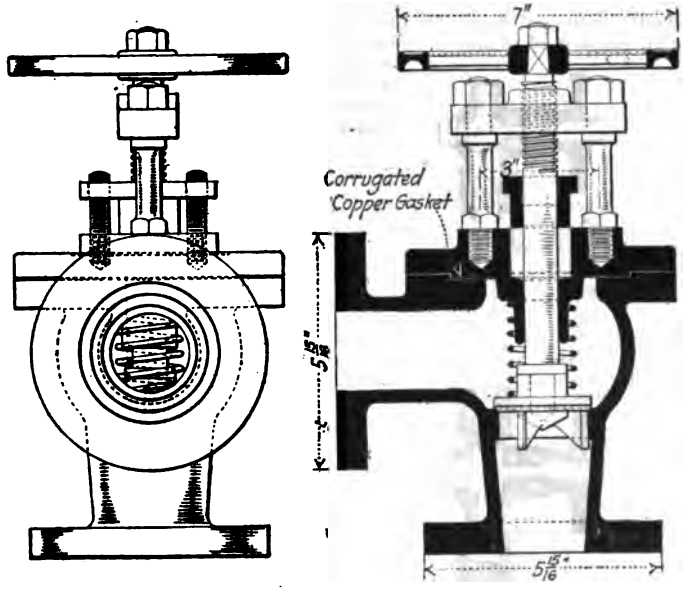


FIG. 66.—Feed Check Valve.

Gage Glasses. The Supervising Inspectors state that "A gage glass shall be considered a reliable water gage, and a float such as used on Western river steamers shall be considered on such boilers as a reliable gage." Needless to say, the float is not used in other service. All boilers, except flash boilers, shall be fitted with at least one reliable gage. In most of the best designs of water-tube boilers, two gage glasses are used. In double-end boilers, a gage glass at each end is required.

Gage-glass Fittings. Gage-glass fittings must be very accurately aligned and held firmly in position by a suitable backing plate to insure that no side strains shall be placed on the gage glass. Provision should be made either by use of gaskets with a square or hexagonal shape on the outside, or by special design of the glands for preventing twisting of the gaskets on the glass when tightening up. The connections to the boiler above and below the water line should be as straight and short as possible, and of such shape as to absolutely prevent pocketing which would show water in the glass when the

water in the boiler is at a lower level. Valves or cocks on the connections to the boiler are always a source of danger. As they are practically indispensable, however, they should always be made of a type which will show at a glance whether they are open or closed.

The Dewrance gage-glass fitting, Fig. 67, is furnished with a small composition ball supported in a bracket immediately under a suitable seat at the lower or water end of the glass. In case of breakage of the glass, the outrush of water lifts the ball from the bracket up against the seat, automatically closing off the exit of the hot water. The upper or steam end of the glass has no automatic closing-off device for the reason that it is not desirable to entirely close off the flow of steam, as in the case of a cracked or broken glass there will be nothing to call the water-tender's attention to the fact. The flow of steam is, however, restricted by means of a small basket-shaped fitting inserted after the

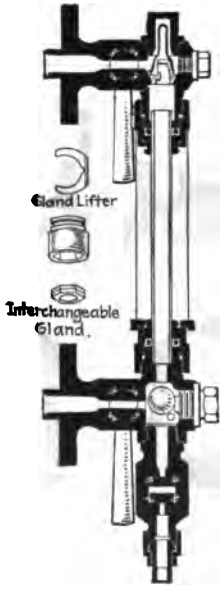


FIG. 67.—Dewrance Gage Glass Fitting.

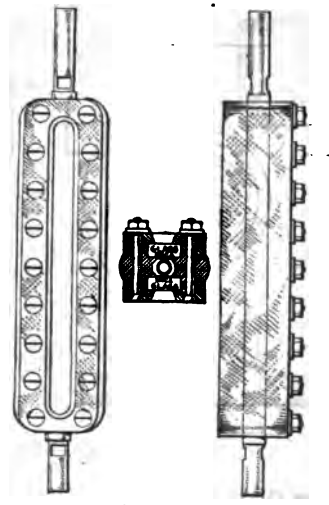


FIG. 68.—Dewrance Water Level Indicator.

glass is in position. This prevents any considerable discharge of steam, but yet is sufficient to call attention to the broken glass. In all cases where plain cylindrical glasses are used, a protector composed of stiff wires or wire gauze should be furnished to prevent damage from the flying glass in case of breakage of same. The use of these protectors should be insisted upon, notwithstanding that they obstruct to some extent a clear view of the water level. This condition should be offset by the careful arrangement of electric lights placed near the gage glass.

Heavy corrugated glass fitted in metallic frames and connected top and bottom to brass tubes for insertion in the gage-glass fittings, have been employed and are much safer against breakage than the plain glass. The reflection of light from the corrugations when the glass is clean, clearly marks the

water level. Their effectiveness is greatly decreased by scum or dirt on the inside, and they considerably reduce the range of visible level. Also, the height of water is visible only from one side. A better arrangement consists of two pieces of heavy plate glass in metal frames. The water is admitted between the plates and mica sheets are introduced to protect the glass from corrosion. The water level is visible from two sides, Fig. 68.

Whatever type of gage glass is employed, an effective means of blowing out the glass and keeping the passage clear must be provided.

Gage Cocks—"Try Cocks." The Supervising Inspectors require that all boilers except flash boilers shall be fitted with at least three gage cocks. Except in water-tube boilers, these shall be attached directly to the boiler. While not so stated, it is understood that these cocks shall be placed at different heights, to cover the usual range of water level.

In fire-tube boilers of more than 48 in. diam., the lower cock shall be placed not less than 4 in.—and in boilers of less than 48 in. diam. not less than 2½ in.—above the top of flues, tubes, or combustion chamber.

In water-tube boilers which have an efficient water column (a fitting containing a body of water—usually a stand-pipe of cylindrical form set vertically) connected at the top to the steam drum and at the bottom to a water manifold of the boiler, and fitted with valves or stopcocks at the points of connection to the boiler, the gage cocks may be placed in the water column instead of being attached directly to the boilers. Water columns fitted with gage cocks may be used with fire-tube boilers, but in this case an additional set of three gage cocks shall be fitted directly to the boilers. It is generally recognized that the proper place for the gage cocks is directly attached to the boiler; and this is always done in the case of water-tube boilers if the design of the boiler permits.

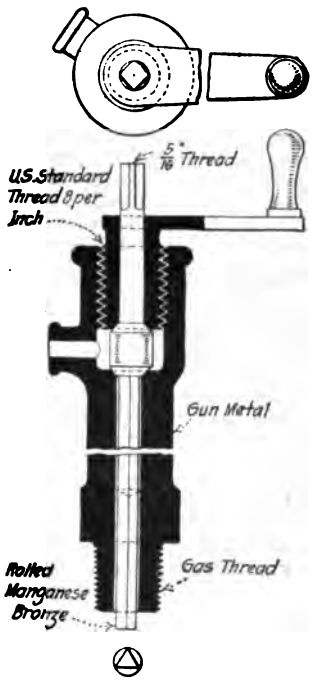


FIG. 69.—Try Cock.

Steam Gage. The Supervising Inspectors' rule calls for one steam gage for each boiler, or each set of boilers, which will correctly indicate a pressure equal to 80 per cent of the hydrostatic pressure.

Safety Valves. The Supervising Inspectors' rule is as follows:

"Safety valves when fitted either to the shell of boiler or steam drum may be fitted with internal dry pipes when made of standard steam pipe or of riveted material equal in thickness, and when the combined openings in the dry pipe equal in area at least 1.5 times the opening of the valves.

"Any spring-loaded safety valve constructed so as to give an increased lift by the operation of steam after being raised from its seat, or any spring-loaded safety valve constructed in any other manner, so as to give an effective

area equal to that of the afore-mentioned spring-loaded safety valve, may be used in lieu of the common lever-weighted valve on all boilers on steam vessels, and each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to $\frac{1}{8}$ the diam. of the valve opening; but in no case shall any spring-loaded safety valve be used in lieu of the lever-weighted safety valve without first having been approved by the Board of Supervising Inspectors.

"The valves shall be so arranged that each boiler shall have at least one separate safety valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety valve or valves employed. This arrangement shall also apply to lockup safety valves when they are employed.

"The use of two safety valves may be allowed on any boiler, provided the combined area of such valves is equal to that required by rule for one such valve. Whenever the area of a safety valve, as found by the rule of this section, will be greater than that corresponding to 4¼ in. in diam., two or more safety valves, the combined area of which shall be equal at least to the area required, shall be used.

"Where escape pipes for safety valves are installed in steam vessels, the area of such pipes shall equal the combined area of all valves to which such pipes are connected.

"Where safety valves are used with beveled seats, the seats shall have an angle of inclination of 45 deg. to the centre lines of their axes. Flat-seat safety valves may be used. The discharge capacity of a flat-seat valve shall be 1.4 times that allowed for a bevel-seat valve.

"No safety valves having a set-screw arrangement on top of the valve casing, designed to hold the valve down while the hydrostatic pressure is being applied, shall be allowed. On such valves now in use, inspectors shall require the set screws to be taken out and the hole permanently closed. This does not apply to any safety valve whose form of construction is such that the hole for the set screw or bolt is securely closed when the valve is locked."

Lever safety valves are approved by the Supervising Inspectors if complying with certain requirements. As these valves are seldom used in marine work except on inland rivers the rules are not quoted herewith.

Size of Safety Valves as required by the Supervising Inspector's rulse may be expressed by the formula $A = 0.2074 \times \frac{W}{P}$ in which, A = area of valve or combined area of valves on boiler, in sq. in.; W = maximum amount of water evaporated by the boiler per hr. in lbs.; P = working steam press. in lbs. per sq. in., gage.

It is not necessary to base safety-valve area on grate surface. For convenient reference in finding the proper diameter of valve, the following table is given:

Diam. in inches	2	2½	3	3½	4	4½
Area in square inches	3.141	4.908	7.068	9.621	12.566	15.904

While the U. S. Supervising Inspectors' rules permit the safety valve to be fitted to the same casting as the main stop valve it is considered better practice to fit the safety valve to a separate outlet either with or without a connection to the dry pipe, but, if the latter, the dry pipe should also be connected to the main steam outlet to insure its being kept clear and open.

Blow-off Valves. Substantial globe or angle valves of the ordinary disk and seat type may be employed for surface blow connections where they are usually subjected to only clear water or scum without gritty deposits. This type of valve is, however, no longer considered satisfactory for bottom blow connections, and the seatless, or plunger type of valve, is rapidly becoming the favorite in first-class installations. The Nolan valve and Bureau of Steam Engineering design (the latter shown in Fig. 70), are both excellent examples of this type of valve. It will be seen from the illustration that when the valve is opened, the water, and whatever small amount of muddy deposit that it may contain, is prevented from coming in contact with the cylinder walls, and the plunger is made tight by compression of circular gaskets or packing. The latter is entirely protected from the flow of water.

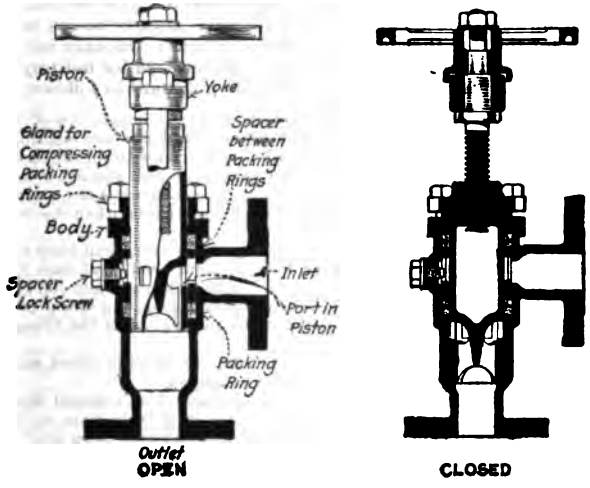


FIG. 70.—Bottom Blow Valve, U. S. Navy Type.

Drum Pads may be made of forged steel or cast steel, and are ordinarily used for the reception of a flange in making a connection to a cylindrical shell. These pads are usually fitted with stud-bolts, and the ordinary construction is such that the gasket will fall entirely within the bolt circle. If the facing of the nozzle for the reception of the gasket is done before attachment to the boiler shell, reasonable care should be exercised so that the face of the nozzle in contact with the shell will fit the latter so that when the riveting is done there will be no distortion of the machined surface. Also, the flanges should be made sufficiently thick to counteract any distortion due to the pull of the bolts.

Fusible Plugs. The use of fusible plugs for indicating dangerously low water levels is given special consideration by the Supervising Inspectors. If not properly constructed and suitably located in the boiler, such plugs may fail to operate to give the desired warning, or they may fuse and blow when the water level is safe, thus shutting down the boiler unnecessarily.

Fusible plugs are usually screwed in from the external or fire side of the boiler, but if it is considered easier to replace a plug from the water side, it may be installed from the interior of the boiler, provided the plug is so made as to comply with the requirements.

The Supervising Inspectors' rules are interpreted to mean that fusible plugs are not required in water-tube boilers, but they are, notwithstanding, frequently installed as a precaution against low water.

Pure tin, the metal used in the core of fusible plugs, has a melting point of 450 deg. F. It may be safely used with steam pressures of 250 lbs. gage, at which pressure the water boils at 406 deg. F. At 400 lbs. gage, the boiling point is so nearly equal to the melting point of tin that this metal could not be used, and would undoubtedly fail at lower pressures. If, therefore, water-tube boilers are installed, with working pressures much above 250 lbs., fusible plugs if used will have to be filled with alloys selected with melting points to suit the steam pressure. This point would have already arisen in practice were it not for the fact that in Naval service, which is the only one so far to use the higher steam pressures, fusible plugs are omitted for military reasons.

The Supervising Inspectors' rules for fusible plugs are as follows:

"Fusible plugs for use in boilers of steam vessels under the jurisdiction of the Steamboat Inspection Service shall be made of a bronze casing with the bore tapering continuously and evenly from end to end, and filled from end to end with tin not less than 99.7 per cent pure and to contain not more than 0.1 per cent of lead and not more than 0.1 per cent of zinc. The small end of the bore may be countersunk not more than 1/32 of an in. in depth and width, but no recess, thread, or cavity other than this countersink shall be allowed.

"Fusible plugs, except those which are hereafter provided for, shall have an external diam. of not less than 3/4 of an in. pipe tap, and the filling shall be at least 1/2 of an in. in diam. at the smaller end, and shall have a larger diam. at the opposite end of the plug: *Provided, however*, that all fusible plugs fitted in boilers carrying a steam pressure exceeding 150 lbs. to the sq. in. may be reduced at the smaller end of the filling to 5/16 of an in. in diam.

"Every boiler other than boilers of the water-tube type shall be fitted with at least two fusible plugs as described above, and located as follows:

"Upright boilers shall be fitted with two fusible plugs of an external diam. of not less than 3/4 of an in. pipe tap, the filling to be at least 1/2 of an in. in diam. at the smaller end and shall have a greater diam. at the opposite end. The fusible plugs shall be located in separate tubes not more than 2 in. below the lowest gage cock.

"Externally heated cylindrical boilers, with flues, shall have one plug in the top of the upper flue, not more than 4 ft. from the back end of the flue, and shall also have a plug fitted to the shell of the boiler immediately below the fire line, and not less than 4 ft. from the front end: *Provided, however*, that when the flues are not more than 6 in. in diam. fusible plugs of not less diam. than 3/4 of an in. pipe tap may be used in such flues.

"Fire-box, Scotch, and other types of shell boilers not specially provided for, having a combustion chamber common to all furnaces, shall have two plugs fitted to the crown sheet of the combustion chamber at or near the centre of the crown sheet and not more than 12 in. apart.

"Double-end boilers with separate combustion chambers, each of which combustion chambers is common to all furnaces in its own end of the boiler, shall have two plugs fitted to the crown sheet of each combustion chamber at or near the centre of the crown sheet, and not more than 12 in. apart.

"Boilers fitted with a separate combustion chamber for each individual furnace shall be fitted with a fusible plug in the centre of the crown sheet of each combustion chamber.

"Boilers of types not herein provided for shall be fitted with at least two fusible plugs of such dimensions and located in such parts of the boiler as will, in the judgment of the local inspectors, best meet the purposes for which they are intended.

"Fusible plugs shall be renewed at each annual inspection, but in cases where the plugs were installed or renewed not more than 6 months prior to the annual inspection,

they may be allowed until the next following annual inspection or for a period not to exceed 18 months. In such cases, however, the inspector shall satisfy himself at the annual inspection that the plugs are in such condition as to warrant their continued use until the next annual inspection, after which the plugs shall be renewed at each and every annual inspection.

"Fusible plugs shall be so fitted that the smaller end of the filling is exposed to the fire, and shall be at least 1 in. higher on the water side than the plate or flue in which they are fitted.

"Notwithstanding anything which may be contained in this rule, fusible plugs shall be so fitted that the end of the filling on the water end of the plug is not less than 1 in. above the dangerous low-water level.

"Each manufacturer of fusible plugs shall number all plugs in accordance with the number of the heat from which the plugs were filled. For instance, the first pouring shall be number 1, and all plugs filled from this heat shall be numbered 1; the next pouring shall be number 2, and all the plugs filled from this heat shall be numbered 2, etc.

"The heat number shall be plainly stamped on each end of filling with numbers not less than $\frac{1}{2}$ in. in height.

"Manufacturers of fusible plugs shall furnish the Supervising Inspector General a sample plug from each heat for examination and test, but where more than 250 plugs are poured from the same heat, a sample plug shall be furnished for each 250 plugs or fraction thereof. The samples furnished shall bear the same number for any one heat and shall represent the heat from which the sample is poured.

"In transmitting samples of fusible plugs to the Supervising Inspector General, the fusible plugs and the letters of transmittal shall be addressed as follows: 'Supervising Inspector General, Steamboat Inspection Service, Department of Commerce, Washington, D. C.'

"One letter of transmittal shall be required for each heat.

"In letter of transmittal the following information shall be stated:

"Number of heat.

"Number of fusible plugs manufactured from the heat.

"Number of samples of fusible plugs from the heat transmitted.

"Name of manufacturer or initials stamped on casing of plug.

"Manufacturers of fusible plugs shall stamp on the larger end or face of the casing their name or initials for identification and shall file with the Supervising Inspector General of the Steamboat Inspection Service at Washington, D. C., a certificate duly sworn to that such plugs are filled with tin of the character required by this rule and made in accordance therewith."

Fire Tools. A set of fire tools should consist of the following: A shovel, a slice-bar, a short hoe or "devil's claw" for breaking up and pushing back the fuel, a long-handled hoe for cleaning, and a lazy-bar for laying across the fire-door opening as a rest for the handle of the hoe when cleaning fires. A pricker-bar is sometimes used for inserting between the grate bars, underneath, to clean out ash and break up any clinker which may have formed directly on the grates. A small sledge-hammer or button-set should also be furnished for breaking up the larger lumps of the coal.

Considerable diversity of opinion exists in regard to the design of these tools and weight of material used, and as on account of the hard usage, these tools are often being repaired, the design is frequently changed to meet the views of various individuals concerned. The accompanying illustrations show types of fire tools which will be a safe guide for the designer. (See Fig. 71.)

A lazy-bar attached to each individual firedoor is used by The Babcock & Wilcox Company, from the design of their Marine Superintendent of Erection, *Mr. John Coleman*. This has the advantage of always being ready for installation in place when needed. (See Fig. 72.)

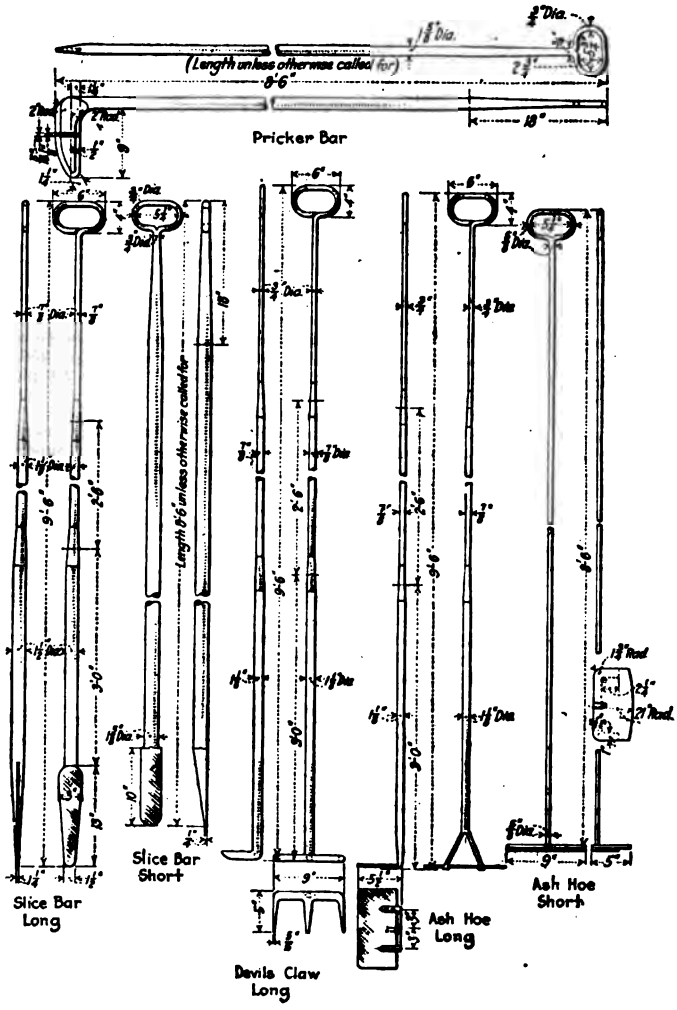


FIG. 71.—Fire Tools.

FURNACE MOUNTINGS

Front Plates. Furnace fronts are usually constructed of steel plates provided with suitable openings for the fire and ash doors. In Scotch boilers and in many water-tube boilers, particularly when arranged for coal fuel, the fronts are backed by air-cooled iron castings, made in sections for easy renewal when burnt out. These castings not only protect the front plates, but form door jambs and arches or lintels surrounding the door openings and giving the necessary flare to facilitate the firing.

When oil fuel is used, the fronts are frequently protected by firebrick tile properly formed and bolted to the fronts with suitable non-conducting material installed between the brick and fronts to keep the latter cool.

Fire and Ash Doors. The counterbalanced, in-swinging, sheet-iron door, properly stiffened with bar-iron, and provided with a cast-iron liner protected from overheating by air ventilation, is undoubtedly the best fire-door yet devised. It is easily opened and closed with the shovel, and, being hinged at the top, it is much more durable than the side-swinging door. No latch is required to make this door stand in the open or closed position, as the counterbalance weights perform this office. It has the further advantage that it can close automatically in case of a rupture of any of the pressure parts, which would liberate steam to the furnace, and this door also provides ideal conditions for slicing the fire. (See Fig. 72.) The old-fashioned side-swinging

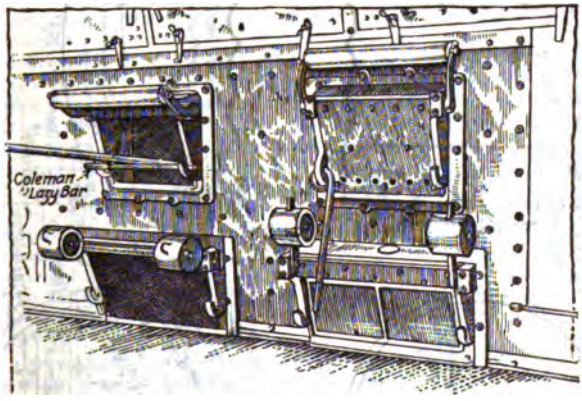


FIG. 72.—Counterbalanced Fire and Ash Doors.

door is still largely used in Scotch boilers, possibly because it is easier to make tight against outward leakage of gas. This is merely a question of design and does not in any way detract from the superiority of the counterbalanced door.

Ash doors when made on the counterbalanced principle are a further safeguard against accident. In this case, however, the door usually consists of a light casting. Care should be taken to so arrange the counterbalance weights that they will not interfere with the fireman when handling the fires.

Grate Bar Bearers may consist of forged bars or suitable castings of sufficient strength to support the entire weight of the grates and the fuel and

clinker. In Scotch boilers the bearers are usually supported by cast-iron brackets formed to fit the corrugations of the furnaces, resting on same and

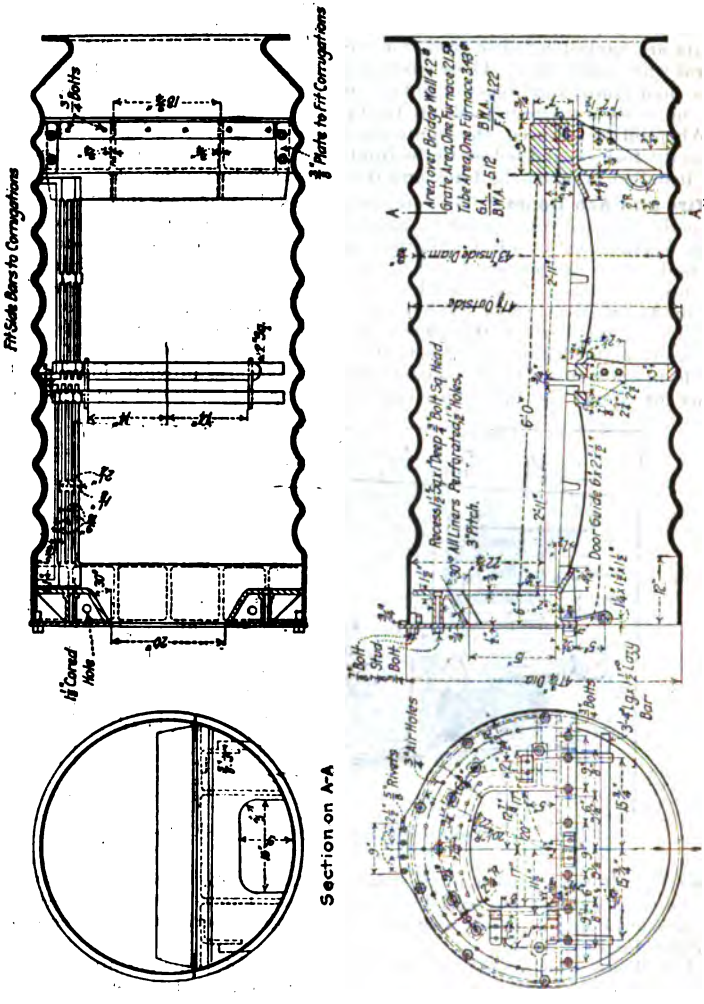


FIG. 73.—Grate Bar Bearers, Scotch Boilers.

held in place by gravity. The dead-plates and bridge-wall supports are arranged to carry the ends of the grate bars. (See Fig. 73.)

In water-tube boilers with wide furnaces, the bearers may be made in two or more lengths and supported on brackets bolted to the sides and on feather

plates dividing the ashpit longitudinally. Another construction consists of bearers in single lengths supported in wall-sockets in the side walls and brackets or columns in the centre. (See Fig. 74.)

Dead-plates and Bridge-wall Supports are shown in Fig. 73. Fig. 74 shows another design of bridge-wall support in greater detail, in which the dimensions for different furnace diameters are as given in Table 28. The door under the bridge wall is provided for the purpose of cleaning out the back connections of the boiler.

Table 28. Bridge Wall Supports, Fig. 74

A in inches.....	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
B in inches.....	17¾	18¼	18¾	19¼	19¾	20¼	20¼	21¼	21¾	22¼	22¾	23¼	23¾	24¼	24¾
C in inches.....	32½	33½	34½	35½	36¾	38	39	40	41	42¼	43¼	44¼	45¼	46¼	47½

Grate Bars. The maximum length at which grate bars can be effectively operated depends upon conditions. Where anthracite coal is used and the fires do not require "working" with the fire tools, 8 ft., 10 ft., and, under exceptional conditions, even 12 ft. have been employed in combination with dumping grates at the rear to facilitate cleaning. With bituminous coal which must be sliced and raked with the fire tools, 8 ft. is the maximum limit, and 7 ft. is the limit in sea-going vessels. Grates of this length, however, can only be used in connection with water-tube boilers of the type in which all the fire-doors are on the same convenient level (28 in. to 30 in.) above the fireroom floor. For boilers of this type, 7 ft. can be employed with distinct advantage, although shorter bars are as a rule preferable if the necessary grate surface can be secured, particularly if forced draft is used. Grate bars for Scotch boilers where, as in the case of a three- or four-furnace boiler, some of the furnaces are located very high above the fireroom floor and others very low, sometimes in pockets in the floor, 6 ft. is the maximum length of bars which can be used to advantage, or possibly 6 ft. 6 in. with large diameter furnaces and natural draft. Some authorities have even gone so far as to place the limit at 4 ft. for forced draft. In further explanation of this difference between water-tube and Scotch-boiler practice, it is to be noted that the cylindrical furnaces of the Scotch boilers with their corrugated surfaces are a serious handicap. The corrugations at the sides of the grate retard the hoe and the limited headroom in the furnace makes it difficult to coal the fires at the back.

Grates above 5 ft. 6 in. in length should be made in two sections, a clearance of not less than ¼ in. being allowed at the front and rear and where the bars adjoin at the centre, or other means provided for expansion. An effective method of preventing the bars from slipping out of position is to provide lugs at one end which fit over the bearing-bar thus fixing the longitudinal position. The bars may be made single, double or quadruple; that is, two (double) or four (quadruple) bars may be cast together in a single piece, stiffened at the sides to give the necessary strength. The quadruple bar is preferable on account of lightness, but is less flexible in filling out the width of the grate, and, when broken, requires replacement of a larger piece. Although slightly heavier, the cast-iron bar is to be preferred to forged bars of iron or steel on account of its cheapness and greater ability to withstand the action of clinker and slag.

The width of air space in grate bars depends entirely upon the character of fuel. The average figures are represented in Table 26.

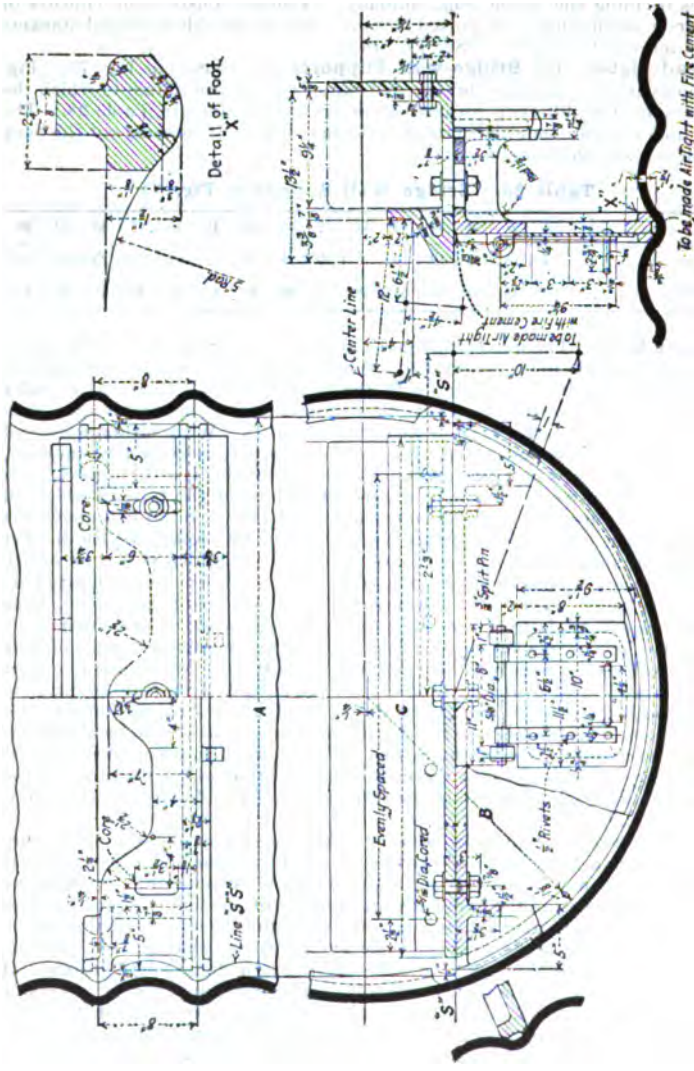


FIG. 74.—Bridge Wall Supports, See Table 28.

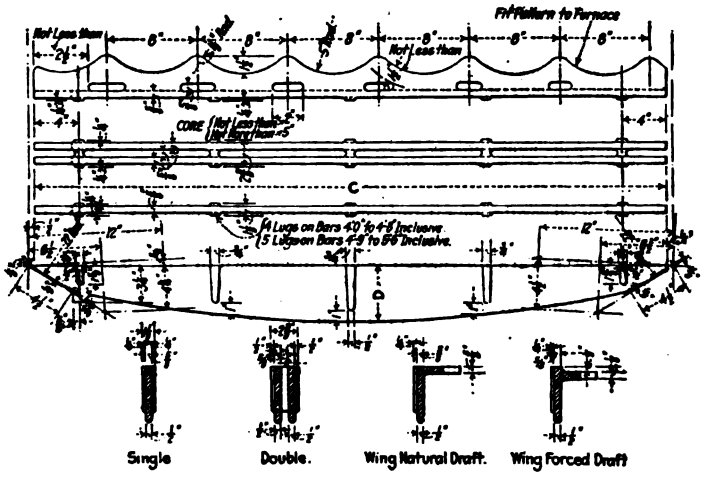


FIG. 75.—Grate Bars, Scotch Boilers, See Table 30.

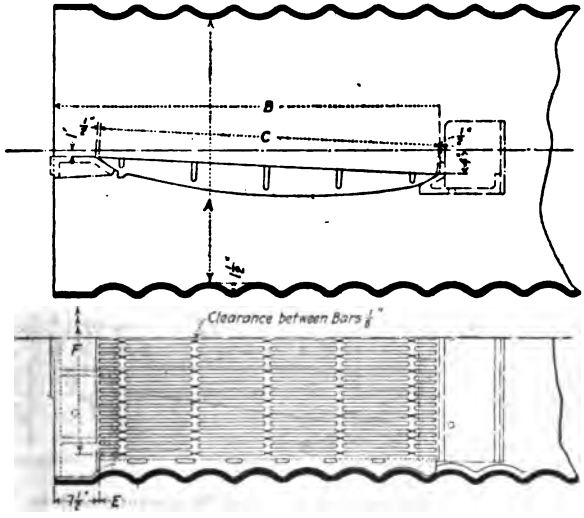


FIG. 76.—Grate Bars Assembled, See Table 30.

Table 26. Width of Air Space in Grate Bars

Kind of fuel	Width of air space
Anthracite, egg or lump	5/16 in. to 3/4 in.
Anthracite pea	1/4 in. to 5/16 in.
Anthracite bird's-eye	5/16 in. to 5/16 in.
Anthracite culm, pin holes, or perforations	1/4 in. diam.
Bituminous hand-picked lump	1/2 in. to 5/8 in.
Bituminous mine-run	3/8 in. to 1/2 in.
Bituminous slack or screenings	1/4 in. to 5/8 in.

It is not unusual to find grates in Scotch boilers with air spaces 5/8 in. and even 3/4 in. for all grades of bituminous coal except screenings. This results in the excessive loss of good fuel in the refuse.

Typical Grate Bars for Scotch boilers are shown in Fig. 75. The "**wing bars**" are used at the sides to fill in the spaces formed by the corrugations of the Scotch boiler furnace.

Fig. 76 shows the bars assembled. The proportions and number of bars for different diameters of furnace and different lengths of grate for Scotch boilers given by the Bethlehem Shipbuilding Corporation are shown in Table 27.

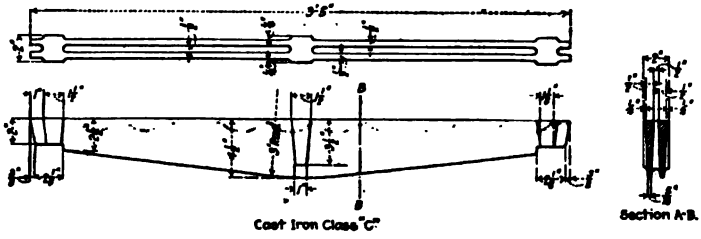


FIG. 77.—Double Grate Bars, Water-tube Boilers.

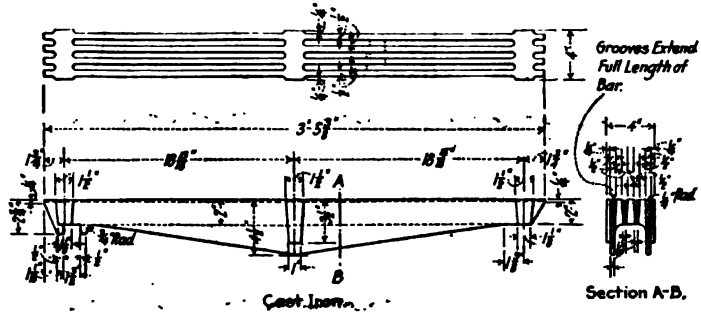


FIG. 78.—Quadruple Grate Bars, Water-tube boilers.

Typical **double and quadruple grate** bars for water-tube boilers as used by The Babcock & Wilcox Company for 7-ft. grates are shown in Figs. 77 and 78 respectively.

Table 30. Grate Bars and Supports

(See Figs. 75 and 76)

		A	36-in.			37-in.			38-in.			39-in.			40-in.			41-in.			42-in.			43-in.		
			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates		
B	C	D	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F
4' 0"	3' 11"	4¼"	3	11	31¼"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"	3	14	38¾"
4' 1"	4' 0"	4¼"	3	11	31¼"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"	3	14	38¾"
4' 2"	4' 1"	4¼"	3	11	31¼"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"	3	14	38¾"
4' 3"	4' 2"	4½"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"	2	14	38¾"
4' 4"	4' 3"	4½"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"	2	14	37½"
4' 5"	4' 4"	4½"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37¼"	2	14	37½"
4' 6"	4' 5"	4¾"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	3	13	36¼"	2	14	37½"
4' 7"	4' 6"	4¼"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	3	13	36¼"	2	14	37½"
4' 8"	4' 7"	4¾"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	3	13	36¼"	2	14	37½"
4' 9"	4' 8"	5"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	2	13	35"	3	13	36¼"	2	14	37½"
4' 10"	4' 9"	5"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	2	13	35"	3	13	36¼"	2	14	37½"
4' 11"	4' 10"	5"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	2	13	35"	3	13	36¼"	2	14	37½"
5' 0"	4' 11"	5¼"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 1"	5' 0"	5¼"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 2"	5' 1"	5¼"	2	11	30"	3	11	31¼"	2	12	32½"	3	12	33¾"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 3"	5' 2"	5¼"	2	11	30"	3	11	31¼"	2	12	32½"	2	12	33½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 4"	5' 3"	5¼"	2	11	30"	3	11	31¼"	2	12	32½"	2	12	33½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 5"	5' 4"	5¼"	2	11	30"	3	11	31¼"	2	12	32¼"	2	12	33½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"
5' 6"	5' 5"	5¾"	2	11	30"	3	11	31¼"	3	11	31¼"	2	12	32½"	3	12	33¾"	2	13	35"	3	13	36¼"	2	14	37½"

Table 30.—*(continued)*

		A	44-in.			45-in.			46-in.			47-in.			48-in.			49-in.			50-in.		
			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates			No. of grates		
B	C	D	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F	Single	Double	F
4′ 0″	3′ 11″	4¼″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	3	17	46¼″
4′ 1″	4′ 0″	4¼″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	3	17	46¼″
4′ 2″	4′ 1″	4¼″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	3	17	46¼″
4′ 3″	4′ 2″	4½″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	2	17	45″
4′ 4″	4′ 3″	4½″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	2	17	45″
4′ 5″	4′ 4″	4½″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″	2	17	45″
4′ 6″	4′ 5″	4¾″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	3	16	43¾″	2	17	45″
4′ 7″	4′ 6″	4¾″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	3	16	43¾″	2	17	45″
4′ 8″	4′ 7″	4¾″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	3	16	43¾″	2	17	45″
4′ 9″	4′ 8″	5″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	2	16	42½″	3	16	43¾″	2	17	45″
4′ 10″	4′ 9″	5″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	2	16	42½″	3	16	43¾″	2	17	45″
4′ 11″	4′ 10″	5″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 0″	4′ 11″	5¼″	3	14	38¾″	2	15	40″	3	15	41¼″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 1″	5′ 0″	5¼″	3	14	38¾″	2	15	40″	3	15	41¼″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 2″	5′ 1″	5¼″	3	14	38¾″	2	15	40″	3	15	41¼″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 3″	5′ 2″	5½″	3	14	38¾″	2	15	40″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 4″	5′ 3″	5½″	3	14	38¾″	2	15	40″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 5″	5′ 4″	5½″	3	14	38¾″	2	15	40″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″
5′ 6″	5′ 5″	5¾″	3	14	38¾″	3	14	38¾″	2	15	40″	3	15	41¼″	2	16	42½″	3	16	43¾″	2	17	45″

Shaking and Dumping Grates. Shaking grates are, as a rule, objectionable at sea on account of added weight, complication and increased cost of maintenance. Also, if carelessly used, they are wasteful of fuel shaken into the ashpan with the clinkers. Their use is practically limited to harbor vessels using anthracite coal.

Dumping grates on the contrary, while infrequently used at sea, greatly facilitate cleaning of fires and may be used to advantage with water-tube boilers where there is plenty of headroom in the furnace. They are impracticable with Scotch boilers on account of interference with the rounded, arching roof of the corrugated furnace.

SUPERHEATERS

The early types of superheaters were located in the boiler uptakes and in character of construction were similar to the boilers with which they were installed. The reduced temperature of the gases surrounding the superheaters as thus located made it necessary to use a large amount of surface, and the inaccessible position and difficulty of cleaning led to deterioration, and excessive upkeep costs.

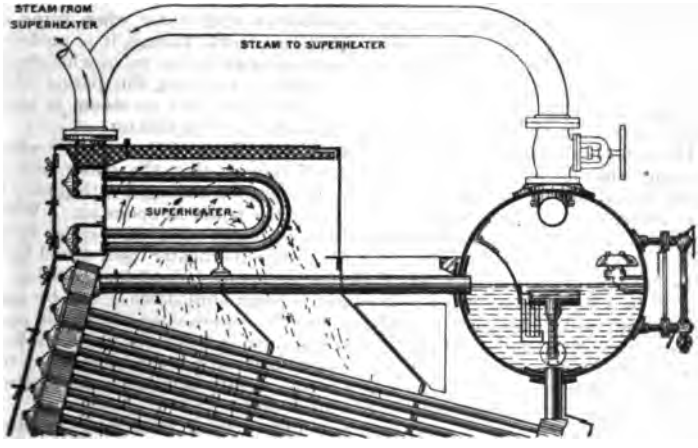


FIG. 79.—Babcock and Wilcox Superheater.

While the saving in fuel resulting from the use of superheated steam was well recognised, difficulties with the superheaters, and the inability to procure lubricating oils which would stand the high temperatures, caused the use of the superheater to be abandoned for a period of some thirty years.

Babcock and Wilcox Superheaters. Superheaters did not appear in marine practice until re-introduced in the early years of the present century,

when *The Babcock & Wilcox Company* brought out their well-known design. This is shown in Fig. 79, from which it will be noted that the saturated steam, after leaving the boiler drum, enters the forged steel box or manifold, thence passing through a series of 2-inch tubes of U shape, where it is superheated, and thence into another manifold, from which it passes to the propelling machinery. If desired, diaphragms or partitions may be introduced in the manifolds in such position as to cause the steam to pass from the first box through a portion of the tubes into the second box, and thence through another portion of the tubes into the first box, and so on, any desired number of times, thus ensuring any required velocity of steam through the superheater tubes.

The **superheater is so located** in the boiler setting that the gases of combustion reach the superheater tubes after passing only part way through the boiler setting. The gases thus retain a comparatively high temperature (1000 deg. F. to 1200 deg. F.), and a much smaller amount of superheating surface is needed to give the desired amount of superheat than if the superheater is placed in the uptake and exposed only to the gases at the uptake temperature. The reduced weight, and the superiority of its position, both as to effect on centre of gravity and accessibility for cleaning and overhaul, have resulted in this becoming recognized as the best standard practice.

The superheater manifolds or boxes may be placed across the back of the boiler (as shown in the illustration) or on the side of the setting, with the superheater tubes running at right angles to the boiler tubes, as desired to suit special conditions.

Pipe connections for flooding the superheater with water when firing up a cold boiler are not found necessary in marine work, though it is probable that a certain amount of water of condensation exists in the tubes when starting up which is effective in preventing overheating during this period while hot gases are passing around the superheater tubes but no steam is being taken away from the boiler and passed through the superheater.

The superheater may be considered an inherent part of the boiler, in which case any trouble with the superheater will require that the boiler unit to which it is attached shall be shut down, or it may be arranged with piping so as to be by-passed in case of trouble, leaving the boiler in operation. While entirely possible that the by-pass arrangement might result in a boiler being kept in service during some emergency when the superheater itself would have to be cut out, it should be noted that it is very rare indeed for anything to happen to a superheater which will cause its being shut down. If a by-pass is provided, and it becomes possible to cut the superheater out of the path of the steam flow and thus out of communication with the boiler safety valve, a relief valve should be provided, set to blow off a few pounds above the pressure at which the main safety valve blows, merely as a precaution against excessive pressure in the superheater at such times.

In this respect, marine practice again differs from stationary practice, where safety valves are frequently fitted to the superheater, designed to blow at a lower pressure than the main safety valves of the boiler for the purpose of insuring at all times a flow of steam through the superheater and thus preventing the possibility of the superheater being burned out. It is to be observed, however, that superheaters in land installations have been built for very high degrees of superheat, while those in marine service of the type described above have heretofore been limited to about 100 or 150 deg. F. Should the developments of marine practice require higher degrees of superheat and larger superheaters, the present practice with regard to flooding devices and safety valves may require some modification.

The Foster Superheater, Fig. 80, is made of cold drawn steel tubes expanded into headers with handholes opposite each tube end. The tubes are protected from the hot gases by cast iron sleeves tightly fitted to the outside. The outside surface of the sleeves is corrugated by flanges, greatly increasing the heating surface in contact with the gases. By means of this construction it is possible to arrange superheating surface in a minimum of space—greatly reduce the number of joints as the heating surface per lineal foot is two or three times that of a bare tube. The weight of metal prevents sudden changes in steam temperature and no other provision is necessary to protect the superheater in starting up or standing by. To obtain proper steam flow, inner cores are used to obtain the area required.

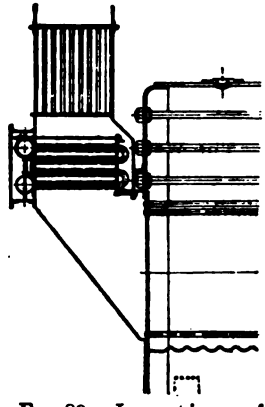


FIG. 80.—Location of Foster Superheater in Scotch Boiler.

The Foster Superheater is used with all types of marine boilers. The cut shows the arrangement used with Scotch Marine boilers. In this application waste heat only is used and the efficiency of the boiler improved by reducing the stack temperatures 100 deg. to 200 deg., while 50 deg. to 100 deg. superheat is obtained.

The fire tube superheater, manufactured by the Locomotive Superheater Co., consists of collector castings, and a system of cold drawn seamless steel U-tubes. Return bends are formed from the metals of the superheater units themselves by a patented machine forging process, making units a single continuous pipe. The units, which lead in and out of the uptake end of the flues, are expanded into flanges which in turn are fastened to the collector castings by means of a simple clamp, using but one bolt, or stud, thus facilitating removal of units.

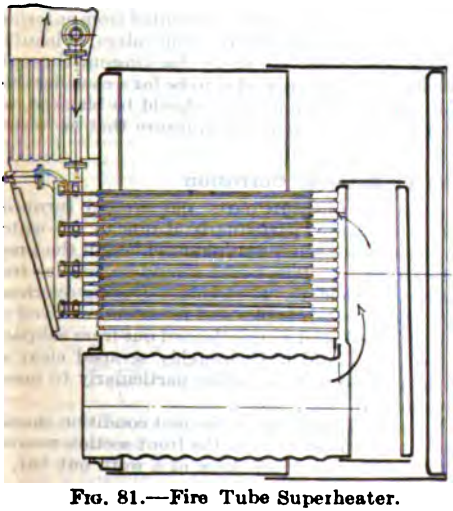


FIG. 81.—Fire Tube Superheater.

The superheater units are so connected to the collector castings that one end is in communication with header from the boiler and the other with the steam pipe to the engines. Thus steam in passing from the boiler to engines must pass through the units when superheating takes place. The units are supported in the boiler tubes by small bridges that prevent contact between tubes and superheater units.

CARE OF BOILERS

Cleanliness is the key-note in the preservation and maintenance of boilers in an efficient steaming condition. Also, **prevention is better than cure**; and if sea-water, oil and other objectionable matter is kept out of the feed supply, it greatly simplifies the cleaning process and lengthens the life of the boilers.

Preservation of Idle Boilers

There are two methods of "**laying-up**" boilers, the wet method and the dry method, the latter being preferable for prolonged periods.

The Wet Method consists merely of pumping the boiler full of water to the stop valves and thus excluding all air. The water in the boiler must be absolutely free from acid to prevent corrosion from this cause, and it is preferable to add sufficient soda to make the water slightly alkaline. It is also essential in order to prevent all corrosion to boil the water to drive off all air which might be held in solution in the water. Great care must be taken to prevent boilers in this condition being subjected to cold which would cause freezing. Previous to laying-up, the bottom and surface blows and drains from gage glasses should be opened to remove all sediment, and then tightly closed.

The Dry Method consists of emptying the boiler entirely of water, removing all moisture and closing up so that no moist air can get into the boiler. To do this remove all manhole plates from drum, and, after emptying the boiler, start a slow fire under same until all moisture is evaporated and the boiler thoroughly dry inside. Then insert several pans as large as practicable in the boiler drum and cover these with a layer of unslacked lime (ordinary quick lime) several inches thick, then close up the boiler air tight. It is imperative during the laying-up period that all water be prevented from entering the boiler, either through leaky feed connections or leaky stop valves, or insufficiently drained escape pipes from safety valves, or by backing-up through bottom and surface blows. If the laying-up period is to be for a considerable length of time, the steam, feed and blow connections should be blanked, or sections of pipe removed and the openings plugged to insure that no water may get into the boiler.

Prevention of Exterior Corrosion

Corrosion on the fire side of boiler pressure parts and around furnaces and casings is usually caused by moisture, particularly if due to sea-water and deposits of ash or refuse from fuels containing sulphur, which, in the presence of moisture, produces sulphuric acid. The boiler should be kept as free from deposits of refuse as possible, particularly when cold. During overhaul periods, the boiler surface should be brushed, baffles and retarders removed or cleaned, grate bars and bearers removed, and refuse cleaned out from ashpans and all pockets. All metal surfaces should be thoroughly scraped clear of scale and rust and painted with red lead. This applies particularly to lower parts of casings and boiler seatings.

When replacing grate bars, the grates which are in the best condition should be placed in the rear section, and those most worn in the front section nearest the fire doors, so that in case of the failure or breakage of a worn out bar, it can be more readily replaced with a new one.

Raising Steam

In raising steam in **water-tube boilers**, the circulation of the contained water is usually so rapid that temperatures throughout the setting are equalized,

and the fires may be forced immediately, without injury to the boiler. (See test of boilers of U. S. S. Cincinnati, p. 508.) **Boilers of the Scotch type,** however, require special care in raising steam and particular attention must be given any available means to promote circulation, such as hydrokineters or the use of pumps for taking cold water from the bottom of the shell and discharging it into the heated zone. Twenty-four hours should be allowed, if possible, and not less than six hours, even in an emergency. **A safe rule is to warm up the boiler and raise steam at the rate of 15 deg. F. increase in temperature per hour.** The same rules apply when cooling off the boiler. After steam begins to form the hourly temperature increase may be denoted by the pressure, approximately, as follows:

Gage pressure, lbs.	0	6	13	21	31	43	58	75	96	120	148	181	219
Temperature, deg. F.	212	230	245	260	275	290	305	320	335	350	365	380	395

Removal of Oil

"Boiling out with soda" is a process which should invariably precede placing a new boiler in service, and may become necessary at any time if oil and grease is allowed, either through ignorance or carelessness, to get into a boiler with the feed water.

The instructions issued to Babcock & Wilcox erectors will serve to illustrate an excellent procedure to follow.

"In order to assure yourselves that all boilers are thoroughly cleaned of all foreign matter, including oil or grease it will be necessary for you to proceed as follows:

"Make certain that there is no foreign matter or débris, such as hammers, expander rolls, wrenches, pieces of waste, cinders, or dirt in the drums, headers or cross boxes; obviously whoever is responsible can only make sure on this point by a personal inspection, or an inspection by someone who will really inspect.

"When the boiler is known to be clean, then all of the lower openings, that is, all openings except the steam drum manhole covers, should be put in place and the boiler filled with water up to 4 or 6 in. in the steam drums. With the water at this level a second inspection should be made to make sure that there are no leaks and that all of the lower valves are tight, in the sense that it is known definitely that there is no foreign substance between the valve and its seat. There should then be put into the upper steam drum the following quantities of common washing soda. ("Soda ash" may be used if desired.)

"In the case of an all 4-in. tube boiler, 1 lb. of soda for every 10 square feet of heating surface.

In the case of a 2-in. tube boiler, 1 lb. of soda for every twelve square feet of heating surface. This is equivalent to one pound of soda for about every 50 to 60 lbs. of water contained in the boiler), this soda to be scattered over the length of steam drum; the manhole covers then fitted in place, the boiler filled with water to the working level.

"Fire can then be made slowly and it should take at least 10 or 12 hours to get the pressure up to ten or twelve pounds. Immediately the pressure is brought up to the 10 or 12 lbs., blowing down and refilling should begin. The blowing down should never be less than one and a half or two gages of water, the filling-in an equal amount.

"During the blowing down and filling, which should take place at 15-or 20-min. intervals, the fire should be kept sufficiently active so that a pressure of 10 or 15. lbs. will be maintained constantly.

"The purpose of the continuous blowing down is to get rid of the grease and oil which the soda is intended to remove.

"Obviously, grease and oil cannot be removed if soda does not come in contact with it and pick it up. Soda cannot come in contact with the oil and grease in the tubes and headers unless the water is circulating through the boiler. The water cannot circulate through the boiler unless heat is applied and you cannot apply heat unless you take it out of the boiler, and the only way you can take heat out of the boiler under the condition

described herein is through the blowoff valves. The greater the frequency of blowings down and the application of heat the quicker will the cleaning process be accomplished, and after the boiling and blowing has continued for 24 or 36 hours uninterruptedly, then the fire should be allowed to die down and the boiler emptied, the blowoff valves and manheads taken off and the drums and each section rinsed out with clean fresh water to remove all soda (having in mind that soda is one of the worst priming elements that we can have in a boiler). After the boilers have been rinsed out, then an inspection should be made of the steam drums, the cross box and two or three of the lower tubes, to make sure that all grease and dirt has been removed.

"If you find that the oil and grease has been removed then the boiler can be closed up, filled up and be ready for service. On the other hand, if it has not been removed the operation should be repeated until the boiler is clean."

Internal Corrosion

Causes. Internal corrosion may be due to any one, or a combination of four causes: (*a*) air in the boiler water, (*b*) magnesium chloride introduced with sea water, (*c*) acidity, (*d*) electrolysis.

(a) **Air.** All water has a marked tendency to absorb air, which increases with the pressure and decreases with the temperature. Oxygen has a strong affinity for iron, producing oxide of iron or iron rust when in the presence of moisture—thus air in boiler water is ideally situated to produce corrosion or pitting, and as the iron rust promotes the opportunity for air bubbles to collect on the boiler surfaces, the corrosion thus started increases rapidly.

(b) **Magnesium Chloride.** Sea water contains the following salts:

Approximate Analysis of Sea Water

Chloride of Magnesium	245 grains per gal.
Sulphate of Magnesium	135 grains per gal.
Sulphate of Calcium	115 grains per gal.
Carbonate of Calcium	10 grains per gal.
Chloride of Sodium	1705 grains per gal.
Total salts	2210 grains per gal.

The carbonate and sulphate of calcium produce scale in a boiler but do not cause corrosion. The chloride of sodium (common salt) is not corrosive, and unless in very great excess remains in solution in the water and does no harm except to increase the tendency of the water to foam and cause priming. The presence of this salt, however, provides a ready means for detecting and measuring the amount of sea water in the boiler.

The magnesium chloride **is very corrosive** and is a very great source of danger to the metal surfaces of the boiler. Under the temperature conditions of steaming this salt dissociates forming free hydrochloric acid and magnesium hydrate. The acid attacks the iron wherever it comes in contact with it and forms iron chloride, which immediately unites with the magnesium hydrate to again form magnesium chloride, which again dissociates, producing a fresh supply of acid. It is evident therefore that a small amount of sea water in the boiler creates a condition whereby the iron is continually being wasted away and it is this danger as well as the scale deposited that render sea water such a menace in ocean-going steamships.

(c) **Acidity.** Organic and fatty acids produced from grease or from animal or vegetable oil used as lubricants may get into the boiler with the feed water. They are all corrosive and constitute a distinct menace to the life of the pressure parts.

(d) **Electrolysis.** It is claimed that all metallic corrosion is electrolytic in character and this theory has much to recommend it. However, as ordinarily used the term implies the creation of an electric current between two different metals in metallic contact and immersed in an electrolyte with the consequent wasting away of the metal in the direction of the current. This condition is supposed to be possible in a boiler particularly when the electro-conductivity of the water is increased by the presence of salt.

Prevention of internal corrosion may be accomplished in three ways: (1) Protection of the heating surfaces by a thin coating of a neutral substance not subject to corrosion; (2) elimination of the corrosive element by keeping it out of the boiler or getting it out before it can do any damage; and (3) neutralisation of the corrosive element by treatment of the water in the boiler. The application of these three methods in practice is more or less imperfect so that all three are utilized and on the extent to which they are successful will depend the life of the boiler.

(1) **Protection Against Corrosion.** A light scale, of egg-shell thickness so that the transmission of heat through the metal is not materially interfered with, forms an adequate protection so long as it is continuous and remains intact. Such a scale may be formed, by applying with a brush a thin coating of Portland Cement to a perfectly dry, clean surface. It is usually more practicable, however, to feed "**milk of lime**" to the boiler for a certain period after putting a new boiler into commission or retubing an old one. Milk of lime is a mixture of lime, either unslacked or hydrated, in water—about 1 lb. to the gallon. It should be thoroughly mixed and as much of the lime dissolved in the water as possible, and should be introduced into the hot well. One to 1½ gal. of this mixture for each 100 sq. ft. of heating surface should be fed into the boilers extending over a period of about one week.

(2) **Elimination of Corrosive Elements.** Air should be kept out of the feed system as much as possible—by using independent feed pumps rather than having these attached to the main engine—by making sure that all suction lines are at no time permitted to suck air into the system, either through leaky packing or by temporary exposure of the end of the pipe. The feed in the hot well should be kept as near the boiling point as possible without interfering with the operation of the pumps and means provided on heaters and delivery lines for pocketing air and discharging same to atmosphere. It has been recommended also that iron strips be fitted in the filter box which would attract the air and rust the strips rather than the boiler. The feed should also be delivered to the boiler in a manner calculated to liberate the air in the steam space and not allow it to get carried into the regular circulation of the water.

Sea water should be prevented from reaching the boilers by keeping the condensers absolutely tight—installing large evaporators that need not be forced and with vapor chambers large enough to prevent priming, especially in heavy weather. The evaporators should operate against a back pressure as a safe-guard against priming in spite of the apparent economy of operating at condenser pressure, and valves on vapor lines and steam lines arranged conveniently so that instant adjustment is possible if priming occurs.

All direct connection between the feed pump suction and sea water should be avoided, or if installed for emergency fire service, etc. should be blanked off. Reserve feed tanks should preferably never be used for salt water but if this ever becomes a necessity great care should be taken to completely drain them of salt water before filling with fresh, and every precaution should be taken to prevent salt water leaks into these tanks and to maintain the reserve feed supply absolutely fresh.

No system for heating salt water should by any method be so connected that the leakage of a steam coil could allow salt water to mix with the condensate. All precautions taken to keep the boiler water free from contamination with sea water will be repaid by longer life of the boilers and reduced expense for cleaning and repair.

Grease and oil must be kept out of the boiler, both on account of possible acidity and the damage which will arise from overheating the metal, not to mention the loss in economy resulting from coating the heating surfaces with oil. Efficient grease extractors should be provided on the feed delivery lines, and these and the filter box constantly looked after and kept freshly supplied with filtering material.

The best grade of mineral oil should be used for lubricating the main engines and auxiliaries and only the least possible amount employed for the purpose. The best practice permits no oil whatever being put in the cylinders other than what little may be introduced by flushing the piston rods. This applies to auxiliaries as well as the main engine.

(3) Neutralization of Corrosive Elements by Treatment of Boiler Water. An alkaline solution greatly reduces and under certain conditions entirely prevents corrosion due to oxygen in air contained in boiler water, neutralises all acids and prevents their destructive action, and converts magnesium chloride ($MgCl_2$) (introduced with sea water) to a harmless soluble salt with the precipitation of magnesia (hydrated oxide of magnesium, $Mg(OH)_2$ which is also harmless.

Alkalis furthermore dissolve animal and vegetable oils and grease, and reduce scale-making materials such as sulphate of lime ($CaSO_4$) which are contained in sea water to other forms which are more easily removed. **Use of Soda.** Lime may be used for rendering boiler water alkaline, but, on account of its convenience and more rapid action the material almost universally used is soda, either in the form of **sodium carbonate** (Na_2CO_3) or **sodium hydroxide or "Caustic Soda"** (NaOH). In its cheapest commercial form sodium carbonate is known as **"Soda Ash"** which is very satisfactory for preventing corrosion in boilers.

In a book entitled "*Marine & Naval Boilers*" by *Lyon and Hinds* published by the *U. S. Naval Institute*, 1910, *Capt. Frank Lyon, U. S. N.* describes the results of some exhaustive experiments on the use of soda for preventing boiler corrosion in which it was found that a solution having a 3-per cent normal alkaline strength rendered the water practically non-corrosive.

A normal solution of calcined sodium carbonate contains 53 grains of soda per liter of solution. A 3-per cent solution would therefore contain 1.59 grains per liter or about 0.93 grain per U. S. gallon.

Subsequent practice demonstrated the correctness of Capt. Lyon's conclusions but, the practice was discontinued on account of the fact that with a soda solution of the above strength it is possible for concentrated solution to collect in the joints causing the steel to become brittle as referred to on page 405. There is therefore a distinct **danger in the use of soda** and it must be employed only with the greatest caution and in small amounts. When so used its advantages are manifest.

Boiler compound including those brands which are of secret composition and alleged unlimited curative properties, are largely composed of soda with a mixture of materials such as Cutch, or other substances containing tannic acid, and usually a little tri-sodium phosphate. The soda reduces corrosion and the phosphate tends to reduce surface tension and prevent priming. Tannic acid, starch, dextrines, etc., affect the particles of scale-making mate-

rial precipitated from the water so as to prevent their adhering together in a hard mass, thus simplifying the cleaning process. Many of these compounds are decidedly effective and under certain conditions promote efficiency and long life of the boiler. On the other hand they may be very harmful to the boiler and should only be used when all the conditions are fully understood and watched, and it may be safely said that nothing is gained by the use of any boiler compound which cannot be duplicated by treating the boiler with soda and in certain cases a little cutch and di-sodium or tri-sodium phosphate mixed with it. The purchase of compound really represents the most expensive method of buying soda.

Rules for Use of Soda. In order that the recognized advantages of the use of soda may be realized and the dangers attending its use may be avoided, *Messrs. F. N Speller* of the *National Tube Company* and *I. Harter* of *The Babcock & Wilcox Co.* and *Mr. W. W. Smith* of the *Federal Shipbuilding Co.*, have recently prepared a pamphlet on boiler corrosion including the description of a method and apparatus for testing the boiler water and determining its alkalinity and prescribing rules for the introduction of soda to the boilers. They select 15 grains of soda ash per gallon as the amount to be carried in the boiler water and point out that this represents a slight excess over the amount necessary to combine with the magnesium chloride, which excess if maintained always insures neutralization of acids and sufficient soda to largely counteract the corrosive effect of sea water which may constantly be admitted to the boilers in small amounts.

The apparatus and method are described as follows:

There is a distinction between the two uses of soda ash, namely, the one use of bringing about a neutralization of the magnesium chloride content of the water and the other use of maintaining alkaline water in the boiler.

(1) To neutralise magnesium chloride, etc., with which soda ash reacts. In this case the additions must be repeated as long as magnesium chloride is being added to the system.

(2). To maintain alkaline water in the boiler. In this use the soda ash is intended to remain uncombined and if no substances are present with which it can react, then one addition would be sufficient.

The use of soda ash for maintaining alkaline water in the boiler is distinct from the use as described for neutralising the magnesium chloride; however, both objects are accomplished and controlled through the aid of the testing kit.

The Testing Kit consists of the assembled glass apparatus and reagent bottles as shown in Fig. 82. Each glass container is marked with an identifying letter and the letters follow in sequence of the different operations in conducting the test.

It is to be suggested that the instructions for conducting the test first be carefully read over, and then the actual manipulations be tried, using a pure or distilled water until the operator becomes thoroughly familiar with the color changes which take place. In the actual determination on samples which are to be reported on the log sheet, time must be allowed for **cooling the sample to room temperature** before conducting the test. The details for the use of this testing kit follow:

It is necessary to have a definite volume of test water to start with. A sample should be taken in a glass receptacle which has been thoroughly washed out with the water to be tested.

For this purpose the measuring pipette *A* is provided. Fill the 50 c.c. pipette *A* with the boiler water by gentle suction with the mouth to a point slightly beyond the scratch mark on the stem of the pipette, holding the charge of water in the pipette through pressure applied at the suction end with aid of the index-finger. Now gently release the pressure until the water drops to the scratch mark level. Deliver the unit charge of water thus taken to the white porcelain dish marked *B*.

With pipette *C* manipulated in the same manner as described under *A* (a piece of rubber tubing supplied with the kit may be put over the suction provided any difficulty

is experienced in filling on the scratch mark) fill exactly to the scratch mark on pipette *C* with:

Solution in bottle marked *D*. The exact quantity of solution *D* should then be added to the water sample contained in porcelain dish *B*.

Now transfer the solution from large bottle *E* by gently blowing in the rubber tubing connected with the bottle through the double hole rubber stopper so that solution *E* passes into:

Burette *F*. Continue the mouth pressure until the level of the solution in *F* is a little higher than the outlet to the fixed level elbow bend at the top of *F*. By releasing the pressure the solution should flow back so that the level stands at a fixed position at the top of *F* located by red graduation mark 5. It is absolutely essential that the volume of solution in *F* be brought to the constant level as shown by red graduation mark 5, otherwise the results will be in error. Having filled burette *F* in the manner described, the next step is to color the sample of water in the porcelain dish *B* with an indicator for this purpose.

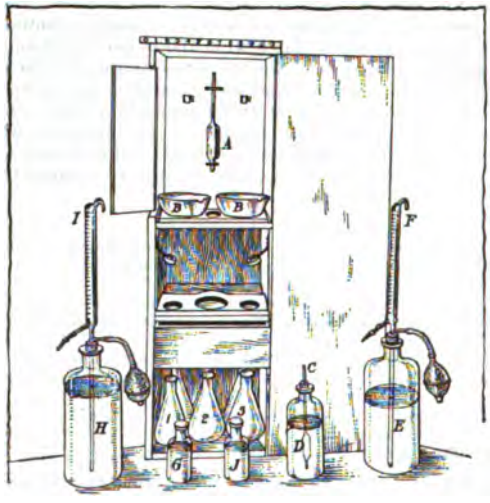


FIG. 82.—Testing Kit for Boiler Water, Speller-Harter Design.

Five drops of solution *G* is added to the water sample in *B*. This quantity of indicator *G* should be sufficient to color the water a light orange or straw color. Now by applying pressure with the thumb and index-finger on the rubber around the tit outlet to burette *F*, the solution in *F* can be dropped gradually into water sample in *B*.

Continue the dropping of solution contained in burette *F* into water in *B*, meanwhile stirring constantly with glass rod provided with the kit until there occurs a change in color from a light yellow-orange to a red color. At the instant this color change takes place stop any addition from burette *F* and immediately take the reading on the burette. This reading is recorded on the log sheet as:

F reading and should be entered at the proper space provided for the *F* reading on the log sheet. This reading should be a white line graduation reading and preferably around the graduation 15 indicating 15 grains of alkalinity in terms of soda ash per gallon of water.

This completes the determination of the alkaline (or acid) condition of the boilers and the next step is to test for the quantity of salt in the boilers. Interchange the position of *B* dishes carefully moving the one containing the sample over under the second burette,

then drop by drop with constant stirring, add solution *D* from pipette *C* until the red color again changes back to a yellow orange. Usually one to two drops of solution *D* will be sufficient for this change.

The solution *H* should be forced over into burette *I* in a manner similar to that employed in filling the first burette.

Care should be taken not to overflow the side outlet tube, at the same time remembering that it is absolutely essential that the solution be brought to the top constant level device and at a level in line with the zero reading. After *I* has been filled the next step is to add:

Five drops of indicator *J*. This addition will not cause a very noticeable change in the color of the solution. Now pressing the rubber pinch cock at the outlet of *I* drop the solution into porcelain dish *B*, constantly stirring the sample in *B*, and continue the dropping until the color of the body of liquid changes from an orange yellow to a red color. On first addition of the solution a white curdy precipitate will form, but this in no way will interfere with the final red color. The moment this change takes place stop the addition of solution from *I* and note the reading *K* on the burette *I*.

The reading *K* should preferably be below the 100 graduation mark which would be an indication that the boilers were carrying less than 100 gr. per gal. of sodium chloride (salt). The burette *I* is graduated so that the smallest subdivision is equivalent to 10 gr. per gal. Provided the reading *K* happened to come to the first division below the 200, then the proper entry for *K* reading would be 210 gr. of sodium chloride (salt) per gal. Entry of the *K* reading should be made at the proper place provided for the same on the log sheet.

The *L* number indicates the increase in gr. per gal. of sodium chloride over a certain period of time (preferably 24 hours) and therefore is arrived at by subtracting the previous log reading from the present log reading. In other words assume that the water sample on starting the boilers showed K_1 reading of 10 gr. per gal. of sodium chloride (and this had already been treated for in a manner which will be described later) and on the next twenty-four-hour period the K_2 reading gave 30 gr. per gal. then 10 the previous K_1 reading, subtracted from 30 the present K_2 reading, equals *L* number of 20.

Sampling, Soda Ash Additions, and Log Records. It is advisable to sample and treat each boiler individually for the reason that the water in different boilers will vary, and it is the water condition of each boiler which determines whether or not that particular boiler is liable to corrosion.

In conducting the tests and administering the soda ash treatment it is to be recommended that a twenty-four-hour period be established as the working basis. All of the engineering officers should have a working knowledge of the details of the method, however, the second assistant engineer aboard the ship shall be held directly and specifically responsible for the fulfillment of the duties as set forth.

Samples shall be taken from the salimometer cock or from a special valve directly connected to boiler shell of each boiler every twenty-four hours and the readings as the result of testing each water sample entered on the log sheet at the time the tests are made.

The proper amount of soda ash to be added to each boiler over a twenty-four-hour period is determined by the *L* number. The *L* number indicates the **increase of sodium chloride** in comparison with the previous twenty-four-hour period, and therefore is arrived at by subtracting the previous K_1 reading from the present twenty-four-hour period K_2 reading. The amount of soda ash corresponding to the *L* number will naturally vary depending upon the water capacity of each boiler.

Table 31 shows the requisite amount of soda ash for corresponding *L* numbers covering boiler water capacities of each boiler from 10,000 to 19,000 lbs. After having conducted the test and thereby determined the *L* number, the corresponding quantity of soda ash required for the particular boiler in question should be added to a suitable tank and put in solution through the addition of ten to twenty gallons of water, at the same time stirring until completely dissolved, then feed to this boiler by introducing the solution through the auxiliary feed pump suction.

Daily tests of water from the feed tank should be made in the same way. This water will normally be neutral and free from more than a trace of chlorides. Should the K_2 reading be higher than the K_1 to any extent in the feed tank sample and a high *L* number is found on the boiler samples, the source of the leak should be found and should receive attention as soon as possible. Should the *L* number in boiler samples be insignificant under these conditions the chlorides in feed water probably came from foaming

in one or more of the boilers. With a little experience the Engineer will be able to use these means to trace the origin of other leakages.

A facsimile of the form to be used in reporting the test follows:

Boiler Water Test Log

Ship ..... Date .....
Time .....A.M. or P.M. Tested by .....

	Boilers							Feed tank
	1	2	3	4	5	6		
F-Reading (red line)								
F-Reading (white line)								
K_2-Reading (present K reading)								
K_1-Reading (previous K reading)								
L number $= K_2 - K_1$								
Water capacity of each boiler								
Soda ash added, lbs								
Time added								

NOTE.—If the F reading falls within the red lines, this indicates an acid condition of water which is extremely dangerous.

Rule for Soda Ash. Having determined the L number as above add soda ash to the boilers as per the following Table 31:

Table 31. Soda Ash to be Added to Boilers for Corresponding L Number. Figures in Table Represent Pounds of Soda to be Added

L-reading	Total water content of boilers in pounds									
	10,000	11,000	12,000	13,000	14,000	15,000	16,000	17,000	18,000	19,000
10	0.37	0.413	0.451	0.489	0.527	0.565	0.603	0.641	0.679	0.717
20	0.75	0.83	0.90	0.98	1.05	1.13	1.21	1.28	1.36	1.43
30	1.12	1.24	1.35	1.47	1.58	1.69	1.81	1.92	2.04	2.15
40	1.50	1.65	1.80	1.96	2.11	2.26	2.41	2.56	2.72	2.87
50	1.87	2.06	2.25	2.45	2.63	2.82	3.01	3.21	3.40	3.58
60	2.25	2.48	2.71	2.93	3.16	3.39	3.62	3.85	4.07	4.30
70	2.62	2.89	3.16	3.42	3.69	3.95	4.22	4.49	4.75	5.02
80	3.00	3.30	3.61	3.91	4.21	4.52	4.82	5.13	5.43	5.74
90	3.37	3.72	4.06	4.40	4.74	5.08	5.43	5.77	6.11	6.45
100	3.75	4.13	4.51	4.89	5.27	5.65	6.03	6.41	6.79	7.17
110	4.12	4.54	4.96	5.38	5.80	6.22	6.63	7.05	7.47	7.89
120	4.50	4.96	5.41	5.87	6.32	6.78	7.23	7.69	8.15	8.60
130	4.87	5.37	5.86	6.36	6.85	7.35	7.84	8.33	8.83	9.32
140	5.25	5.78	6.31	6.85	7.38	7.91	8.44	8.97	9.51	10.04
150	5.62	6.19	6.76	7.33	7.91	8.48	9.04	9.61	10.18	10.75
160	6.00	6.61	7.22	7.82	8.43	9.04	9.65	10.25	10.86	11.47
170	6.37	7.02	7.67	8.31	8.96	9.60	10.25	10.90	11.54	12.19
180	6.75	7.43	8.12	8.80	9.49	10.17	10.85	11.54	12.22	12.90
190	7.12	8.26	8.57	9.29	10.01	10.73	11.45	12.18	12.90	13.62
200	7.50	8.67	9.02	9.78	10.54	11.30	12.06	12.82	13.58	14.34
210	7.87	9.08	9.47	10.27	11.07	11.86	12.66	13.46	14.26	15.05
220	8.25	9.50	9.92	10.76	11.60	12.43	13.26	14.10	14.94	15.77
230	8.62	9.91	10.37	11.25	12.12	13.00	13.87	14.74	15.62	16.49
240	9.00	10.32	10.82	11.73	12.65	13.56	14.47	15.38	16.30	17.21
250	9.37	10.73	11.27	12.22	13.17	14.12	15.07	16.02	16.97	17.92
260	9.75	11.15	11.72	12.71	13.70	14.69	15.68	16.66	17.65	18.64
270	10.12	11.56	12.17	13.20	14.23	15.25	16.28	17.30	18.33	19.36
280	10.50	11.97	12.63	13.69	14.76	15.82	16.88	17.95	19.01	20.08
290	10.87	12.39	13.08	14.18	15.28	16.38	17.48	18.59	19.69	20.79
300	11.25	12.80	13.53	14.67	15.81	16.95	18.09	19.23	20.37	21.51

Maintaining Alkalinity. The method of determining the proper amount of soda ash has been outlined in detail; however, nothing has been said with regard to maintaining the proper degree of alkalinity—15 gr. per gal.—within the water system. Attention is called to the fact that the required amount of soda ash corresponding to the *L* reading takes care only of the fresh sea water which has found its way into the system and does not allow for the excess (15 gr. per gal.) which is desirable. This is readily calculated, and below is shown the pounds of soda ash equivalent to 15 gr. per gal. in excess for the total water capacity of the ship's system varying from 50,000 to 150,000 lbs.

Water capacity, lbs.	Lbs. of soda ash required to give an excess of 15 gr. per gal.	Water capacity, lbs.	Lbs. of soda ash required to give an excess of 15 gr. per gal.
50,000	13	110,000	28
60,000	16	120,000	31
70,000	18	130,000	33
80,000	21	140,000	36
90,000	23	150,000	38
100,000	26		

When starting the boilers with fresh water for the first time, the amount of soda ash to be added to the water system is determined by the total water capacity of the ship's system. If, for example, the total water capacity happens to be 60,000 lbs., then by referring to the foregoing tabulation the amount of soda ash to be added to the system would be 16 lbs. or with a water capacity of 120,000 lbs. the amount of soda ash would be 31 lbs. After making the requisite addition of soda ash, and allowing time for complete solution and distribution through the system, the correctness of the amount should be checked through the taking of a sample of water and testing for alkalinity. The reading should give between 10 and 20 gr. per gal. of alkalinity in each boiler.

One possible case which has not been discussed is where the water is slightly acid or in other words gives an *F* reading within the red line marks of burette *F*. In such a case, obviously the water has used up the excess alkalinity and therefore in addition to the soda ash required by the *L* reading an amount must be added equal to the quantity shown in the following table for the corresponding water capacity plus an additional amount proportionate to the acidity.

In event it becomes necessary to blow down the boilers and use make-up water addition, the only complication this condition adds to the method is that it becomes necessary for the engineer to estimate the quantity of make-up water to be added to the system. Having estimated the quantity of make-up water required and by referring to the table it is a matter of simple division and multiplication to estimate the requisite amount of soda ash to be added in order to establish a condition of 15 gr. per gal. of alkalinity.

Should the feed tank sample or the boiler water samples consistently show an alkalinity several grains below the 15 mark, say around 5 or 7, then it would be advisable for the second engineer to slightly increase the soda ash additions as indicated by the *L* reading. In this case, care should be taken not to overdose the boilers, remembering that 20 gr. per gal. of alkalinity must not be exceeded, and that 15 gr. is the desired amount.

Babcock & Wilcox Testing Kit. In cases where it is desired to determine salinity only, and not alkalinity, The Babcock & Wilcox testing kit, Fig. 83, provides a very simple and accurate method, giving results very much more delicate than can be obtained with the rather crude determinations of the salinometer, see page 381. The following description is taken from "*Marine Steam*":

"The apparatus here described gives a convenient and accurate method of ascertaining the exact number of grains of chlorine per gallon in the water tested. It is based on the scheme for the volumetric determination of chlorine devised by *Fr. Mohr*, an eminent chemist, and requires one graduated bottle, one bottle of silver solution containing 4.738

gr. of silver nitrate to 1000 g. of distilled water, and one bottle of chromate indicator, which is a 10 per cent solution of pure neutral potassium chromate.

"**To Make Test.** Fill the graduated bottle to the zero mark with the water to be tested; add one drop of the chromate indicator; then slowly add the silver solution; keep shaking the bottle. On nearing the full amount of silver solution required, the water will turn red for a moment, and then back to yellow again when shaken. The moment it turns red and remains red, stop adding the silver. The reading on the graduated bottle at the level of the liquid will then show the amount of chlorine in grains per gallon. For example, if a permanent red color is shown when the level is midway between 150 and 200, there are 175 gr. of chlorine per gal.

"The principle of the process depends upon the fact that if some of this silver solution be dropped into water containing a chloride, a curdy white precipitate of chloride of silver will be formed. If there is also present in the water enough potassium chromate to give a yellow color, the white precipitate will continue to form as before, owing to the silver having a greater affinity for chlorine than for the chromatic acid in the chromate. But, at the moment when all the chlorine in the sample has been converted, the silver will attack the yellow potassium chromate, and chromate of silver will be formed, which is red in color. The amount of chlorine present is, therefore, shown by the amount of silver solution required to convert it all to silver chloride, and the determination of the exact point at which the chloride precipitate ceases to form is greatly facilitated by observing when the chromate indicator turns from yellow to red.

FIG. 83.

"It is not necessary to add the silver solution until the color becomes very red, as the delicacy of the reaction would be destroyed, but the change from yellow to yellowish red must be distinct and must not change on shaking. The sample of water to be tested should be neutral, as free acids dissolve the silver chromate. If it should be acid, neutralize by adding sodium carbonate. Slight alkalinity does not interfere with the reaction, but should the sample be very alkaline, it may be neutralized with nitric acid.

"Should it happen that the color does not change within the limits of the graduations, the sample may by tested by diluting with distilled water. For example, add three parts of distilled water to one part of the sample. If then, on testing the mixture, the color changes at 200, the number of grains per gallon in the original sample will be four times this reading, or 800 grains."

Use of Zincs. A time-honored method of neutralising corrosive tendencies in boilers is to suspend zinc plates in the boiler water, metallically connected to the iron. The plates are usually suspended on brackets and separated by ferrules so that the water may have free access to the entire surface, with baskets so arranged as to catch pieces of zinc oxide which scale off. Rolled zinc should be used, about ½ in. thick, and of convenient size for handling and to avoid waste in cutting; for **specifications of boiler zincs** see p. 49. The **location** should be near the incoming feed to neutralize the effect of air, and in such other places as are specially subject to pitting. It is customary to allow ¾ sq. ft. of zinc (not counting the edges) for each 100 sq. ft. of heating surface.

The use of zinc is entirely optional, and its net value is not wholly proved, particularly when other preventives such as soda are employed. However, under certain conditions, zinc prevents corrosion to the pressure parts, and where these conditions exist, zinc may be used effectively.

The theory on which zinc acts to save the boiler metal is based on the fact that it is electropositive as compared to iron, so that when in metallic contact with the iron and both immersed in an electrolyte (such as water—particularly salt water) a current is set up and flows from the zinc to the iron, with the result that the zinc oxidizes and corrodes and the iron does not. It happens that zinc oxide is electronegative to iron so that if sufficient zinc oxide adheres

to the zinc plates the current may reverse, and the iron will corrode faster than if no zinc were present. The **effectiveness of the zinc** therefore **depends on** keeping the zincs clean and free from scale and in metallic contact with the boiler.

It is probable that zinc is beneficial in another way also, namely as an absorbent of oxygen in the boiler water, by direct oxidisation.

Use of Electric Current, Cumberland System. As no corrosion can occur at the negative electrode or cathode, it has been suggested, and tried with some degree of success, that a metal electrode be suspended in the boiler water and connected by wire, insulated from the boiler, to a battery or other source of current and the circuit completed by connection to the boiler. If then, a current is sent through the circuit in the direction of the suspended metal, and through the water to the boiler parts, the metal becomes the positive electrode (anode) and is corroded while the boiler parts remain intact. If the current reverses, the reverse action takes place.

BOILER REPAIRS

The importance of keeping the steam generator in good repair is admitted by all. The fact is not always recognised, however, that "an ounce of prevention is worth a pound of cure," and it frequently happens that those in charge of boilers wait until something happens before undertaking repairs. The **cost of maintenance** will be much reduced if in addition to keeping the boilers clean, the personnel will see to it that repairs are made in ample time to prevent serious consequences.

Sectional boilers, including all of the water-tube variety, are much simpler to repair than shell boilers or those having large stayed surfaces. A defective tube can be removed and a new one installed, making the boiler practically as good as new; and this type of generator is particularly immune from the various types of patches which are characteristic of the shell boiler. The cost of maintenance of the properly designed water-tube boiler is usually decidedly less than the cost of maintenance of shell boilers, including the well-known Scotch type.

Wherever new material enters into repair work, it must comply with the requirements of the **Supervising Inspectors,** and the Inspectors should be notified of cases where anything but slight repairs are necessary. It is their prerogative to inspect the repairs and to test the boiler before again being put into service.

Autogenous welding has been found useful in many instances of repairs to water-tube boilers, but particularly effective in the upkeep of Scotch boilers. The **Supervising Inspectors** have carefully studied this subject, and provide in their Rules the requirements quoted below. **A word of caution** is necessary in this connection; namely, that where cracks develop in the plates, at calking edges or in buttstraps, it must be absolutely ascertained that the cause of these cracks is not due to brittleness of the plate brought about by the excessive use of soda, or other cause, in which case the mere closing up of a crack by welding is not sufficient to make the boiler secure. This point deserves greater attention than has been given to it in the past; and it is hoped that this feature, which has only recently been forcibly brought to the attention of engineers, should be carefully studied before some serious accident results in the necessity for drastic rules. The **Supervising Inspectors' rules for welding** follow:

"**Welding and Reinforcing by the Electric, Oxy-acetylene, or Other Process.** All calking edges on internally fired boilers may be reinforced by these processes,

"Calking edges of the shells of externally fired boilers, above the fire line only, may be reinforced.

"Cracks extending from the edge of lap to rivet, except on seams below the fire line in externally fired boilers, may be welded.

"Cracks not exceeding 30 in. in length in back connection sheets, wrapper sheets, bottoms of combustion chambers, heads, and other stayed surfaces may be repaired by welding.

"Where cracks are repaired by welding, holes shall be drilled entirely through the plate at each extreme end of the crack, except in small cracks from rivet to calking edge.

"Circumferential or lengthwise cracks not exceeding 16 inches in length in plain or corrugated furnaces may be welded.

"Where plates in back sheets of back connections, wrapper sheets of sides and bottoms of back connections of any boilers, side sheets and legs of furnaces and bottoms of furnaces of fire-box boilers, and other stayed surfaces are reduced in thickness not exceeding 40 per cent. of the original thickness, they may be reinforced, such reinforcing not to exceed an area of 200 sq. in. in any one plate.

"When such reinforcing extends over stays and braces, such stays and braces shall come completely through the reinforcing so as to be plainly visible to the inspectors.

"When the corroded portion of stayed or riveted surfaces of the back sheets or wrapper sheets or bottoms of back connections of any boilers, or side sheets and bottom sheets of furnaces or legs of fire-box boiler exceeds 300 square inches, the same may be repaired by the removal of the corroded portion and the replacement thereof by a new piece of plate, the edges of the new plate being welded in position.

"Stay bolts, braces, or rivets shall pass through the body of the new plate as before, the area of the new piece not to exceed 24 in. by 24 in., or 30 in. in any one direction, the welded edges to be V'd or beveled along the joint prior to welding.

"Where plates of shells and other parts of internally fired boilers subject to tensile strain are reduced in thickness by corrosion not to exceed 25 per cent of the original thickness, they may be reinforced, such reinforcing not to exceed an area of 200 sq. in.

"Where calking edges and laps have been reinforced, local inspectors shall require the rivets to be cut out and redriven if they find on inspection that it is necessary.

"Cracks not exceeding 24 in. in length in the shells of internally fired boilers may be repaired by reinforcing the plate to cover the crack with a piece of tested steel plate of not less than ¼ in. in thickness, said reinforcing plate to be drilled with ¾ in. holes not more than 3 in. apart. Such holes in the reinforcing plate shall be countersunk and spot welded through these to the shell of the boiler. The edges of the reinforcing plate shall also be welded to the shell of the boiler. This form of repairs shall only be allowed when made by the electric metallic arc process. Cracks not exceeding 24 in. in length in the shells of externally fired boilers above the fire line and in mud and steam drums may be reinforced by the above process.

"Cracks extending through rivet holes in single-riveted or double-riveted seams in stayed surfaces of back connections of any boilers or side sheets of legs or bottoms of fire-box boilers which are stayed surfaces may be welded up to a length of 6 feet exclusive of rivet holes.

"Where cracks extend through rivet holes in stayed surfaces, the piece extending from the rivet to the edge of the lap may be removed where convenient to do so, and the place where the piece has been removed may be replaced by being built up and reinforced by either of these processes.

"Where leaks develop around stay bolts and the stay bolts are otherwise intact, the nuts may be removed from the ends of the stay bolts, and the stay bolts may be welded into the shell by welding a beveled collar or ring around the stay bolt. The width and depth of such collar shall equal one-half of the diameter of the stay bolt. In all such cases of applying welding rings or collars around stay bolts, the material shall be hammered while in a glowing state as it is applied.

"In all cases where metal is deposited on stayed surfaces, the operator shall hammer, when practicable, the deposited metal while it is in a glowing state.

"No repair work by any welding process shall be allowed until coupons showing the character of the work proposed to be done by the applicant have been tested and submitted, together with an explanation and report of the test, to the local inspectors of the district where the work is being done. The local inspectors shall then satisfy themselves whether or not such process can be used with safety on the boilers of steam vessels.

"In every case where repairs are to be made by these processes on the boilers of steam vessels subject to the inspection of this service, the parties making the repairs are required to notify the office of the local inspectors, in writing, giving a full detailed description of the repairs to be made, the location of the vessel, and the time the repairs are to be begun, so that inspection may be had, if practicable, prior to and during the time the work is being done.

"The application for permission to use this process on boiler repairs of any particular vessel implies a guaranty on the part of the applicant that the work shall, in material, flux, and workmanship, be equal to that of the samples furnished.

"Cracks in wrought-iron or wrought-steel headers, and cracks or sand holes in cast-steel, semi-steel, ferro-steel, malleable-iron or cast-iron headers, manifolds, crosses, tees and ells may be repaired by welding cracks or flowing metal into sand holes. Such repaired material other than headers and manifolds shall be subjected to a hydrostatic test of three and one-half times the working pressure after such repairs are made. Reinforcing by building up of any of the above-mentioned articles other than headers shall not be allowed.

"When crown-bar bolts have deteriorated or wasted away at top of combustion chamber under the crown bars, such deterioration not to exceed 25 per cent of the original diameter of the bolt, such bolts may be built up or reinforced by any process of autogenous welding.

"Where tube sheets of boilers have deteriorated not to exceed 25 per cent of their original thickness, or where cracks have developed in the sheets, the same may be reinforced or repaired by any process of autogenous welding, and the beading on the ends of tubes may be welded to the tube sheets by the same process.

FUEL AND FIRING

The fuels met with in marine work are oil, wood, anthracite coal and bituminous coal. Oil is the ideal fuel, and it possesses so many advantages that its universal use is only a question of cost and supply. For **"Oil Fuel Burning,"** see p. 281. Wood is frequently employed as fuel on inland rivers and lakes, where, notwithstanding its lower heat value, its cheapness and the ease with which it may be obtained warrant its use. It requires large furnaces, small grates and restricted draft, large furnace volume being particularly necessary with the very resinous varieties. Anthracite coal has a limited use in marine work for harbor craft in certain localitites, and being smokeless and non-coking it requires no handling in the furnace and no skill in firing except with the shovel. The large sizes of anthracite such as egg, give fairly high capacity with natural draft, but the small sizes pack closely on the grate and require forced draft, even at moderate rates of combustion. Bituminous coal, or more strictly, semi-bituminous coal, forms the great staple fuel of the marine world, and it is for this fuel that the standard coal-burning apparatus is designed.

For **heat values and analyses of various fuels,** see pp. 248. and 256.

Firing

There are two distinct methods of firing bituminous coal: **"spreading"** and **"coking."** The first consists of covering the entire surface of the fuel bed with a light even layer of fresh coal—the thinner the better—and using the rake or hoe very lightly to level the fires, as this may be required. Anthracite and free-burning bituminous coal require in the hands of a skilled fireman almost no raking at all. Coking coals, however, must be broken up on the surface and leveled after each firing.

In water-tube boilers having a single wide furnace with several fire-doors, the best results are obtained by firing alternate doors, leaving a bright incandescent bed of fire between the layers of coal freshly charged. The same result may be obtained in the shell boiler with individual furnaces, each with

a single door, if the fireman is sufficiently conscientious to cover the right or left half of the grate at alternate firing.

The second method, "coking," can only be used with coking coals. It consists of charging the fresh fuel on the front of the furnace only, leaving it there a few minutes while the radiant heat from the rear of the firebed drives off the volatile matter, and cokes the exposed portion of the pile. This should then be broken up with the slice-bar, to let the heat get to the inner portion. Finally, the bank of coked fuel is pushed back and leveled, covering the rear of the grate; and after a few moments, a fresh charge is shoveled in at the front to be coked, and the process is repeated.

When strictly followed, this method of firing is very nearly smokeless, and promotes excellent combustion, particularly in those types of furnace in which the gases pass entirely from the front to the rear.

It takes considerable judgment and skill, however, to adjust the pile at the front so that it will become thoroughly coked and at the same time be sufficient in quantity to properly supply the rest of the grate and prevent choking on the one hand or too thin a fire and one full of holes on the other.

The most common method and the one doubtless giving the best steam with the least skill is really **a combination of spreading and coking.** The fire is covered to a moderate depth about two-thirds the distance from the front to the back. This bed of fuel is sliced, and then broken up and partly pushed back to cover the rear of the grate, using either the hoe or devil's claw.

A few **axioms for firing** may be stated as follows: Break up the lumps before firing. Fire light and often. Keep the fire as even and bright as possible. Do not use the fire tools too much, just enough to accomplish the result, and no more. When the fire-door is opened, shut it again as quickly as possible. Do not slice the clinkers up into the green coal or get the green coal down on the grates. Do not leave holes in the fire, particularly at the back and at the corners. Keep the fire tools in good condition.

Time Firing

Men who become skilled firemen acquire a sense of time which results in remarkably even and regular firing without the aid of a clock. It has been found, however, that in vessels fitted with a considerable number of boilers, much better and more uniform results are obtained by prearranged signals in accordance with which each door of each boiler is fired, sliced or raked at certain stated intervals, thus insuring that every man and every boiler does a proper share of work.

This method was first introduced in this country by *Commander W.W. White, U.S.N.*, and has been employed with much success, particularly in high-speed cruisers and battleships using coal fuel.

Various ingenious types of electrically operated time-firing signals have been developed. The time interval may be varied, and is controlled from the engine-room. The signal may be given by a bell or flash light, and the number of the door and operation indicated by illuminated dials. The following example is representative of good practice. Assume the boilers to each have 80 square feet of grate surface, and four fire-doors numbered in sequence from left to right.

Assume the necessary rate of combustion as 25 lbs. per sq. ft. of grate per hour. A shovel of coal as fired will usually weigh 15 to 18 lbs. Each door will therefore require $25 \times 20 = 500$ lbs. or say, 30 shovels per hour. About five shovels at each firing is good practice. Therefore, each door will require charging six times per hour, or every ten minutes. In the intervening

interval it should be sliced and raked. The time interval will therefore be set at 2½ minutes, and the signals to the fireroom will be given as follows:

Time	Rake	Slice	Fire
	Door No.	Door No.	Door No.
Zero	1	3	2
2½	3	2	4
5	2	4	1
7½	4	1	3
10	1	3	2
12½	3	2	4
15	2	4	1
17½	4	1	3
20	1	3	2

Time firing signals may be variously arranged. Red, white or blue lights may be used to indicate firing, slicing and raking; and if the signals are given for numerals in rotation, the proper sequence on the doors may be secured by interchanging the numbers, instead of having them read from left to right in order.

The object to be accomplished is regularity in the various operations on all boilers, at intervals depending on the rate of forcing. It is to be noted finally that the slice bar and rake must be used only just enough to break up and level the coal, and any further work with these tools is a waste. In fact, if the firing is very light and frequent, the slicing may be cut out entirely, or used only when the fires get dirty and the clinker requires breaking up.

Cleaning Fires

One of the advantages of oil fuel is that the fire requires no cleaning—a process which always results in loss of heat and temporary loss in capacity. Wood also (on account of the small amount of ash) requires little cleaning. With coal, on the other hand, the ash and clinker must be removed at frequent intervals, and cleaning fires is part of the regular routine.

There are **two methods of cleaning** coal fires. In the first method, the good "fire" on the portion of the grate to be cleaned is "winged" over by means of a slice bar and hoe onto the fire-bed at one side or the other, and after burning down the remaining coal as much as possible without too much loss from cold air through the fire-bed, the refuse is hauled out clean from bridge wall to dead plate. The incandescent fuel banked up at the side is then pulled or "winged" back onto the grate to ignite the fresh fuel which is then fired. By the second method the fire is first pushed back to the rear as far as possible and the refuse pulled out. The good coal at the rear is then pulled forward clear of the uncleaned portion at the rear and the refuse remaining dug up with the hoe and pulled over the good coal and out through the fire-door. The good coal is then leveled and fresh coal fired. There is little to choose between the two methods. By the winging-over process it is easier to remove the refuse, but difficult to get all the good coal over to the side. By the second method it is easy to handle the good incandescent coal, but difficult to clean the rear of the grate and keep the clinker from mixing with the good coal. A **grate with a dumping rear section** is of great benefit in cleaning fires by the second method, and while little used in marine work undoubtedly repays the extra cost and upkeep.

Ash and refuse must not be allowed to accumulate in the ash pit. This seems to be self-evident, but many a set of grates and bearers have been ruined

by ignoring this maxim. The ideal condition is reached in river steamers where the boiler floor is above the water line, and openings through the hull are provided in front of each fire-door and the discharge from the circulating pump is arranged to continually wash the refuse in the ash pit through this opening.

It has already been noted that steam admitted to the ash pit tends to soften the clinker on the grates. The same thing may be partially accomplished by keeping water in the ash pits. This is most effective with freshly cleaned fires and also tends to keep the grate bars cool at a time when they are unprotected by refuse.

Number of Fire- and Ash-doors

In shell boilers with individual furnaces, the invariable and satisfactory practice is to install one door per furnace, the diameter of the furnace (representing the width of the furnace) seldom exceeding 44 in. to 48 in. In water-tube boilers the number of doors is optional, and it is frequently the case that designers space doors too close in an effort to make the entire grate accessible. It must be remembered, however, that to the average fireman a fire-door represents a furnace, and if he has five doors instead of four to a given width of grate, he considers that his work is 25 per cent harder. He is usually inclined, also, to put as much coal into a door as he can get in, so that close spacing of the doors tends to increase the objectionable practice of too heavy firing and consequent loss of efficiency.

Too many fire-doors also leads to a greater net amount of air entering the furnace during the firing process. On the other hand, the doors must not be spaced so wide apart that the grates between the doors will become bare. Four feet or a few inches more between centres may be taken as about the maximum for good practice.

Ash-doors in water-tube boilers are usually made larger, and frequently fewer in number, than the fire-doors. A large area is essential in natural-draft jobs, and those employing forced draft on the induced-draft or closed-fireroom principle, so that no restriction shall be placed on the entrance of the air to the ashpit. In closed-ash-pit draft, the contrary is the case, as the large doors tend to increase the air leakage. In this case small, tightly closed doors are used, the number being controlled merely by the need of accessibility for cleaning.

The firing height, *i.e.*, distance from floor to bottom of door, should be about 28 in. or 30 in., but may be reduced to 18 in. or 20 in. in extreme cases. In this respect the water-tube boiler is far more advantageous than the shell boiler, where the height of grate from the floor of necessity depends on the location of the furnace, and varies through very wide limits.

AIR REQUIRED FOR COMBUSTION

The combustible matter found in fuels used for steam purposes consists of Carbon, Hydrogen and, in comparatively small amounts, Sulphur. The hydrogen exists in various chemical combinations with carbon known as hydro-carbons and also sometimes with sulphur. The carbon may exist in both free and combined form.

Air is a mechanical mixture of Oxygen and Nitrogen in the relative proportions of 23.15 per cent O and 76.85 per cent N by weight, and 20.91 per cent O and 79.09 per cent N by volume. One pound of air at 60 deg. F. occupies a volume of 13.095 cu. ft. The air required for complete combustion or oxidation of the various combustible substances may therefore be calculated.

Air Required for Combustion

Name	Symbol	Molecular weight	Air required for complete combustion per lb.		Heat value B.t.u. per lb.
			Lbs.	Cu. ft. at 60 deg. F.	
Carbon to CO_2	C	12	11.52	150.85	14,540
Carbon to CO	C	12	5.76	75.43	4,380
Carbon monoxide to CO_2	CO	28	2.47	32.34	10,160
Hydrogen	H_2	2	34.56	452.56	62,000
Methane	CH_4	16	17.28	226.28	23,850
Acetylene	C_2H_2	26	13.29	174.03	
Ethylene	C_2H_4	28	14.81	193.93	
Ethane	C_2H_6	30	16.13	211.22	
Sulphur to SO_2*	S	32	4.32	56.57	
Hydrogen, Sulphide	H_2S	34	6.10	79.88	

* Complete combustion of sulphur produces SO_2 requiring 6.48 lbs. and 84.85 cu. ft. of air per lb. and giving a heat value of 5940 B.t.u. This reaction does not usually occur in the boiler furnace.

The importance of the complete combustion of carbon to carbon dioxide (CO_2) is indicated in the table where it is shown that the incomplete combustion of carbon resulting in the formation of carbon monoxide (CO) produces only 4380 B.t.u. per lb. instead of 14,540 B.t.u. produced by complete combustion.

If the ultimate analysis of a fuel is obtained, *i.e.*, the percentage by weight of the contained C, H and S, it is a simple matter to calculate the amount of air theoretically required for combustion, thus, pounds of air per lb. of fuel (*A*) equals 11.52C + 34.56H + 4.32S. It is to be noted, however, that if the fuel contains inherent moisture this will appear in the analysis as H and O in the proportion by weight of 1 to 8. This additional H cannot strictly be considered as combustible as it is derived from water and returns to that state on combustion. In this case the H content must be corrected by subtracting a weight equal to $\frac{1}{8}$ of the free oxygen shown in the analysis and the formula becomes

$$\text{Air theoretically required per lb. of fuel} = 34.56\left(\frac{C}{3} + (H - \frac{O}{8}) + \frac{S}{8}\right)$$

Ordinary semi-bituminous coal requires theoretically about 11 lbs. or 144 cu. ft. of air for complete combustion and ordinary oil (containing about 12 per cent H) requires about 14 lbs. or 183 cu. ft. of air.

It is impossible in the practical operation of the furnace to secure complete combustion without admitting some excess air. This may be reduced to 10 per cent for oil in the best furnaces and to about 40 per cent with coal. Therefore, in practice the best conditions require at least about 15.4 lbs. or 200 cu. ft. of air per lb. of either oil or coal.

In ordinary operation these figures may run considerably higher and it is advisable when calculating the size of ventilators or fans to allow at least 260 cu. ft. of air at 60 deg. F. per lb. of oil and 300 cu. ft. per lb. of coal.

ANALYSIS OF THE GASES OF COMBUSTION

No other single test gives so much valuable information concerning the performance of a boiler as an analysis of the waste gases escaping to the uptake. Coupled with the analysis of the fuel and the temperature of the entering air and the escaping gases, the gas analysis provides a ready means for not only indicating the degree of perfection being obtained in the handling of

the furnace, but the actual heat losses in B.t.u. which are occurring at the uptake, and whether these are due to excess air, imperfect combustion, or a combination of both.

The products of combustion in boiler furnaces contain principally four distinct and separate gases: (1) Carbon Dioxide or carbonic acid gas (CO_2), the result of complete combustion of carbon; (2) Carbon Monoxide (CO), the result of partial or incomplete combustion of carbon; (3) Oxygen remaining as a free gas; and (4) Nitrogen.

Where hydrogen is present in the fuel the result of the combustion is water (H_2O) which appears in the waste gases as superheated steam, but which on being drawn into the gas sampling apparatus rapidly condenses and disappears from the volumetric analysis.

Sulphur in the fuel is, when present, usually in very small amounts. It appears in the gases as Sulphur Dioxide (SO_2), which will be absorbed in the same solution that absorbs the CO_2. This introduces an error which, however, is usually so slight as to be negligible.

As all the carbon in the fuel (barring, of course, that which may be lost in the refuse) must appear in the gases of combustion either as CO_2 or CO, the weight of dry gas escaping from the furnace per pound of carbon may be calculated as follows:

$$\text{Dry gas per lb. of carbon} = \frac{11CO_2 + 8O + 7(CO \times N_2)}{3(CO_2 + CO)}$$ in which CO_2, O and N_2 are the percentages by volume of these gases in the products of combustion.

If the percentage of total carbon in the fuel is known it is possible to deduce from the above formula the actual amount of air used per pound of fuel. However, this is more conveniently estimated from the Nitrogen content of the gases as follows:

$$\text{Dry air per lb. of fuel} = C \times \frac{3.032N}{CO_2 + CO},$$

in which C is the percentage of carbon in the fuel.

The following description of the method and apparatus for analyzing flue gases was prepared especially for this article by *E. G. Bashore, B. S.*

Sampling and Analysis of Flue Gases.

Two distinct operations are involved in determining the composition of flue gases; namely, (1) sampling of the gases, (2) analysis of the sample. Of the two operations the sampling is of more importance for the reason that if the sampling is not properly done the analysis itself will lead to misinterpretation of furnace conditions. Misleading results are worse than no results at all; therefore, the necessity of carefully planning the details of the sampling operation at the outset so that the subsequent gas analysis will represent as nearly as possible true furnace conditions.

The sampling operation may be considered under two subdivisions, (1) the type of tube used as a sampler, (2) method of collecting the sample for actual analysis.

A plain standard ¼-in. iron pipe placed with the open end in the centre of the uptake affords the type of sampler most commonly used. Obviously, a sample taken in this manner will represent the gas only from the centre stream flowing through the uptake. However, in a boiler the setting of which is fairly tight against air leakage the central portion of the gases should average within about 0.5 per cent of the CO_2 content of the whole stream. On the other hand, under unfavorable conditions, the variation in CO_2 content may be as great as 2 per cent.

An attempt to secure a representative sample from the entire area of the flue is sometimes made by using a perforated sampling pipe extending across the entire width of the

chamber. This method is not looked upon with favor by engineers in general for the reason that there is no assurance that the different perforations will admit a proportionate amount of the final sample, particularly if, as is likely to happen, the openings in the pipe become partially closed with soot. It is preferable, therefore, to have several pipes in different portions of the chamber or a movable sampler, the position of which can be altered as described.

In order to insure a uniform flow of gas through the sample tube a steam or water aspirator is used.

The sample of gas for analysis may be collected over water or may be led directly into the testing apparatus. The latter method seems preferable on account of the possibility of leakage and also because of the solubility of CO_2 in water.

It is possible to protect the gas against absorption by the liquid in the jars by the use of oil on the surface or if the constituents of the gas do not vary much, the water in the jars may be saturated with this gas before beginning the experiment. Notwithstanding these precautions, samples collected over a considerable period of time by means of the sampling jars do not check satisfactorily with the individual samples drawn directly into the apparatus, and the latter are to be preferred if taken with sufficient frequency and in such order as to cover the fluctuations in furnace conditions.

The principle upon which flue gases are analyzed depends upon the fact that certain chemical solutions have the property of absorbing certain gases and not absorbing others. Thus, a solution of potassium hydroxide or caustic potash in water will absorb CO_2, but will not absorb oxygen, carbon monoxide or nitrogen. If pyrogallic acid is added to the caustic potash solution an alkaline solution of pyrogallol is formed which has the property of absorbing both CO_2 and O. If, however, the CO_2 is first removed by means of a solution of caustic potash the gas may then be passed into the pyrogallol solution for the determination of the oxygen. The carbon monoxide is removed by a solution of cuprous chloride which has the property of absorbing that gas. Nitrogen, not being absorbed by any of the above solutions will be the remaining gas and may be determined by difference.

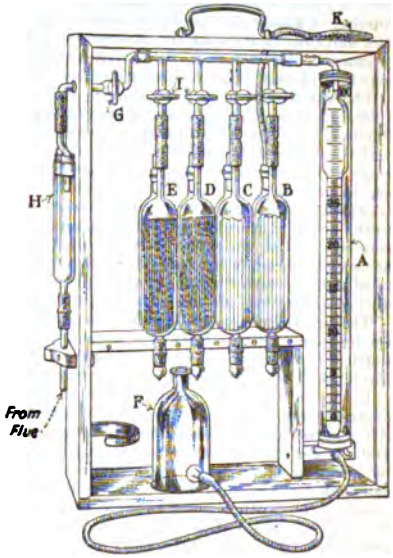


FIG. 84.—Orsat Gas Analysis Apparatus.

The Hempel Apparatus, much used for the analysis of illuminating gas, provides for the absorption of the various constituents of the gas in pipettes of varying types which are all detachable from the measuring burette. This apparatus, while capable of accurate results, requires painstaking manipulation, and is too cumbersome for work where a portable instrument is desirable. **Orsat,** on the contrary, devised an apparatus in which the measuring is done in one chamber and the various absorption processes carried out in separate chambers, the gas being passed back and forth from one chamber to the other by means of a water bottle which is alternately raised and lowered. Orsat's apparatus being self-contained, portable and requiring no washing of the various chambers, has been found very satisfactory for boiler work, and is almost universally used. Various modifications to the Orsat have been introduced and widely advertised. While it is possible that some slight improvements have been made, the original design of the Orsat is in itself so simple that it still remains a strong favorite. This apparatus is shown in Fig. 84.

The Burette *A* may have any specified volume but is usually made about 50 c. c. in the

small apparatus and 100 c.c. in the larger size. The lower portion contains about ⅓ or ½ of the total volume and is drawn down to a smaller diameter in order to extend the scale. It is to be noted that the nitrogen is always at least 79 per cent of the volume of the furnace gases so that it is the lower part of the scale (below 21) that is used in the analysis. A convenient point near the bottom of the burette is selected as the zero mark on the scale. This represents the basic volume of gas to be analysed. The unit graduations on the scale indicate hundredth parts of this volume, the subdivisions being indicated in decimals of these units. Thus, a reading of, say, 11.6 on the scale shows that there is 88.4 per cent of the original volume above this mark and 11.6 per cent below. Thus, any diminution in volume by absorption of part of the gas is indicated directly as a percentage of the original gas.

All connections to the burette and also to the pipettes are capillary tubes which have a volume which is negligible as compared to the volume of the measuring burette and absorption pipettes.

The burette *A* is used for measuring the sample and is surrounded by a water jacket to prevent any change in temperature from affecting the density of the gas being analysed.

For accurate work it is advisable to use four pipettes, *B*, *C*, *D*, *E*, the first containing a solution of caustic potash for the absorption of carbon dioxide, the second an alkaline solution of pyrogallol for the absorption of oxygen, and the remaining two an acid solution of cuprous chloride for absorbing the carbon monoxide. Each pipette contains a number of glass tubes to which some of the solution clings, exposing a large wetted surface and thus facilitating the absorption of the gas. In the pipettes *D* and *E*, copper wire or gauze is placed in these tubes to re-energise the solution as it becomes weakened. The rear half of each pipette should be drawn up to the mark on the capillary tube.

The gas is drawn into the burette through the U-tube *H*, which is filled with spun glass, or similar filtering materia', to remove soot and particles of ash, and otherwise clean the gas. To discharge any air or gas in the apparatus, the cock *G* is opened to the air and the bottle *F* is raised until the water reaches the capillary tube at the top, thus emptying the burette. The cock *G* is then turned so as to close the air opening and allow gas to be drawn through *H*, the bottle *F* being lowered to produce a suction. The gas is drawn into the burette to a point below the zero mark, the cock *G* then being opened to the air and the excess gas expelled, until the level of the water in *F* and in *A* are at the zero mark, with the water bottle held so that the level of the water in it is in the same horisontal plane as the water level in the burette. This operation is necessary in order to obtain the zero reading at atmospheric pressure. The cock *G* must be closed immediately to prevent air being drawn into the measured sample.

The apparatus, as well as all connections leading thereto, should be carefully tested for leakage. Simple tests can be made; for example: If after the cock *G* is closed, the bottle *F* is placed on top of the frame for a short time and again brought to the zero mark, the level of the water in *A* is above the zero mark, a leak is indicated.

Before taking a final sample for analysis, the burette *A* should be filled with gas and emptied once or twice, to make sure that all the apparatus is filled with the new gas. After measuring the sample and closing the cock *G*, the cock *I* in the pipette *B* is opened and the gas driven over into *B* by raising the bottle *F*. The gas is drawn back into *A* by lowering *F* and when the solution in *B* has reached the mark in the capillary tube, the cock *I* is closed and a reading is taken on the scale of the burette, the level of the water in the bottle *F* being brought to the same level as the water in *A*. The operation is repeated until a constant reading is obtained, this reading showing directly the percentage of CO_2 in the flue gases.

The gas is then driven over into the pipette *C* and a similar operation is carried out. The difference between the resulting reading and the first reading gives the percentage of oxygen in the flue gases.

The next operation is to drive the gas into the pipette *D*, the gas being given a final wash in *E*, and then passed into the pipette *C* to neutralise any hydrochloric acid fumes which may have been given off by the cuprous chloride solution, which, especially if it be old, may give off such fumes, thus increasing the volume of the gases and making the reading on the burette less than the true amount.

The process must be carried out in the order named, as the pyrogallol solution will absorb not only oxygen but also carbon dioxide, while the cuprous chloride solution will in addition to carbon monoxide also absorb oxygen.

The absorption of carbon monoxide in the acid cuprous chloride solution is especially slow and a considerable number of passes are often necessary in order to insure complete absorption. The exact mechanism of the action between acid cuprous chloride and carbon monoxide has not been definitely determined. There exists a definite compound as the result of a combination of cuprous chloride and carbon monoxide; however, in dilute solutions present in gas analysis, the reagent behaves as if it dissolved the carbon monoxide. Due to the feeble union which exists, an acid cuprous chloride solution which has absorbed considerable carbon monoxide is liable to give off part of its carbon monoxide content to a gas containing no carbon monoxide. This explains the advisability of having a second pipette of practically fresh acid cuprous chloride which has not suffered deterioration through excessive carbon monoxide absorption.

It will be noted that the constituent gases are absorbed in the three pipettes *B*, *C* and *D* by coming in contact with considerable surface which has been immersed in the absorbing solution and to which a thin film of the solutions still adheres. While this extended film-covered surface greatly facilitates the absorbing action on first contact, the amount of actual solution is of course small and almost immediately becomes saturated with all the gas it will absorb. It is necessary therefore to draw the gas out of the pipette after the first contact, thus wetting the tube surfaces with fresh solution and preparing them to absorb more gas on forcing the sample into the pipette a second time. This operation must be repeated in each pipette until two consecutive readings show no further diminution in volume. This is especially necessary in pipette *B*, where the CO_2 is absorbed, for if the absorption of that gas is not completed in pipette *B* it will be completed in *C*, in which the solution absorbs both CO_2 and O, but the reading will be set down as O instead of CO_2 thus giving erroneous and very misleading results.

Each time that the gas is *measured* in burette *A* after removal from the various pipettes, care must be taken to bring the level of the solution in the pipettes up with the capillary tube, as otherwise it is obvious that errors will be introduced in the analysis. However, it is not necessary to carefully and completely empty the pipettes when the object is merely to wet the tubes with fresh solution and complete the absorption. Much time may be saved by running the level of the liquid in the pipette up to the top of the tubes only and immediately driving the gas back again.

A skilled operator can hasten results and take two or more samples closely together in rapid sequence by running them through the apparatus at the same time. Thus, after measuring the first sample and completing the absorption of CO_2, it may be forced into pipette *C* and left there while the second sample is taken, measured and forced into pipette *B*. Leaving the second sample thus "stored" in *B*, the first sample is taken up, the absorption of the oxygen completed and observed and this sample then run into the pipette *D* where it remains while the second sample is put through the CO_2 absorption and stored in pipette *C*. The apparatus is then ready for the third sample of gas, and so on, as fast as the process can be repeated. Considerable attention is required to avoid mixing up the samples, and usually there will be time to complete one entire analysis before taking a fresh sample.

It sometimes happens that one of the glass tubes in the pipettes *B*, *C* and *D* may take up a position in the pipette directly over the bottom U-connection to the other part of the pipette. When this occurs, very obvious errors may be caused (particularly if the gas is driven into the pipette rapidly) by a portion of the sample being forced completely through the tube and the U-connection into the other half of the pipette, and so lost. The loss thus entailed is noted on the next burette reading and assumed to be due to absorption.

The analysis made by the Orsat apparatus is volumetric; if the analysis by weight is required, it can be found from the volumetric analysis as follows:

Multiply the percentage by volume of each gas by either the density or the molecular weight of the gas, and divide the product by the sum of all the products; the quotients will be the percentages by weight. For most work sufficient accuracy is secured by using the even value of the molecular weights, namely: carbon dioxide, 44; carbon monoxide, 28; oxygen, 32; nitrogen, 28.

The Solutions. To prepare the solution for the absorption of carbon dioxide, one (1) part by weight of potassium hydroxide (caustic potash) dissolved into two (2) parts by weight of water will give a solution of the proper strength. In other words, 500 g. of potassium hydrate in 1 litre of water.

For preparation of potassium pyrogallate for the absorption of oxygen, 5 g. of pyro-

gallic acid are dissolved in 15 c.c. of water. To this is added a solution of 120 g. of caustic potash dissolved in 80 c.c. of water. The pyrogallol solution and the caustic potash solution should not be mixed in the air but the pyrogallic acid solution should first be introduced into the absorbing pipette, then followed by the caustic solution. Pyrogallic acid in combination with a caustic solution has a marked chemical reducing action. Weak caustic solutions in combination with pyrogallic acid should be avoided for the reason that they lead to erroneous results through a reduction of part of the residual oxygen to carbon monoxide. A corresponding error might be introduced if an old or spent solution of pyrogallic acid and caustic is used. Normally 1 c.c. of potassium pyrogallate will absorb about 8 c.c. of oxygen. Stating this another way, with a gas averaging 8 per cent. oxygen and with the use of 100 c.c. of potassium pyrogallate, the quantity of reagent should be good for 100 determinations. The deterioration of the absorbing power of the potassium pyrogallate can be checked by determining the oxygen value in a sample of air. The reagent should be rejected unless it absorbs 20.6 to 20.8 per cent of oxygen, assuming the practical absence of carbon dioxide in the sample of air.

To prepare the cuprous chloride solution for the absorption of carbon monoxide, dissolve 150 g. of cuprous chloride in 250 g. of water. Then add 750 c.c. strong hydrochloric acid (specific gravity 1.2). This solution should then be introduced into a bottle which contains small rolls of 20 to 60 mesh copper gauze extending to the top. The quantity of solution made at each time should be of sufficient bulk to completely fill the bottle to the stopper. Within 48 hours this solution should clarify to a straw color.

Potassium hydroxide is not always readily procurable due to market conditions and locality. The substitution of sodium hydroxide (caustic soda) for potassium hydroxide is permissible; however, the solution made from the sodium hydroxide will prove slightly more sluggish in the absorption of the gases. This is especially true of the sodium pyrogallate used for oxygen absorption. In the preparation of the potassium pyrogallate, 120 g. of potassium hydroxide is dissolved in 80 c.c. of water or a ratio of 3 parts by weight of potassium hydroxide to 2 parts by weight of water. With the substitution of sodium hydroxide, the solubility is very much less, and this condition probably accounts for the fact that the oxygen absorption is relatively lower. A saturated solution of sodium hydroxide in water should be made for the preparation of sodium pyrogallate in the event the potassium hydroxide cannot be procured.

From a chemical standpoint, potassium hydroxide is the same as caustic potash and sodium hydroxide is the same as caustic soda; however, if circumstances necessitate the purchase of either of these chemicals, then the article should be specified as "potassium hydroxide, pure, in sticks" or "sodium hydroxide, pure, in sticks."

The normal oxygen content of air by volume is 20.9 per cent. The symbol for oxygen is O_2, for carbon dioxide CO_2 and for carbon monoxide CO. In the case of combustion of pure carbon and in the absence of the formation of carbon monoxide, the total of the carbon dioxide and oxygen constituents as shown by the gas analysis should equal 20.9. Carbon monoxide (CO), produced by the imperfect combustion of carbon, will occupy twice the volume of the oxygen entering into its composition and will increase the volume of the flue gases over that of the air supplied for combustion in the proportion of 1 to $\frac{100 + \frac{1}{2}\text{ the per cent of CO}}{100}$. Therefore, provided pure carbon is the fuel, the sum of the carbon dioxide and the oxygen content plus one-half the carbon monoxide content as determined by the gas analysis should equal 20.9. However, this statement is of purely theoretical interest, and is a condition which never obtains in ordinary flue gas analysis.

The fact that the gases absorbed in a gas analysis invariably never totals 20.9 is because chemically pure carbon is never used as fuel, and practically all commercial fuels contain a variable amount of hydrogen. Part of the oxygen of the air combines with the hydrogen content of the fuel to form steam which condenses to a liquid while the nitrogen associated with this oxygen remains as a gas. From this it is apparent that the percentage of nitrogen in the flue gas must be more than the 79.1 per cent present in the air. The extent of this change of volume may be calculated. Assume (as might be the case with fuel oil) the analysis shows 84 per cent nitrogen. One hundred cubic feet of air contains (in round numbers) 79 cu. ft. of nitrogen which now becomes 84 per cent of the flue gas. However, the volume of the gas is 0.79 ÷ 0.84 = 0.94 or 94 per cent of the initial volume of air measured under the same conditions of temperature and

pressure. From this it follows that 6 of the 21 volumes of oxygen in 100 volumes of air have combined with hydrogen to form water. According to the following equation

$$2H_2 + O_2 = 2H_2O$$

it will be seen that the volume of water formed, so long as it remains in the form of water vapor, will be twice that of the oxygen from which it was formed.

Gas analysis from an engineering standpoint is discussed more in detail in *Gas & Fuel Analysis for Engineers* by *A. H. Gill* and *Gas & Fuel Analysis* by *Alfred H. White.*

Removal of Scale

Scale-making salts in the feed water which actually enter the boiler remain therein on the evaporation of the water into dry steam and are deposited in the form of scale which varies in density and hardness according to the character of the material. This scale has to be removed by mechanical means, only a very small percentage being removable by means of the bottom and surface blows. Various types of scaling tools are used in Scotch boilers, to fit the outside of the tubes and furnaces, and as elsewhere noted in this paper the tubes are frequently arranged in banks, with vertical passages between same, to facilitate the internal cleaning.

In the case of water-tube boilers, some types are absolutely inaccessible, and cannot be cleaned, and these must be used with absolutely pure water or else radically dismantled or the parts renewed.

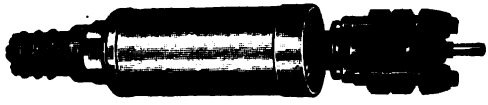


FIG. 85.—Roto Tube Cleaner.

In other types of water-tube boilers like the Babcock & Wilcox, special attention is given to accessibility in the design, and the straight tubes greatly facilitate internal cleaning. Hand scrapers may be used for soft scale and mud. The best method of cleaning the interior of the tubes, however, is by means of rotary cleaners provided with cutters and driven by small turbines operated by compressed air or hydraulic pressure. These are very effective in removing scale, but care must be taken not to allow the cutters to revolve for any length of time in one place, as there is the possibility of actually cutting into the metal. In Fig 85. an illustration is given of the **Roto cleaner** which shows this type.

Summary of Care and Operation

1. Keep boilers clean, inside and outside.
2. Keep boilers and furnaces in repair.
3. Make frequent tests of samples of boiler water, and keep water slightly alkaline.
4. Keep air out of feed water.
5. Keep sea water out of the boilers.
6. Prevent grease from entering boilers.
7. In tending water, remember that continuous feed results in more even steaming and aids in obtaining efficiency. Avoid opening the check wide or closing tight while the boiler is in operation.
8. Avoid excessive use of blow valves. The surface blow is to be operated only to remove scum, which may have a tendency to cause priming. The

bottom blow is to be used only as occasion may require when emptying the boiler or freshening the feed.

9. Coal Fuel. Break up lumps. Fire light and often. Keep fires bright and level. Avoid holes and thin spots. Avoid excessive use of fire tools.

10. Oil Fuel. Keep atomisers clean and in repair. Avoid use of excess air. Reduce air supply to give slight haze at top of funnel. For "Oil Fuel Burning," see p. 281.

BOILER TESTS

The object of making boiler tests is usually to determine the rate of evaporation of water per square foot of heating surface, referred to as boiler "**capacity,**" or the amount of water evaporated per pound of fuel, termed "**economy,**" or both. The conditions of draft and various other data collected are often of great importance, and should always be noted.

The American Society of Mechanical Engineers has codified the method to be followed in conducting boiler tests, and it is recommended that these rules be referred to and closely followed by those making such tests (see A.S.M.E. Power Test Code).

The report of the tests of the Babcock & Wilcox boiler at the U. S. Navy Fuel Oil Testing Plant includes a discussion of the steps to be followed in taking data and working up the results of a boiler test in which oil fuel is used; see p. 500.

Boiler Efficiency. With the development of reliable methods of determining the heat values of fuels when completely consumed, as by use of the Bomb calorimeter, it has become possible to express the economic results obtained from boilers as a "**percentage of efficiency**" *i.e.*, the percentage of total heat contained in the fuel which is absorbed by the boiler and imparted to the steam. This includes the efficiency of the furnace and is properly **defined as combined efficiency of boiler and furnace.** It may be expressed by the formula:

$$\text{Combined efficiency in per cent} = \frac{\text{Total Heat in Fuel per lb.}}{\text{Total Heat Absorbed by boiler per lb. of fuel}} \times 100.$$

It is generally preferable to base the heat value of fuel and the heat absorbed by the boiler on dry fuel, owing to the accidental variation in the moisture content. However, the results are identical if these values are based on the fuel "as fired." It is to be noted, however, that any moisture in the fuel must be evaporated in the furnace, and results in economy are influenced thereby. On the other hand, sometimes moisture in coal may, through mechanical effects such as prevention of clinkers, or reducing loss of fine coal, actually give higher economic results than if the fuel were perfectly dry.

Strictly speaking, "boiler efficiency" is represented by the proportion which the heat absorbed by the boiler bears to the heat contained in the gases which pass over the heating surfaces, but while this is of scientific interest, the practical point of view requires that the boiler must be considered in combination with the furnace which is an indispensable element in the generation of steam from fuel. Therefore the heat in the fuel as fired without deduction for losses through grate bars, etc., is the basis on which boiler efficiency is usually considered.

Care Required in Making Tests. Not only should the water fed into the boiler and evaporated and the fuel used be carefully determined (preferably by weighing), but special pains must be taken to prevent errors due to leakage and to difference in conditions at starting and stopping the test.

Many absolutely worthless tests have been and are conscientiously reported by people who have failed to realise that skill and experience is required if accurate results are to be obtained.

Heat Balance. As a check on the accuracy of boiler tests the heat balance, while of necessity somewhat of an approximation, is valuable, and it is important to secure the data required to make the necessary calculations: namely, (1) Heat value of fuel in B.t.u. per lb. of fuel, (2) Water (equivalent) evaporated into dry steam per pound of fuel from and at 212 deg. F. (this allows for temperature of feed water, boiler pressure and dryness factor of steam, or superheat if any), (3) Moisture in fuel per pound, (4) Percentage of carbon and hydrogen in fuel, (5) Analysis of waste gases, (6) Temperature of waste gases, (7) Temperature of air entering furnace.

The calculations are as follows:

Heat Balance

Based on Dry Fuel

Item	B.t.u.	Per cent
1. Heat absorbed by boiler: equals water evaporated per lb. of fuel F. & A. 212 deg. F. × 970.4.		
2. Loss due to evaporation of moisture in fuel = percentage of moisture based on dry fuel × [(212 − t) + 970.4 + 0.48 (T − 212)]. This assumes fuel at temperature t.		
3. Loss due to superheated steam formed by the combustion of hydrogen = $9H$[(212 − t) + 970.4 + 0.48(T − 212)].		
4. Loss due to (dry) escaping gases of combustion and excess air = (T − t) × 0.24 × W.		
5. Loss due to incomplete combustion of carbon = $C \times \frac{10{,}150\ \mathrm{CO}}{\mathrm{CO_2} + \mathrm{CO}}$.		
6. Loss due to combustible (carbon) in refuse = C' × per cent refuse × B.t.u. per lb. of combustible in the ash (usually taken as 14,600 B.t.u.).		
7. Loss due to evaporating moisture in the air entering the furnace = W^1 × 0.48(T − t).		
8. Loss due to radiation and unaccounted for = the difference between the total B.t.u. in the fuel and the sum of all the losses plus the heat absorbed by the boiler (Item 1).		

In which

970.4 = latent heat of steam at 14.8 lbs. absolute pressure (atmosphere).

0.48 = specific heat of steam at atmospheric pressure.

t = temperature of air in boiler room.

T = temperature of flue gases.

(212 − t) = B.t.u. necessary to evaporate 1 lb. of water at 212 deg. F. into steam at atmospheric pressure.

0.48(T − 212) = B.t.u. necessary to superheat 1 lb. of steam at atmospheric pressure from 212 deg. F. to temperature T.

H = percentage by weight of hydrogen from the ultimate analysis of the dry fuel.

0.24 = average specific heat of chimney gases.

W = weight of dry chimney gas per lb. of dry-fuel.

C = per cent of carbon in dry fuel by ultimate analysis.

C' = per cent of carbon in refuse.

CO + CO_2 = per cent of CO + CO_2 by volume from flue gas analysis.

10,150 = number of heat units generated by burning to CO_2 one pound of carbon in carbon monoxide.

W^1 = weight of moisture in air supplied per lb. of fuel.

Dry Steam—Calorimeter Tests. The quality of steam produced by a boiler is of paramount importance as affecting the results of boiler tests and also of special interest in operation, as the production of reasonably dry steam (say, under one per cent) is an essential requirement of a well-designed boiler. The throttling calorimeter, invented by Prof. Cecil H. Peabody, one form of which is shown in Fig. 86, provides a ready means of making this test.

Great care is necessary in sampling the steam to prevent drawing in with the sample, moisture of condensation from the sides of the steam pipe. Perforated or slotted nipples which extend across the pipe are erroneously supposed to draw the steam equally through such openings. Generally, however, the opening at the side of the pipe nearest the calorimeter supplies the sample, with the introduction of excess moisture. Such sampling nipples, while recommended by some are therefore to be avoided. The following is taken from "Marine Steam:"

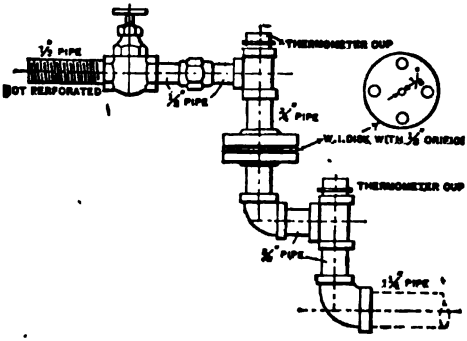


FIG. 86.—Throttling Calorimeter.

"The method best adapted **to insure obtaining a fair sample of steam** for testing, is to take it from the centre of the vertical portion of the steam pipe as near the boiler as possible. Use a straight open-ended nipple, provided with a long thread on one end so that it may be screwed into the steam pipe far enough to bring the open end at or near the centre of the current of steam ascending from the boiler, and as far removed as possible from the sides of the pipe, which are always coated with a thin film of moisture. **Do not use perforated or slotted nipples,** as they have been found to give very inaccurate results.

"The throttling calorimeter depends on the principle that dry steam when expanded from a higher to a lower pressure, without doing external work becomes superheated, the amount of superheat depending on the two pressures. If, however, some moisture be present in the steam, this must necessarily first be evaporated, and the superheating will be proportionately less. The limit of the instrument is reached when the moisture present is sufficient to prevent any superheating.

"Assuming that there is no back pressure in the exhaust, and that there is no loss of heat in passing through the instrument, the total heat in the mixture of steam and moisture before throttling, and in the superheated steam after throttling, will be the same, and will be expressed by the equation

$$H - \frac{xL}{100} = 1150.3 + 0.48(t - 212) \quad \text{or}$$

$$x = \frac{H - 1150.3 - 0.48(t - 212)}{L} \times 100$$

in which x = percentage of moisture; H = total heat above 32 deg. F. in the steam at boiler pressure; L = latent heat in the steam at boiler pressure; 1150.3 = total heat in the steam at atmospheric pressure; t = temperature shown by lower thermometer of calorimeter; 212 = temperature of dry steam at atmospheric pressure.

"Theoretically the boiler pressure is indicated by the temperature of the upper thermometer; but, owing to radiation, etc., it is usually too low, and it is better to use the readings of the boiler gage, if correct, or better still to have a test gage connected on the ½-inch pipe supplying the calorimeter.

"If the instrument be well covered, and there is as little radiating surface as possible, the above assumption that there is no loss of heat in passing through the instrument may be nearly, though never quite, correct. On the other hand it is more than likely to be very far from correct, and, to eliminate any errors of this kind, Mr. Geo. H. Barrus recommends a so-called "calibration" for dry steam. This, again, involves an assumption which is open to some doubt, which is that steam, when in a quiescent state, drops all its moisture and becomes dry. No other practical method, however, has been proposed, and this is, therefore, the only method used at the present time. Some engineers, however, refuse to make any calibration, but, instead, make an assumed allowance for error.

"To make the calibration, close the boiler stop valve, which must be on the steam pipe beyond the calorimeter connection. Keep the steam pressure exactly the same as the average pressure during the test, for at least fifteen minutes, taking readings from the two thermometers during the last five minutes. The upper thermometer should read precisely the same as during the test, and the lower thermometer should show a higher temperature; this reading of the *lower* thermometer is the calibration reading for dry steam, which we will call T.

"Calculation of results, allowing for radiation by calibration method:

$$x = \frac{0.48(T - t)}{L} \times 100$$

in which x = percentage of moisture; T = calibration reading of lower thermometer; t = test reading of lower thermometer; L = latent heat of steam at boiler pressure.

"There is a limit in the range of the throttling calorimeter which varies from 2.88 per cent at 50 lbs. pressure to 7.17 per cent at 250 lbs. When this limit is reached a small separator may be interposed between the steam main and the calorimeter, which will take out the excess of moisture. By weighing the drip from the separator and ascertaining its percentage of the steam flowing through, and adding this to the percentage of moisture then shown by the throttling calorimeter, the total moisture in the steam may be ascertained. It is seldom, however, in a well-designed boiler, that any but a throttling calorimeter becomes necessary."

Measurement of Superheat. The common method of inserting thermometers in mercury wells placed in the steam pipe almost invariably results in errors due to radiation, the reported temperatures being too low. These errors increase with the temperature and frequently amounting to 6 or 8 deg. F. Corrections, while possible, are difficult to make with certainty, and the use of thermo-electric couples, or the type of instrument utilizing the variation of electrical resistance of metal as affected by heat, are coming to be recognized as the most reliable methods of determining superheat. Such instruments should be frequently calibrated to eliminate errors.

Smoke. Observations of the density of smoke are at best merest approximations, depending not only upon the personal equation of the observer but upon distance, angle of observation, and, particularly, on atmospheric and sky conditions. No better method, however, seems to have been developed than to judge by the eye the relative color and density of the smoke. The process has been simplified and partially standardized by the use of the well-known Ringelmann Charts. These are six in number, a white card, No. 0, representing no smoke, and five cards graded in tint representing varying density of the smoke up to No. 5, solid black. The intermediate tints are obtained by drawing black lines of specified thickness at right angles to each other on white paper and based on centres of 10 millimeters. The checker-work takes a distinct shade according to the relative width of the lines and spaces. Thus, No. 1 has lines 1 mm. wide and spaces 9 mm.; No. 2, lines 2.3 mm., spaces 7.7 mm.; No. 3, lines 3.7 mm., spaces 6.3 mm.; No. 4, lines 5.5 mm., lines 4.5 mm.

The charts may be made of any suitable dimensions (not less than 5-in. × 8-in. shaded portion) to give a distinct color at a distance of about 50 feet. When set up in sequence in a horizontal line at this distance they provide a means of comparison by the eye with the color and density of the smoke.

It is customary to interpolate between the numbers when making observations, particularly with oil fuel or stokers, where the conditions are steady and the smoke usually very light. Thus, ¼ would represent a light haze, 1½ a moderate light smoke, etc. With bituminous coal, fired by hand, the readings are not usually so refined, but they should be taken at very frequent periods to obtain an average and maximum and minimum figures, to cover the very fluctuating smoke conditions.

Tests of Scotch Boilers

Authentic and reliable tests of Scotch boilers are extremely rare. Table 32 is submitted through the courtesy of *Mr. Charles F. Bailey*, Engineering Director of the Newport News Shipbuilding and Dry Dock Company. This includes published data and also the results of tests made under Mr. Bailey's supervision.

Estimated Performances Scotch Boilers

Mr. W. W. Smith, Chief Engineer of the Federal Shipbuilding Company, has prepared the following valuable curves, which form an excellent basis for estimating the performance of Scotch boilers under various conditions, see Figs. 87 and 88.

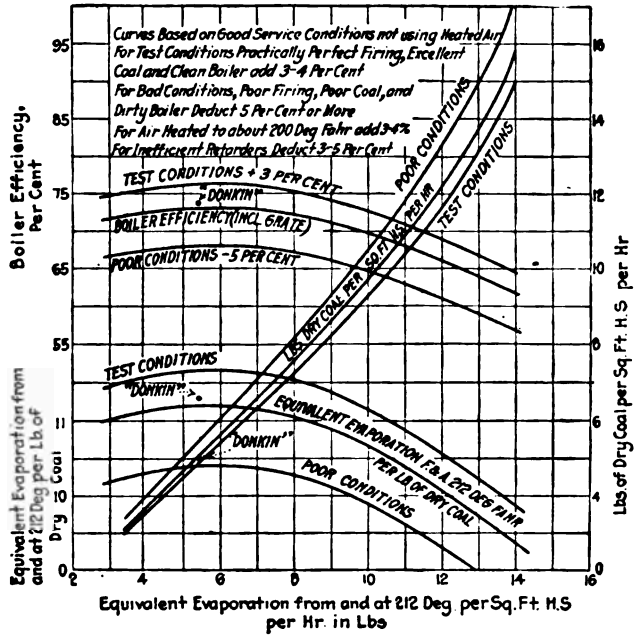

Curves Based on Good Service Conditions not using Heated Air
For Test Conditions Practically Perfect Firing, Excellent Coal and Clean Boiler add 3-4 Per Cent
For Bad Conditions, Poor Firing, Poor Coal, and Dirty Boiler Deduct 5 Per Cent or More
For Air Heated to about 200 Deg Fahr add 3-4%
For Inefficient Retarders Deduct 3-5 Per Cent
TEST CONDITIONS + 3 PER CENT
"DONKIN"
BOILER EFFICIENCY (INCL GRATE)
POOR CONDITIONS - 5 PER CENT
LBS. DRY COAL PER SQ. FT. H.S. PER HR.
POOR CONDITIONS
TEST CONDITIONS
TEST CONDITIONS
"DONKIN"
EQUIVALENT EVAPORATION F. & A. 212 DEGS. FAHR. PER LB. OF DRY COAL
"DONKIN"
POOR CONDITIONS
Boiler Efficiency, Per Cent
Equivalent Evaporation from and at 212 Deg per Lb of Dry Coal
Lbs of Dry Coal per Sq. Ft. H.S. per Hr
Equivalent Evaporation from and at 212 Deg. per Sq. Ft. H.S. per Hr. in Lbs

FIG. 87.

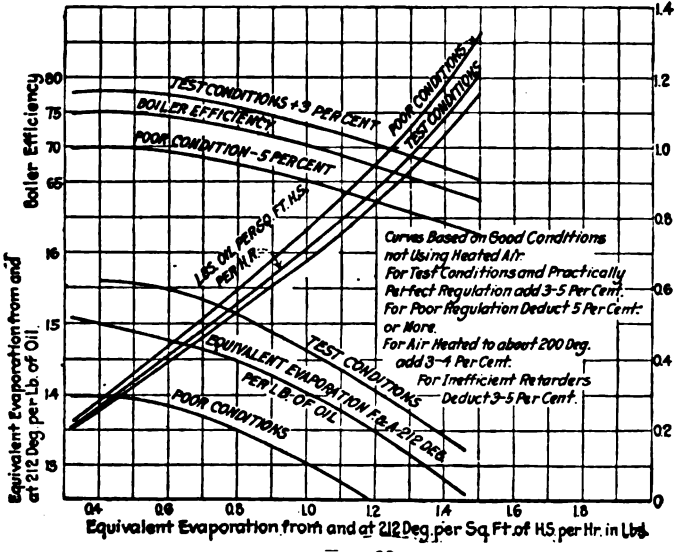

TEST CONDITIONS + 3 PER CENT
BOILER EFFICIENCY
POOR CONDITION - 5 PER CENT
POOR CONDITIONS
TEST CONDITIONS
LBS. OIL PER SQ. FT. H.S. PER HR.
Curves Based on Good Conditions not Using Heated Air
For Test Conditions and Practically Perfect Regulation add 3-5 Per Cent.
For Poor Regulation Deduct 5 Per Cent. or More.
For Air Heated to about 200 Deg. add 3-4 Per Cent.
For Inefficient Retarders Deduct 3-5 Per Cent.
TEST CONDITIONS
EQUIVALENT EVAPORATION F. & A. 212 DEGS. PER LB. OF OIL
POOR CONDITIONS
Boiler Efficiency
Equivalent Evaporation from and at 212 Deg per Lb. of Oil
Lbs of Oil Burnt per Sq. Ft. of H.S. per Hr.
Equivalent Evaporation from and at 212 Deg. per Sq. Ft. of H.S. per Hr. in Lbs.

Table 32. Tests of Scotch Boilers Burning Coal

Ship	U.S.S. Minneapolis	U.S.S. Minneapolis	N. N. S. & D. D. Co. power house	Meteor	Fusi Yama	Colchester	Iona	Ville de Douvres
Type	S.E.	D.E.	S.E.	D.E.	S.E.	D.E.	S.E.	S.E.
Date of test	July, 1895	July, 1895	1/20/92	6/24/88	11/14/88	11/9/88	7/14/90	9/8/91
Duration, hours	12		10	17	14	11	16	9
Number boilers tested	.....	.....	.....	2	1	2	2	4
Diameter	10′ 3⅛″	15′ 9″	.....	.....	.....	.....	.....	.....
Length	8′ 6″	20′ 0″	.....	.....	.....	.....	.....	.....
No. and Diam. furnaces per blr.	2–37″	8	2	6	3	6	2	3
Length of grate	70″	84″	.....	.....	.....	.....	.....	.....
Diam. of tubes	2½″	2¼″	.....	.....	.....	.....	.....	.....
Heating surface total	976	6,401	1,338	6,648	2,257	5,820	3,160	7,340
Grate surface total	32	186.7	42	208	52	220	42	236
Ratio H.S. to G.S.	30.5	34.3	31.85	32	43.4	26.5	75.2	31.1
Boiler pressure	83	142	100	145	57	80	165	106
Air pressure	.....	.....	.....	.....	.....	.....	0.2	.....
Lbs. coal per hour	260	1,740	679	4,005	987	5,742	942	7,380
Lbs. coal per sq. ft. of G. S.	8.1	9.31	16.14	19.25	18.98	26.1	22.4	31.3
Per cent of refuse	11.32	12.22	8.0	.....	.....	.....	.....	.....
B.t.u. per lb. coal as fired	.....	.....	.....	12,270	12,760	13,280	14,830	14,390
Lbs. water per hour	.....	.....	.....	29,860	7,860	43,020	8,616	66,180
Lbs. water per lb. coal, actual	7.69	8.39	7.41	7.46	7.96	7.49	9.15	8.97
Lbs. water per lb. combustible	8.67	9.55	8.05	.....	.....	.....	.....	.....
Lbs. water from 212 deg. per lb. coal	9.04	9.89	7.91	8.21	8.87	8.53	10.63	9.84
Lbs. water from 212 deg. per lb. combust.	10.19	11.25	8.59	9.62	10.10	9.34	10.42	9.94
Lbs. water per sq. ft. of H.S. actual	2.05	2.28	3.65	4.49	3.48	7.39	2.73	9.02
Quality steam per cent	.....	.....	97.4	.....	.....	.....	.....	.....
Temp. feed	.....	.....	.....	163	129.5	113	106	158
Temp. stack	.....	.....	.....	791	578	835	452	910
Boiler efficiency, per cent	.....	.....	.....	62.0	67.2	62.0	69.2	66.1
Type of draft	.....	.....	.....	.....	.....	.....	.....	.....
Factor of evaporation	1.175	1.177	1.067	1.100	1.113	1.138	1.162	1.096
Data obtained from	.....	.....	.....	←——*The Engineering Magazine*, Jan., 1913, page 501.——→				

Ship	El Sud	Minerva	Smithfield	Mobjack	John D. Rockefeller	N. N. S. & D. D. Co. power house	N. N. S. & D. D. Co. power house	N. N. S. & D. D. Co. power house
Type	D.E.	S.E.	S.E.	S.E.	S.E.	S.E.	S.E.	S.E.
Date of test	6/13/12	5/29/01	5/1/12	5/14/12	9/5/14	8/4/14	8/5/14	8/6/14
Duration, hours	4	8½	3½	4	6	8	8	8
Number boilers tested	1	4	2	2	3	1	1	1
Diameter	13' 10" M.	.....	10' 0"	10' 0"	14' 4" I.D.	12' 10"	12' 10"	12' 10"
Length	20' 6"	.....	13' 5"	13' 2"	11' 6"	11' 9"	11' 9"	11' 9"
No. and diam. furnaces per blr.	6—42¾"	3—46⅜"	1—58"	1—58"	3—48"	3—42"	3—42"	3—42"
Length of grate	6' 6"	7' 0"	7' 0"	7' 0"	5' 6"	6' 0"	6' 0"	6' 0"
Diam. of tubes	3¼"	2½"	3½"	3½	2¾" O.D.	2¾"	2¾"	2¾"
Heating surface total	3,455	9,232	2,310	2,292	7,035	2,044	2,044	2,044
Grate surface total	139	325	67.6	67.6	177	60	60	60
Ratio H.S. to G.S.	24.8	28.4	34.1	34.0	39.75	34.1	34.1	34.1
Boiler pressure	164	147	56	58	216	112	114	114
Air pressure	.....	0.65 F.R.	.....	.....	0.23" ash pit	−0.05 A. P. −0.9 Stack	−0.01 A. P. −0. 9 Stack	−0.01 A. P. −0.9 Stack
Lbs. coal per hour	2,170	9,452	1,814	1,600	3565	900	1,200	1,500
Lbs. coal per sq. ft. of G.S.	15.6	29.1	26.9	23.7	20.1	15	20	25
Per cent of refuse	19.8	.....	.....	.....	8.89	10.75	9.89	8.36
B.t.u. per lb. coal as fired	14,420	14,612	15,080	15,080	14,470	14,296	14,324	14,207
Lbs. water per hour	17,000	83,550	15,670	15,332	35,190	8,631	11,799	14,276
Lbs. water per lb. coal actual	7.84	8.84	8.63	9.58	9.89	9.59	9.83	9·52
Lbs. water per lb. combustible	.....	.....	.....	.....	11.0	11.05	11.12	10.52
Lbs. water from 212 deg. per lb. coal	8.84	10.34	9.5	10.1	10.55	11.15	11.34	10.95
Lbs. water from 212 deg. per lb. combust.	.....	.....	.....	.....	11.76	12.50	12.55	11.91
Lbs. water per sq. ft. of H.S. actual	4.9	9.05	6.78	6.68	5.0	4.22	5.77	6.98
Quality steam per cent	95.4	.....	.....	.....	98.78	98.4	98.0	98.4
Temp. feed	141	95	143	185	188	110	113	108
						←	—Intake—	→
Temp. stack	330	699	600	.....	466	637	78.5	64.7
Boiler efficiency, per cent	59.2	68.4	60.8	64.6	70.5	75.4	76.4.	74.4
Type of draft	Nat.	C.F.R.	Nat.	Nat.	Howd	Nat.	Nat.	Nat.
Factor of evaporation	1.126	1.170	1.100	1.054	1.081	1.148	1.145	1.149
Data obtained from	Test	British Navy ← Report →	Test	Test	Sea Trial	Test	Test	Test

Table 33. Trial Trip Data. S. S. "Edward Luckenbach"

November 26, 1916

Time	Revolutions			S.H.P.		Fuel oil		Water for turbine only		
	Counter reading	Period	R.P.M.	T. meter reading, 4834	Period	Meter reading	Lbs. per hr.	Lbs. per S.H.P.	Lbs. per hour Period	Water rate Period
12.00	125,436	...	....	70.0	....	900				
12.10	126,282	846	84.6	78.0	3190	967				
12.20	127,135	853	85.3	76.0	3140	1075				
12.30	127,999	864	86.4	81.5	3400	1114				
Average....	... ...	...	85.4	....	3265	.....	3200	0.980	39,700	12.17
12.40	128,876	877	87.7	85.0	3600	1178				
12.50	129,764	888	88.8	85.5	3670	1268				
1.00	130,658	894	89.4	89.5	3865	1362				
1.10	131,547	889	88.9	86.5	3715	1422				
1.20	132,440	893	89.3	88.5	3820	1522				
1.30	133,346	906	90.6	89.5	3920	1610				
Average....	.....	...	89.1	....	3787	....	3710	0.980	42,600	11.25
1.40	134,261	915	91.6	90.0	3980	1715				
1.50	135,189	928	92.8	94.0	4210	1812				
2.00	136,122	933	93.3	94.0	4235	1902				
2.10	137,055	933	93.3	95.0	4280	1986				
2.20	137,986	931	93.1	94.0	4230	2065				
2.30	138,916	930	93.0	92.5	4150	2175				
Average....	... ...	...	92.8	....	4181	....	4210	1.007	46,650	11.17
Measuring water for all purposes										
2.30	138,916	.....	.....	.....	.....	2173				
2.40	139,844	928	92.8	96.0	4300	2254				
2.50	140,773	929	92.9	96.0	4310	2352				
3.00	141,702	929	92.9	97.0	4350	2450				
Average....	... ...	...	92.9	....	4320	....	4140	0.960	54,800	12.67
Overload.	98			110.5		5230		With 2 overload nozzle valves open.		

Draft = 10 ft. 4½ in. forward, 18 ft. 9 in. aft. Displacement = 7300 tons.
Speed by log 14 knots.
Fuel Oil 18,170 B.t.u. per lb.
0.894 Sq. Gr. at 60 deg. F.

Trials of S. S. Edward Luckenbach

The following trial trip data of the S. S. Edward Luckenbach is furnished through the courtesy of *Mr. Robert Warriner*, Chief Engineer of the Bethlehem Shipbuilding Corporation; see Tables 33 and 34.

Table 34. S. S. "Edward Luckenbach"
Trial Trip Data

Period	Engine room			Vacuum	Temperature, deg. F.				
	Pressure, lbs. gage				Bearings		Injection	Disch.	Eng. Room
	Main steam	Steam chest	Oil		For'd	Aft			
12.00 to 12.30	176.8	137.3	8.8	28.75	103.5	85.5	44.0	69.0	74.0
12.30 to 1.30	179.4	159.3	12.0	28.75	105.3	98.0	45.0	64.7	75.3
1.30 to 2.30	191.0	176.1	17.0	28.75	103.0	117.7	45.0	66.3	79.0
2.30 to 3.00	194.0	179.0	17.0	28.90	102.0	116.5	43.5	65.0	78.5
Overload	195	170	28.75						

Period	Fire room			Fuel oil at burners	Temperature, deg. F.			Air pressure, in.
	Pressure, lbs. gage				Fuel oil heater			
	Main steam at boilers							
	Stbd.	Centre	Port		Inlet	Outlet	Room	
12.00 to 12.30	185.5	182.8	183.5	121.7	103.7	247.8	96.7	0.9
12.30 to 1.30	182.1	179.3	179.9	146.4	104.6	248.2	92.9	0.87
1.30 to 2.30	192.4	189.2	189.4	165.7	99.3	253.6	88.9	0.93
2.30 to 3.00	196.0	192.5	192.5	158.8	97.5	254.0	86.0	0.98

Table 35. Trials of S. S. "La Brea" Fitted with G. E. Geared Turbines

Voyage	Date	Total distance in knots	Average speed in knots per hour	Total barrels delivered	Name of port	Total fuel used steaming on voyage in bbls.	Total bbls. per knots steaming	Total bbls. used in port	Total bbls. per knot steaming and in port	Time in port, hrs.	Barrels discharged per hour.	Cost at 80c. per bbl. per knot steaming and in port (fuel)	Shaft horsepower hours	Pounds of oil per shaft horsepower
1	Mar. 9 to 15, 1916	650	9.5	62,578	Oleum	574	0.883	142	1.10	36½	1714	$0.88	116,000	1.67
2	Mar. 15 to 26	2037	11.33	73,600	Seattle	1459	0.716	152	0.79	37½	1962	0.632	505,000	0.941
3	Mar. 28 to Apr. 6	2108	11.01	64,676 8,642	Vancouver Seattle	1584	0.751	155	0.824	26 13½	2487 640	0.659	530,000	1.01
4	Apr. 8 May 16	9254.5	10.97	52,109 19,045	Taltal Antofagasta	6896	0.745	117	0.757	33¾ 15½	1544 1228	0.605	2,330,000	1.00
5	May 17 to 23	450	11.20	77,292	Oleum	321	0.713	134	1.01	39	1980	0.808	118,000	0.911
6	May 24 to July 1	9196.5	10.65	71,824	Antofagasta	6900	0.75	109	0.762	25	2660	0.609	2,200,000	1.06
7	July 1 to Aug. 9, '16	9184	10.39	71,791	Port San Luis to Antofagasta Chile and return	6875	0.748	107	0.76	42.3	1697	0.609	2,230,000	1.04

Total pounds of oil steaming = 8,270,000. Total S.H.P.-hrs. 8,029,000. Pounds of oil per S.H.P.-hr. = 1.03.

Table 36. Trials of S. S. "Los Angeles" Fitted with Triple Expansion Engines

1	Apr. 9 to 15th	423	9.4	67,674	Oleum	556	1.31	164	1.70	64½	1039	$1.36	74,600	2.48
2	Apr. 16 to 25	1845	10.13	74,739	Vancouver	1656	0.897	218	1.01	33¾	2214	0.808	394,000	1.41
3	Apr. 27 to May 25	6549	10.22	73,734	Panama	5579	0.851	169	0.877	46¾	1577	0.701	1,420,000	1.315
4	May 26 to 29	220	9.1	72,572	Oleum	221	1.00	123	1.56	41	1770	1.24	39,000	1.88
5	May 30 June 27	6348	10.6	72,538	Bal-Boa	5462	0.86	159	0.885	35	2072	0.708	1,460,000	1.255
6	June 29 Aug. 7–16	9151	10.24	71,007	Port San Luis to Antofagasta Chile and return	8293	0.906	186	0.926	56.3	1261	0.74	2,130,000	1.31

Total pounds of oil steaming = 7,210,000. Total S.H.P.-hrs. 5,538,000. Pounds of oil per I.H.P.-hr. = 1.300.

Trials of S. S. La Brea and S. S. Los Angeles

Through the courtesy of *Mr. W. L. R. Emmet*, Consulting Engineer of the General Electric Company, the comparison of the performance of the steamships "La Brea" and "Los Angeles" are given in Tables 35 and 36. These vessels are sister ships of approximately 2500 S.H.P., built at the same time, in the same yard. The S. S. "La Brea" is equipped with General Electric geared turbines and the S. S. "Los Angeles" is fitted with reciprocating engines of the standard triple expansion type.

Tests of Machinery of S. S. "Oklahoma"

The following results were obtained in some tests of the machinery installation of the Tanker "Oklahoma" by *Messrs. M. B. Myrick, M. C. Sherman, R. A. D. Preston and J. Stuart Sneddon*, students of the graduating class of 1910 of the Massachusetts Institute of Technology, in Feb., 1910. These tests were carefully conducted, and all measuring tanks and instruments carefully checked. The vessel was operated under working conditions, and it is believed that the results represent a fair example of marine installations of this character.

The vessel was equipped with three single-ended Scotch boilers, each 16 ft. 6 in. in diameter × 10 ft. 8⅜ in. long, designed for a working pressure of 185 lbs. per sq. in. Each boiler had 604 tubes 2.5 in. diameter × 7 ft. 3 in. long, and four 41 in. corrugated furnaces with independent combustion chambers.

The Howden system of forced draft was installed. Oil fuel was used, atomized by Lässoe-Lovekin oil burners using compressed air for atomising at 1 pound pressure. Retarders were installed in the boiler tubes.

The vessel was fitted with one main propelling engine of the vertical triple-expansion open-frame type, with cylinders 28 in. × 46 in. × 76 in. diam. × 54 in. stroke. The valves were of the piston type, operated by Stephenson links and fitted with Lovekin assistant cylinders.

The vessel was fitted with the usual allotment of auxiliaries, the feed pumps and air pumps being independent of the main engine. The following is a summary of the results:

Average revolutions per minute	57.8
Indicated horsepower main engine	2190.0
Oil burned per hour	2740.0 lbs.
Oil per hour per I.H.P. of main engine	1.25 lbs.
Steam consumption per I.H.P., main engine only (water rate)	14.55 lbs.
Steam consumption per I.H.P. main engine, including steam used by auxiliaries	17.2 lbs.
Steam used by auxiliaries per hour per I.H.P. of main engine	2.65 lbs.
Per cent of total steam used by auxiliaries	15.40
Steam pressure at boiler	167.0 lbs.
(Reduced from original working pressure account of condition of boilers.)	
Per cent of moisture in steam	1.7
Water evaporated F. & A. 212 deg. F. per sq. ft. of heating surface	4.18 lbs.
Water evaporated per lb. of oil F. & A. 212 deg. F.	14.65 lbs.
B.t.u. per lb. of oil	19,500.0
Combined efficiency of boiler and furnace	72.8

Report of Tests on Experimental Oil-burning Babcock & Wilcox Boiler at the Fuel Oil Testing Plant

By Angelo Conti

"Following the adoption of the Babcock & Wilcox type of boiler for the battleship 'Oklahoma,' burning fuel oil exclusively, a boiler of the 'Oklahoma' type was purchased for experimental purposes and thoroughly tried out at the Fuel Oil Experimental Plant, U. S. Navy Yard, Philadelphia.

"Particulars of this boiler are as follows:

No. of sections	22
No. of groups of 4—2-in. diam. tubes in each header	9
Total number of 2-in. tubes	792
Length of tubes between headers	9 ft. 0 in.
Inside diameter of steam drum	3 ft. 0 in.
Total heating surface, sq. ft.	4000
Furnace volume, cu. ft.	445

"The tests reported herein were run under the supervision of *Lieutenant-Commander* (now *Captain*) *J. J. Hyland*, Head of the Fuel Oil Testing Plant, in collaboration with the writer, who was specially detailed by the Bureau of Steam Engineering to assist in the compilation of the technical data.

"An article by Lieutenant-Commander John J. Hyland, U. S. Navy, published in the May, 1914, number of the Jour. A.S.N.E., profusely illustrates the type of burners and registers used in these tests and also gives a diagrammatic plan of the Testing Plant. Observations were taken thoughout each test at 5 min. intervals, with the exception of chimney gas analyses, which were taken every 15 min. by means of an Orsat apparatus. The evaporation was computed at the end of each hour and the tests continued until it was proved that the results obtained were consistent and accurate.

"In order to allow the boiler to be thoroughly heated up, the first two or three hours of each test were purposely discarded, the enclosed summary representing an average of the observations taken during the last hours of each test. Owing to the uniformity of conditions obtained, as well as the care used in eliminating observational errors, and, finally, the consistency of the results themselves, no doubt can be felt as to the accuracy of these tests," of which Table 37 gives a summary.

Table 37. Tests of Oil-burning Babcock & Wilcox Boiler

At Fuel Oil Testing Plant, U. S. Navy Yard, Philadelphia, Pa.

1.	Test No.	93	94
2.	Date, 1913	July 16	July 16
3.	Barometer	30.12	30.095
4.	Length of test averaged, hours	2	3
5.	No. of burners used	3	5
6.	Aggregate opening of air registers, sq. ins.	77.1	128.5
	Steam Observations		
7.	Steam pressure, lbs. gage	300	300
8.	Calorimeter, lower therm., deg. F.	310	309
9.	Calorimeter, normal reading, deg. F.	312	312
10.	Feed temperature at boiler, deg. F.	199	200.6
11.	Water actually evaporated, lbs. per hour	13,052	23,644
	Oil Observations		
12.	Oil burned, lbs. per hour	871	1,578
13.	Oil pressure at burners, lbs. gage	191	218
14.	Oil temper. at burners, deg. F.	180	180
	Air Observations		
15.	Dry bulb, deg. F.	78.9	84.1
16.	Wet bulb, deg. F.	65.3	67.5
17.	Fire-room temper., deg. F.	97	99.7
18.	Stack temper., deg. F.	446	509
19.	Fire-room pressure, ins. of water	1.00	2.00
20.	Velocity head, by Pitot Tube, ins. of water	0.29	0.73
21.	Smoke, Ringelmann scale	0-2	0-1
	Flue Gas Analysis, from First Pass		
22.	Carbon dioxide, per cent.	12.625	14.266
23.	Oxygen, per cent.	5.04	2.71
24.	Carbon monoxide, per cent.	0.00	0.00
25.	Nitrogen (by difference) per cent.	82.335	83.024
	Calculated Results		
26.	Steam quality	0.999	0.9982
27.	Factor of evaporation	1.0684	1.0662
28.	Actual evaporation per pound of oil, lbs.	14.984	14.983
29.	Equivalent evaporation per pound of oil, lbs.	16.01	15.98
30.	Actual evaporation per sq. ft. of H.S.	3.263	5.911
31.	Equiv. evaporation per sq. ft. of H.S.	3.49	6.30
	Oil Results		
32.	Oil burned per sq. ft. of H.S., lbs. per hour	0.218	0.3945
33.	Oil burned per cu. ft. of furnace, lbs. per hour	1.95	3.54
34.	Oil burned per burner, lbs. per hour	290	315.6
	Air Results		
35.	Relative humidity, per cent.	47.5	42.0
36.	Dew Point, deg. F.	57.2	58.5
37.	Moisture, lbs. per cu. ft.	0.0007495	0.000788
38.	Weight of air at atmospheric conditions, lbs. cu. ft.	0.07375	0.0729
39.	Forced draft blower output, cu. ft. per. min.	6,280	9,910
40.	Air supplied to burners, cu. ft. per. min.	3,360	5,710
41.	Velocity through vanes of air registers, ft. per min.	6,280	6,400
42.	Cu. ft. of air per pound of oil.	231.2	217
43.	Cu. ft. of air theoretically required for complete combustion of 1 lb. of fuel	193.3	196.5
44.	Excess air, per cent.	19.7	10.5
	Results from Dry Gases and Ultimate Analysis		
45.	Lbs. of dry gases per pound of fuel	17.156	15.263
46.	Lbs. of dry air per pound of fuel	16.98	15.15
47.	Per cent excess air over amount theoretically required for complete combustion	30.66	14.3

Tests of Oil-burning Babcock & Wilcox Boiler (Continued)

At Fuel Oil Testing Plant, U. S. Navy Yard, Philadelphia, Pa.

95	96	97	98	99
July 17	July 17	July 23	July 24	July 25
30.01	29.96	30.085	29.855	30.095
3	3	3	4	3
8	10	8	10	13
205.6	270	205.6	270	334
300	300	300	300	302
286.2	267.3	303.1	292.3	272.7
312	312	309	309	309
201.3	200.6	200.2	200.7	200.6
37,504	48,216	36,638	47,662	60,325
2,532	3,192	2,486	3,182	4,005
196	207	188	194	193
180	180	180	180	181
78.8	79.8	87.2	86.0	80.35
67.6	68.4	72	73	64.5
94.3	96.6	101	98.4	96.9
620	685	649	690	720
3.26	4.25	3.21	4.20	5.11
1.66	2.48	1.67	2.54	3.516
0–1	0–1	0–1	0–1	0–1
14.2	14.66	14.34	15.106	15.15
2.75	2.15	4.34	3.093	2.73
0.00	0.00	0.00	0.00	0.00
83.05	83.19	81.32	81.801	82.12
0.985	0.974	0.9966	0.9903	0.9789
1.0545	1.046	1.0652	1.0594	1.0501
14.812	15.104	14.738	14.98	15.062
15.62	15.80	15.70	15.87	15.82
9.376	12.054	9.159	11.915	15.081
9.89	12.61	9.76	12.62	15.84
0.633	0.798	0.6215	0.795	1.001
5.69	7.18	5.58	7.16	9.00
316.5	319.2	311	318.2	308
55.5	55.3	47.7	53.2	42
61.5	62.5	64.6	67.0	54.7
0.0008705	0.00090	0.0009646	0.000555	0.0006928
0.07385	0.07375	0.07265	0.0725	0.074
14,700	18,000	14,860	18,340	21,380
9,125	11,450	9,335	11,840	13,980
6,400	6,110	6,540	6,320	6,030
216	215	225.2	223.2	209.6
194	194.2	197.2	197.5	193.5
11.3	10.7	14.25	13.0	8.3
14.645	14.878	15.23	14.49	14.44
15.23	14.78	14.77	14.10	14.11
14.53	10.95	25.72	13.92	14.62

Heat Balance

Test No.	93		94	
	B.t.u.	Per cent	B.t.u.	Per cent
48. Heat utilized	15,536	79.57	15,507	79.42
49. Loss by imperfect combustion				
50. Loss with dry chimney gases	1,472	7.54	1,535	7.15
51-*a* Loss with moisture from hydrogen in the fuel	1,366	.....	1,396	.....
b Loss with moisture in the fuel	6	7.13	6	7.28
c Loss with moisture in the air	20	.....	20	.....
52. Radiation and error	1,125	5.76	1,061	6.15
53. Total	19,525	100.00	19,525	100.00

Method of Obtaining Items in Table

Item.

1, 2. For reference.

3. Observed.

4, 5, 6. For reference.

7. Observed by spring gage connected to steam drum, pressure practically constant.

8. From a Barrus throttling calorimeter, fitted with a closed end standard perforated nipple, connected to the main steam pipe, rising vertically from the steam drum, 18 in. above stop valve.

9. From calibration after each test, taking readings with the pressure both rising and falling with no steam leaving the boiler except through the calorimeter.

10. Observed from calibrated thermometer located on the branch leading from main feed pipe to boiler.

11. Total weight of water fed to boiler, corrected hourly for inequality of water level as found from calibration of steam drum.

12, 13, 14. Observed.

15, 16. From dry and wet bulbs thermometers mounted on a perforated board placed on the suction duct of the forced draft blower to insure a good ventilation.

17. Observed.

18. Observed from a Hohmann and Maurer, nitrogen filled mercurial pyrometer calibrated at the Naval Engineering Experiment Station, U. S. Naval Academy, Annapolis, Md.

19. Observed by U-gage.

20. From a standard Pitot Tube placed in the centre of the forced draft blower discharge pipe about 5 feet from the top. This pipe is 24 in. inside diameter, 15 ft. high and the top is belled out and open to the fire-room. The Pitot tube is connected to an Ellison inclined gage graduated to hundredths of an inch.

21. From observation through a large mirror adjacent to a Ringelmann chart.

22, 23, 24, 25. Orsat apparatus, drawing samples through a long perforated tube from the first pass. This location was selected on account of a leaky casing, as evidenced by the lower percentage of CO_2 shown by a Uehling continuous recording apparatus drawing from the base of the stack.

26. Computed from the formula $X = \frac{0.47(T - t)}{L} \times 100$, in which X = percentage of moisture; T = calibration reading of lower thermometer (item 9); t = test reading of lower thermometer (item) 8; L = latent heat of steam at boiler pressure.

27. Factor of evaporation = $\frac{L + (h - h_o)}{970.4}$ where L is the latent heat of vaporisation, h the heat of the liquid, h_o is the number of heat units contained in the feed water entering the boiler and 970.4 represents B.t.u.'s required to convert one pound of water already heated at 212 deg. F. into steam at atmospheric pressure.

Heat Balance (Continued)

95		96		97		98		99	
B.t.u.	Per cent	B.t.u.	Per cent	B.t.u.	Per cent	B.t.u.	Per cent	B.t.u.	Per cent
15,157	77.64	15,331	78.52	15,235	78.04	15,400	78.88	15,351	78.64
1,857	9.51	2,152	11.02	2,052	10.51	2,111	10.81	2,210	11.32
1,463	.....	1,495	.....	1,471	.....	1,496	.....	1,514	
6	7.64	6	7.80	6	7.69	6	7.77	6	7.87
23	.....	23	.....	25	.....	14	.....	17	
1,019	5.21	518	2.66	736	3.76	498	2.54	427	2.17
19,525	100.00	19,525	100.00	19,525	100.00	19,525	100.00	19,525	100.00

28. Item 11 + item 12.
29. Item 23 × item 27.
30. Item 11 + 4000 sq. ft. H.S.
31. Item 31 × item 27.
32. Item 12 + 4000.
33. Item 12 + 445.
34. Item 12 + item 5.
35, 36. From items 15 and 16 entered in the Psychrometric Tables published by the Weather Bureau.
37. Density of aqueous vapor from Marks & Davis Steam Tables, corresponding to item 36, multiplied by item 35.
38. The weight of one cubic foot of dry air is computed from the formula:

$$P = \frac{0.080723}{1 + 0.0020389\ (t - 32)} \times \frac{b - 0.378e}{29.921}$$

where t is item 15, b is the barometer pressure corrected to the static pressure of the air in the discharge pipe of the forced draft blower, expressed in inches of mercury; and e is the pressure due to the vapor in the air also in inches of mercury. Item 37 is added to P.

39. The velocity of the air at the centre of the blower discharge pipe is computed from the formula

$$V = 1097 \sqrt{\frac{\text{item } 20}{\text{item } 38}} \text{ feet per minute.}$$

The average air velocity is from 8 per cent to 10 per cent lower than the centre velocity, the reduction factor being taken from the results of tests made at the University of Wisconsin and borne out by tests made at the Naval Engineering Experiment Station. (See Journal of the American Society of Naval Engineers, November, 1912, page 1143.) As stated above, the discharge pipe has an internal diameter of 24 in.

40. Item 39 corrected from results of leakage-tests. (See note.)
41. $\dfrac{\text{Item } 40 \times 144}{\text{Item } 6}$.
42. Item 40 + Item 12.
43. Item 43 = $\dfrac{11.5C + 34.8\left(H - \dfrac{O}{8}\right) + 4.35S}{\text{Item } 38}$ cu. ft. per pound of fuel, where C, H, O and S are the percentages by weight of carbon, hydrogen, oxygen and sulphur respectively, as found from the ultimate analysis of the fuel.
44. (Item 42 + Item 43) × 100.

45. Computed from formula

$$W = \frac{4\,CO_2 + O + 7CO}{3\,(CO_2 + CO)} \times C$$

where CO_2, O and CO are items 22, 23 and 24 respectively and C represents the amount of carbon, by weight, per pound of fuel as found from the ultimate analysis.

46. Computed from formula $\frac{3.03\,N}{CO_2 + CO} \times C$ where N, CO_2 and CO are items 25, 22 and 24 respectively and C represents the amount of carbon per pound of fuel, as found from the ultimate analysis.

47. Computed from formula $\frac{O}{0.261N - O} \times 100$, where O and N are items 23 and 25.

Heat Balance.

48. Item 29 × 970.4.

49. Loss in B.t.u. per pound of fuel = $10{,}120\,\frac{CO}{CO_2 + CO} \times C$, in which CO and CO_2 are percentages by volume from the fuel gas analysis, and C is the proportion of carbon in the fuel as determined by ultimate analysis.

50. Item 45 × (Item 18 − Item 17) × 0.246 × C.
The mean specific heat of dry chimney gases is approximately taken as 0.246 and C is the proportion of carbon in a pound of fuel, as determined by the ultimate analysis.

51-a. Loss in B.t.u. per pound of fuel = $9H(H_T - h_t)$ where H = weight of hydrogen contained in one pound of fuel from ultimate analysis.
H_T = total heat per pound of steam at stack temperature.
h_t = heat per pound of water at fireroom temperature.

51-b. Loss in B.t.u. per pound of fuel = $W(H_T - h_o)$ where W = weight of water contained in one pound of fuel.
H_T = total heat per pound of steam at stack temperature.
h_t = heat per pound of water at the temperature of the oil (Item 14).

51-c. Loss in B.t.u. per pound of fuel = $\frac{\text{Item 46}}{\text{Item 38}}$ Item 37 $(H_T - h_t)$ where
H_T = Total heat per pound of steam at stack temperature.
h_t = Heat per pound of water at fireroom temperature.

52. By difference.

53. From ultimate analysis of fuel, as follows:

Kind of oil	Texas	Carbon	86.00
Gravity, deg. Baumé	26	Hydrogen	12.70
Flash Point, deg. F	222	Nitrogen	0.19
B.t.u. per pound	19,525	Sulphur	0.30
		Water and sediment	Traces

"Cincinnati" Tests

The following are excerpts from report of tests of a *Babcock & Wilcox* boiler, built for the *U. S. S. Cincinnati*, by a board composed of *Lieutenant-Commander* (now *Admiral*) *A. B. Willits* and *Lieutenant* (now *Admiral*) *B. C. Bryan, U. S. N.*

Boiler Data. Kind of boiler, Babcock & Wilcox—"Alert" type. Diameter of top drum, 42 in., inside. Length of top drum, 12 ft. Tubes: total No. 565; length, 8 ft. (525, 2 in. O. D., and 40, 4 in. O. D.). Grate surface: length, 6 ft. 8¼ in.; width, 9 ft. 5½ in.; area, 63.25 sq. ft. Grate surface reduced in tests Nos. 5 and 6, to 5 ft. 6 in.; 52 sq. ft. area. Heating surface: area, 2640 sq. ft.; ratio to grate, 41.74:1. Per cent water-heating surface, 100. Grate bars: kind, fixed. Smoke pipe: area, 7.876 sq. ft.; height, 48 ft.

above grate; ratio to grate, 1 : 8.03. Weight of boiler and all fittings except up-takes and smoke pipe:

Without water, lbs.	53,304
Water, 5 in. in glass; steam at 215 lbs., lbs.	9,498
Total with water, lbs.	62,802
Total weight per sq. ft. of grate surface, lbs.	992.9
Total weight per sq. ft. of heating surface, lbs.	23.79

The boiler was erected in a wooden structure built especially for the test and having the following dimensions: Length, 29 ft. 2 in.; width, 17 ft. 2½ in.; height, 21 ft. This was made as nearly airtight as possible, but contained several windows that could be opened or closed to regulate the amount of draft pressure. The blower was driven by belting from the main shop engines and ran continually.

An air heater was built in the up-take by means of which the waste gases imparted heat to the air on its passage to the ash pit. This heater could be placed in and out of service at will by the use of a by-pass flue.

Description and Object of Tests. Seven tests were made in all. Six of these consisted of three pairs, in which the tests of each pair were made under similar conditions in every way except that of using the air heater, one being with and the other being without this heater, in order to define the economy due to its use. The last or seventh test was for maximum capacity, and was made without the air heater and with the full grate. Two pairs of tests, one at a consumption of about 20 lbs. of coal and the other at about 35 lbs. of coal per sq. ft. of grate per hr., were made with the full grate surface in use. These tests will be found in tables of results numbered 1, 2-*H*, 3-*H*, 4, the letter *H* signifying that the air heater was in use during the tests. The grate surface was then reduced to 52 sq. ft. by a course and a half of bricks, seven courses in height, at the back of the furnace, and tests Nos. 5 and 6-*H* were made, burning about 50 lbs. of coal per sq. ft. of grate per hr. The bricks were then removed from the furnace and test No. 7 was made, burning nearly 60 lbs. of coal per sq. ft. of grate per hr. The data and results of these tests will be found in Table 38 on pp. 508 and 509.

Coal and Firing. The fuel used was Pocahontas, Flat Top, coal. It contained considerable slate and clinkered badly. On tests Nos. 1 and 2-*H* run-of-mine coal was used; on tests Nos. 3-*H*, 4, 5 and 6-*H* the coal was screened, using a screen with a 1-in. mesh. On test No. 7 the screenings from the former tests were run over ⅝-in. mesh screen, and the coal thus screened was mixed with the screened coal used in other tests. The firing was good and very regular. Two alternate doors were fired in rapid succession. The other two sections of fires, in wake of the other two doors, were sliced through the slicing door, and then leveled with a hoe, and then coaled, the average time between coalings of the same two furnaces being from 8 to 10 min. The furnace doors were open about 25 sec. when coaling and about 10 sec. in leveling. The coal made comparatively little smoke except when firing or working fires. The data in regard to smoke was taken by using Ringelmann charts.

Description of Apparatus. The water was weighed in two tanks, each supported on a platform scales and run into a third tank below, from which the feed pumps drew water. All pipes were above ground and in plain sight, and wherever connected to other piping or boilers, plugs were left out of *T* connections to show that there was no leakage. The gross and tare weights of each tank were taken, and the temperature was taken at the lower

tank just as each upper tank drained into it. The feed water was heated by steam injection before entering the weighing tanks.

The coal was weighed in barrows on platform scales in the fireroom and dumped on the floor. The time was taken when each lot of barrows were fired.

A sample shovelful of coal was taken from each lot of barrows and thrown into a barrel, and from this, mixed and quartered, the final samples for analyses, calorimeter and moisture determinations were taken. The gases for analyses were drawn from near the centre of the base of smoke pipe by means of a pipe inserted therein connected with an inspirator and a small Orsat instrument.

All draft pressures were taken outside the building, pipes being led there from the different places where pressure determinations were required.

Temperatures were taken at the back and front of the up-take just above the heater; in front by a mercurial pyrometer, and at the back by a metallic pyrometer. When the air heater was used the temperature was taken in addition just below the heater by means of a mercurial pyrometer.

The moisture in the steam was determined by a Barrus universal calorimeter. The steam was found practically dry in all cases. The steam was partly used in the shop and partly blown off into the atmosphere, the pressure being controlled by regulating a small stop valve by hand.

Experiments to show the heat of the gases at various points were made by noting the points at which different metals melted.

Before making test No. 6-*H*, on June 21st, all water was drained from the boiler and the contents of boiler noted for each 1-in. mark of the water gage glass, with the following results:

Weight of Water Contained in Boiler

Temperature of Water, 72 Deg. F.

Height of water in gage, in.	Total water, lbs.	Difference	Height of water in gage, in.	Total water lbs.	Difference
0	9,312	...	5	10,368	231
1	9,498	186	6	10,672	304
2	9,662	164	7	10,943	271
3	9,912	250	8	11,175	232
4	10,137	225			

Fires were started in the boilers with light wood, and blower in use, at 9 : 40 A.M. Temperature of water in boiler, 72 deg.; height in gage glass, 1 in.

For record of raising steam during this test see p. 513.

An examination of the boiler after this test showed no injury or change in its condition in any respect.

In addition to the tests made for the Navy Department, three tests were made for *The Babcock & Wilcox Company* by *E. H. Peabody*, Mem. A. S. M. E. The data and results of these tests are included with the others in the following table:

Table 38.

Engineers conducting tests	Lieut.-Com. A. B. Willits,		
Number of tests	1	2-*H*	3-*H*
Date of tests, 1900	June 15	June 16	June 18
Duration of test, hours	8	8	6
Kind of start	Alternate	Alternate	Alternate
State of weather	Clear	Clear	Clear
Heating surface	2,640	2,640	2,640
Grate surface	63.25	63.25	63.25
Ratio of heating surface to grate surface	41.74	41.74	41.74
Average Pressures			
Barometer, inches	....	30.18	30.12
Steam pressure by gage, pounds	209.3	206.6	210.6
Force of draft at base of pipe, inches of water	−0.10	−0.36	+0.09
Force of draft in furnace, inches of water	+0.10	−0.09	+0.81
Force of draft in ash pit, inches of water	+0.25	+0.11	+1.42
Force of draft in closed fire-room	+0.25	+0.33	+1.78
Average Temperatures, Deg. F.			
External air	....	84.7	80.2
Fire-room	101.5	99.5	98.2
Air entering heater	....	93.4	94.1
Air leaving heater	....	206.2	235.9
Feed water entering boiler	107.5	106.2	103
Air entering ash pit	101.5	206.2	235.9
Escaping gases from boiler	466	493	608.6
Escaping gases from heater	....	458	569
Fuel			
Kind, Pocahontas coal	Run of mine	Run of mine	Screened
Weight of coal as fired, lbs.	10,345	10,600	12,925
Moisture in coal, per cent	4.06	3.62	1.6
Weight of dry coal consumed, lbs.	9,925	10,201	12,718
Weight of ash and refuse, lbs.	996	879	1,088
Weight of combustible consumed, lbs.	8,929	9,322	11,630
Refuse in dry coal, per cent	10.03	8.60	8.55
Fuel per Hour, lbs.			
Dry coal consumed per hour	1,240.6	1,277	2,119.7
Dry coal consumed per hr. per sq. ft. G.S.	19.61	20.19	33.51
Dry coal per hr. per sq. ft. H.S.	0.470	0.484	0.803

"Cincinnati" Tests

U.S.N., and Lieut. B. C. Bryan, U.S.N.				E. H. Peabody		
4	5	6-*H*	7	8	9	10-*H*
June 19	June 20	June 21	June 22	June 14	June 23	June 25
6	4	4	3½	4	4	4
Alternate Clear	Alternate Clear	Alternate Clear	Alternate Clear	Alternate Clear	Alternate Clear	Alternate Clear
2,640	2,640	2,640	2,640	2,640	2,640	2,640
63.25	52.0	52.0	63.25	63.25	63.25	63.25
41.74	50.8	50.8	41.74	41.74	41.74	41.74
30.22	30.20	30.09	30.02			
207.4	213.6	209.8	210.8	200.0	188.0	191
+0.13	−0.34	−0.37	−1.48	−1.99	−0.56	−0.76
+0.95	+0.99	+0.65	+0.54	−0.55	−0.16	−0.46
+1.37	+1.75	+1.45	+1.69	0.00	0.00	−0.32
+1.44	+1.84	+2.08	+1.78	0.00	0.00	0.00
75.8	80	85	90.7			
101.8	109.3	115.4	117.0	114	94	
....	....	106.1	....	....	....	143
....	....	259.5	....	....	....	237
101.8	103.3	102.0	99.7	109.3	107.4	104.3
101.8	109.3	259.5	117	114	94	237
570.6	640.0	674	Less than 900 deg. F	661	569	546
....	....	637	....	....	....	471
Screened	Screened	Screened	Twice screened	Run of mine	Twice screened	Twice screened
13,517	10,680	10,046	12,600	9,800	8,630	5,150
1.5	1.13	1.0	0.88	3.0	0.75	0.75
13,314	10,480	9946	21,489	9,506	8,565	5,110
1,268	1,177	949	1,055	897	702	467
12,046	9,303	9,097	11,434	8,609	7,863	4,643
9.52	11.23	9.54	8.45	9.44	8.20	9.13
2,219	2,620	2,486.5	3,747	2,376.5	1,427.5	1,277.5
35.08	50.38	47.81	59.23	37.54	22.55	20.18
0.840	0.992	0.942	1.419	0.900	0.540	0.484

Table 38. "Cincinnati"

Engineers conducting tests	Lieut.-Com. A. B. Willits,		
Number of tests	1	2-*H*	3-*H*
Quality of Steam			
Per cent of moisture in steam	0	0	0
Degrees of superheating	0	0	0
Quality of steam (dry steam = 100)	100	100	100
Water			
Total weight of water feed to boiler, lbs.*	93,928	100,783	11,3943
Water actually evaporated, corrected for quality of steam, lbs	93,928	100,783	11,3943
Factor of evaporation	1.1657	1.1667	1.1705
Equivalent water evaporated into dry steam F. & A. 212 deg., lbs.	109,492	117,583	133,370
Water per Hour, lbs.			
Water evaporated per hour, corrected for quality of steam	11,741	12,598	18,974
Equivalent evaporation F. and A. 212 deg.	13,686	14,698	22,228
Actual evaporation per sq. ft. G.S.	185.7	199.2	300.0
Equivalent evaporation F. and A. 212 deg. per sq. ft. G.S.	216.3	232.4	351.4
Same per sq. ft. of H.S.	5.18	5.57	8.42
Economic Results			
Water evaporated under actual conditions per lb. of dry coal, lbs.	9.47	9.88	8.96
Equivalent evaporation F. and A. 212 deg. per lb. of dry coal, lbs.	11.03	11.51	10.49
Equivalent evaporation F. and A. 212 deg. per lb. of combustible, lbs.	12.19	12.70	11.47
Efficiency			
British thermal units per lb. of dry coal.	14,616	14,956	14,344
Efficiency of boiler, dry coal basis	73.1	74.3	70.6

*Corrected for inequality of water level and steam pressure at beginning and end of test.

REMARKS AND OBSERVATIONS.—In tests marked *H* the air heater was in use. Tests 8, 9 and 10-*H*, blower not running; stack draft increased by steam jet.

Principal data taken every fifteen and thirty minutes.

Percentage of smoke as observed, using Ringelmann smoke charts: Test 1, No. 0 to No. 1 and as high as No. 4 when firing; test 2-*H*, No. 0 to No. 1, with puffs of No. 4 when firing; test 3-*H*, No. 1 to No. 2, with puffs of No. 4 when firing; test 4, No. 2 to No. 3, with puffs of No. 4 when firing; test 5, No. 1 to No. 3, with puffs of No. 4 when firing;

Tests (*Continued*)

U.S.N., and Lieut. B. C. Bryan, U.S.N.				E. H. Peabody		
4	5	6-*H*	7	8	9	10-*H*
0	0.25	0.16	0	0	0	0
0						
100	99.75	99.84	100	100	100	100
118,330	91,066	86,488	102,545	85,661	82,590	49,060
118,330	90,838	86,350	102,545	85,661	82,590	49,060
1.171	1.1704	1.1715	1.173	1.1628	1.1633	1.1668
138,586	106,317	101,159	120,285	99,598	96,077	57,243
19,722	22,709	21,622	30,763.5	21,420	13,765	12,265
23,098	26,579	25,290	36,086	24,900	16,013	14,311
311.8	436.7	415.7	486.3	338.3	217.4	193.8
365.2	511.1	486.3	570.5	393.4	253.0	226.1
8.75	10.07	9.58	13.67	9.43	6.47	5.42
8.88	8.67	8.68	8.21	9.01	9.64	9.60
10.41	10.14	10.17	9.63	10.48	11.22	11.20
11.50	11.43	11.12	10.52	11.45	12.22	12.32
14,363	14,560	14,253	14,685	14,600	14,800	14,800
70.1	67.5	69.0	63.4	69.4	73.2	73.1

test 6, about No. 2, with puffs of No. 4 when firing; test 7, average about No. 3, with puffs to No. 5 when firing.

Kind of firing: Spreading and slicing.

Average thickness of fires: Tests 1 and 2-*H*, 6 inches; test 3-*H*, 7 inches; tests 4, 5, 6 and 7, 6 inches.

Average intervals between firings for each furnace during time fires were in normal condition: 8 to 10 minutes.

Average interval between times of breaking up: Eight to ten minutes.

Efficiency of firemen: Expert.

Table 39. Analysis of Waste Gases Made During Test of U. S. S. "Cincinnati" Boiler

Date	Time	Condition of fire when sample was taken	CO_2	O	CO	Lbs. dry gas per lb. carbon
1900	4.58	.....	15.2	3.3	1.0	
	5.15	.....	14.3	3.0	2.0	
	5.30	Just before firing	13.0	6.5	0.0	
June 15	5.55	One minute after firing	12.5	6.7	0.8	
	6.16	Just after raking	14.3	3.7	1.0	16.8
	6.27	Two minutes after firing	12.7	6.6	0.7	
	7.05	Three minutes after raking and just before firing	16.0	2.0	2.0	
		Average	14.0	4.5	1.1	
	11.45	.....	13.4	6.4	0.0	
June 16	12.50	Just after firing	12.0	5.0	1.0	
	1.50	Just after slicing	12.0	6.6	0.2	19.1
	3.50	Just after slicing	13.2	4.8	0.7	
		Average	12.7	5.7	0.5	
	11.25	One-half minute after firing	12.3	3.4	2.7	
	12.40	While slicing	14.2	4.0	0.1	
June 18	12.50	Just after slicing	12.5	4.3	1.2	17.3
	12.58	Just before slicing	13.0	4.0	3.4	
	1.03	One minute after firing	13.5	5.4	0.2	
		Average	13.1	4.2	1.5	
	10.10	While slicing	15.0	3.2	1.2	
	10.25	While slicing (all samples except 11 o' clock collected through ¼-in. iron pipe)	13.8	5.2	0.6	
June 19	10.28	Just after raking	14.4	3.1	0.9	
	10.35	One minute before firing	13.2	5.6	0.4	18.8
	11.00	While slicing (sample collected through glass tube)	13.0	5.6	0.6	
	2.20	Just after raking	10.2	8.3	0.5	
	2.40	Just after firing	10.2	9.0	0.3	
		Average	12.8	5.7	0.6	
	10.25	While slicing	13.5	5.7	0.0	
	11.00	While slicing	11.2	8.4	0.3	
June 20	11.04	Just after firing	10.4	8.1	0.5	
	11.13	Two minutes before raking	9.2	9.9	0.0	20.6
	12.25	Just after raking	12.1	5.4	0.7	
	12.36	Just after raking	14.2	4.0	0.8	
		Average	11.8	6.9	0.4	
	11.00	Just after raking	15.7	4.6	0.1	
	11.03	One minute before raking	13.0	6.0	0.0	
June 21	11.13	Just after firing	15.4	3.0	0.6	
	11.50	Just after raking	13.6	5.6	0.1	17.7
	11.55	One minute before raking	13.0	5.3	0.4	
	11.59	Just before raking	16.0	4.2	0.0	
		Average	14.5	4.8	0.2	
	11.21	Two minutes before firing	14.3	4.2	1.1	
	11.45	Two minutes before firing	11.0	9.0	0.0	
	12.38	Just before leveling and firing				
June 23	2.26	Just after firing	11.8	7.9	0.4	18.6
	2.30	Just after firing	13.3	4.2	1.0	
	2.43	Two minutes before firing	14.2	3.8	1.0	
		Average	12.9	5.8	0.7	
	10.16	Just before firing	15.3	4.1	1.0	
	10.21	One minute before firing	13.0	6.0	1.0	
June 25	11.10	Just after leveling	13.7	6.6	0.3	18.5
	11.13	Just after firing	14.0	5.2	0.8	
	11.47	Just after firing	9.0	11.2	0.3	
		Average	13.0	6.6	0.7	

The following is a record of the time required to raise steam to 215 lbs. press. from cold water:

Record of Raising Steam

Time		Steam press., lbs.	Time		Steam press., lbs.
By watch	Elapsed		By watch	Elapsed	
9 : 40	.....	Fire started	9 : 51	11 mins. 0 secs.	125
9 : 45	5 mins. 0 secs.	Steam formed	9 : 51 : 10	11 mins. 10 secs.	135
9 : 46 : 30	6 mins. 30 secs.	25	9 : 51 : 15	11 mins. 15 secs.	145
9 : 47 : 30	7 mins. 30 secs.	35	9 : 51 : 30	11 mins. 30 secs.	155
9 : 48	8 mins. 0 secs.	45	9 : 51 : 40	11 mins. 40 secs.	165
9 : 48 : 30	8 mins. 30 secs.	55	9 : 51 : 55	11 mins. 55 secs.	175
9 : 49	9 mins. 0 secs.	65	9 : 52 : 10	12 mins. 10 secs.	185
9 : 49 : 30	9 mins. 30 secs.	75	9 : 52 : 20	12 mins. 20 secs.	195
9 : 50	10 mins. 0 secs.	85	9 : 52 : 30	12 mins. 30 secs.	205
9 : 50 : 30	10 mins. 30 secs.	95	9 : 52 : 40	12 mins. 40 secs.	215
9 : 50 : 45	10 mins. 45 secs.	115			

Effect of Superheat on the Economy of a Steam Engine—Steam Yacht "Idalia"

The summary of engine tests (Table 40) made on the Steam Yacht Idalia, having a four-cylinder triple-expansion engine with cylinders 11½ × 19

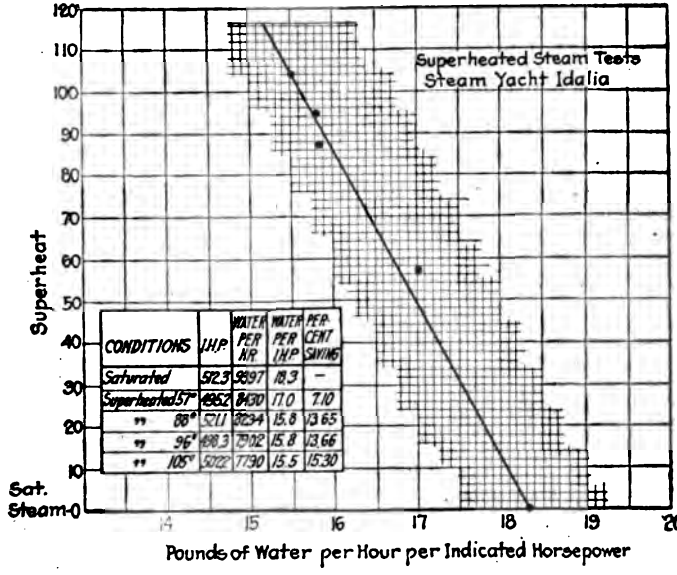

Superheated Steam Tests
Steam Yacht Idalia
Superheat
Sat. Steam
Pounds of Water per Hour per Indicated Horsepower
CONDITIONS | I.H.P. | WATER PER HR. | WATER PER I.H.P. | PER CENT SAVING
Saturated | 512.3 | 9897 | 18.3 | –
Superheated 57° | 495.2 | 8430 | 17.0 | 7.10
" 88° | 521.1 | 8234 | 15.8 | 13.65
" 96° | 498.3 | 7902 | 15.8 | 13.66
" 105° | 502.2 | 7790 | 15.5 | 15.30

FIG. 89.

× 22 1/16 × 22 1/16 × 18 in. stroke, is taken from the report of *Lieutenant (now Captain) John Halligan, Jr., U. S. N.*, as printed in the Jour. A.S.N.E.,

November, 1909. The steam was supplied by a Babcock & Wilcox boiler and superheater, and arrangements were made whereby the engine was furnished with saturated steam and with superheated steam of four different amounts of superheat. Fig. 89 shows graphically the saving in engine economy by the use of superheated steam.

BOILER MATERIAL. INSPECTION REQUIREMENTS. U. S. SUPERVISING INSPECTORS

Drawings of Boilers. The manufacturer of any boiler to be used for marine purposes shall furnish the inspectors of the district where such boiler or boilers are to be inspected duplicate blue prints or tracings fully descriptive of same in detail for their approval, one of which shall be kept on file in the office of the local inspectors and the other returned to the manufacturer. Where more than one boiler is made from a similar design, a drawing of which is on file in the local inspectors' office, if made at a different date, a reference to such drawing on file is all that shall be required. The manufacturer shall also furnish the inspectors an affidavit on either Form 935 or 936, as contained in sections 8 and 9 of Rule I. (See p. 517.)

It shall be the duty of every local inspector of boilers to make, for every new boiler inspected in his district, all computations required by the rules and regulations, and any other necessary computations, from data obtained from blue prints or tracings, boiler manufacturers' affidavits, tensile test reports, and from other reliable sources. A record of such computations in full shall be made on letter sheets and filed with blue prints or tracings of boiler, the first sheet of such computations to be headed with a general description of boiler and with the vessel file number.

Name Plates. There shall be fastened to each boiler a plate containing the name of the manufacturer of the material, the place where manufactured, the tensile strength, the name of the builder of the boiler, when and where built.

Table 40. Summary of Tests, "Idalia"

Date, 1909	Conditions	Pressures			Vacuum	Temperatures		R.P.M.			I.H.P., main engine	Water per hour, total	Water per I.H.P.	Per cent saving of steam
		Throttle	First receiver	Second receiver		Feed	Hotwell	Air pump	Circulating pump	Main engine				
Oct. 11	Saturated	190	68.4	9.7	25.5	201	116.0	57	196	194.3	512.3	9,397	18.3	7.10
Oct. 14	Superheat, 57 deg.	196	66.0	9.2	25.9	206	109.5	56	198	191.5	495.2	8,450	17.0	13.66
Oct. 14	Superheat, 88 deg.	201	64.3	8.7	25.9	205	115.0	53	196	193.1	521.1	8,234	15.8	13.66
Oct. 12	Superheat, 96 deg.	198	61.9	7.8	25.4	202	111.5	54	198	191.5	498.3	7,902	15.8	13.66
Oct. 13	Superheat, 105 deg.	205	63.0	8.4	25.2	200	111.0	45	197	193.1	502.2	7,790	15.5	15.30

The date of the building of the boiler or boilers shall be determined by the month and year of issue of the first certificate of inspection which covers the boiler or boilers in question: *Provided*, That the boiler or boilers have not been used for any purpose previous to the inspection.

Stamps. Every iron or steel plate intended for the construction or repairs of boilers to be used on steam vessels shall be stamped by the manufacturer in the following manner, at two diagonal corners, at a distance of about 18 inches from the edges.

Plates to be used in externally fired boilers the thickness of which shall not exceed one-half inch for steamers navigated under the provisions of Title LII, Revised Statutes, which will be subject to tensile strain in said boilers, shall be tested and inspected by an inspector duly authorized under the provisions of said title, and such plates shall not be stamped until they have been tested by the inspector, and each of such plates shall then be stamped by the manufacturer in the presence of the inspector with the name of the manufacturer, the place where manufactured, and the minimum number of thousand pounds tensile stress it will bear to the sectional square inch as has been determined by the test.

Plates to be used in the construction or repair of boilers other than the externally fired boilers as described in above paragraph, for steamers navigated under provisions of Title LII, Revised Statutes, which shall be subject to tensile strain in said boilers shall be tested and inspected by an inspector duly authorized under provisions of said title and such plates shall be stamped by the manufacturer before same are tested with the name of the manufacturer, the place where manufactured, and the minimum number of pounds the plate will bear to the sectional square inch, expressed in thousands.

Inspectors testing iron or steel plate at the mills where same is manufactured shall allow the mills to shear any plate into butt straps or heads before the coupon from the plate is tested, and they shall accept the statement of the manufacturer that the coupon presented to the inspector for testing was taken from the plate from which the butt straps or heads were cut, and if the coupon meets all the requirements of Rule I, the inspector shall stamp all heads and butt straps cut from the plate.

All plates which conform to the physical, chemical, and other requirements prescribed by these rules shall be stamped by the inspector near the manufacturer's stamp, with the official stamp of the United States Steamboat-Inspection Service, and with the initials of his name and a serial number.

Plates may be tested and inspected at the mills for repairs to marine boilers or to be carried in stock, the report of such test to be in duplicate, one copy to be furnished through the supervising inspector to the local inspectors in the district where the purchaser of such material is located, and the other to the purchaser, who shall deliver a copy of the same to the parties using the material, who, in turn, shall submit the same to the local inspectors in the district where the material is to be used, before being assembled in the boiler. Steamers carrying such repair material to be used in emergencies shall carry the record of each sheet of such material on board.

Boilers built since February 28, 1872, of material stamped and tested according to the requirements of section 4430, Revised Statutes, and having a record thereof in the office of the local inspectors in the district where the boiler was built or intended to be used, may be used for marine purposes, notwithstanding that such boilers may have been used for other purposes, if in the judgment of the local inspectors they are deemed safe.

Chemical Properties, Steel Plates. Steel plates shall be made by the open-hearth process, except that steel for plates to be used in the manufacture of boiler tubes may be made by the Bessemer process.

Open-hearth steel shall contain not more than 0.04 per cent of phosphorus nor more than 0.05 per cent of sulphur

The manufacturer shall furnish the inspector, with each order tested, a certificate stating the process by which the steel was manufactured and a copy of the analysis of each melt. The analysis may, if deemed expedient by the Supervising Inspector General, be verified at the expense of the manufacturer.

Physical Properties, Steel Plates. All steel plates tested shall show an elongation of at least 20 per cent measured in a gage length of 8 inches.

Physical Properties, Iron Plates. The tensile strength shall be not less than 45,000 pounds per square inch. The elongation shall be not less than 15 per cent The reduction of area shall be not less than 15 per cent for 45,000 pounds tensile strength, and for each increase of 1000 pounds tensile strength up to 55,000 pounds, an addition of 1 shall be made to the required percentage of reduction of area. The bend test specimen shall bend cold through 90 deg. around a curve the radius of which is not greater than one and one-half times the thickness of the specimen.

FIG. 90.—Tension Test Specimen.

All tension test specimens shall be milled to the form as shown in Fig 90 with the following dimensions: Length at least 16 inches, ends from 1½ to 3½ inches wide by about 3 inches in length, and parallel section at centre 1 to 1½ inches wide by 9 inches in length. The percentage of elongation shall be measured in a gage length of 8 inches.

Where specimens are to be tested on the testing machines of the Steamboat-Inspection Service, they shall be 1 inch wide at parallel section in centre, and shall not exceed 2 inches in width on the ends.

All bend test specimens shall be at least 12 inches in length and from 1 to 3½ inches in width, and the full thickness as rolled. The edges may be planed. The corners shall not be rounded, but the sharpness may be removed with a fine file. After bending, the specimens shall show no cracks or flaws on the outside of the bent portion.

Bend test specimens for steel plates, before bending, shall be heated to a cherry red as seen in the dark, and quenched in water the temperature of which is about 82 deg. F.

Two tension tests and one quench bend test shall be made from each plate as first rolled from the billet, slab, or ingot, the tensile test specimens to be taken from the diagonal corners of the plate, and the quench bend specimen to be taken from that part of the plate which represents the top of the billet, slab, or ingot.

The quench bend specimen shall withstand, without fracture, being bent over until the ends are parallel and the inner radius equal to one and one-half times the thickness of the test specimen.

The finished material shall be free from all injurious defects, and shall have a good and workmanlike finish.

All measurements of test specimens and material shall be made by any standard American gage, and record of tests shall be submitted on Form 934.

[Form 935]

Affidavit of Manufacturer of Marine Steam Boilers Constructed of Material Tested at the Mills

DEPARTMENT OF COMMERCE,
STEAMBOAT-INSPECTION SERVICE.

STATE OF ———, *County of* ———.

On this ——day of ———, A. D. 19—, personally appeared before me, ——— ———, a notary public in and for the county of ——— and State of ———, Mr. ——— ———, who deposes and says that he is ——— ———, of ——— ———, boiler manufacturer, and has contracted to build— marine boiler— for ——— ———, of ——— ———, from ——— plate manufactured by ——— ———, of ——— ———, which plate was tested at the mills by a United States assistant inspector, as provided in the act of Congress approved January 22, 1894, each of said plates having stamped thereon the words "U. S. Assistant Inspector" and the initials — —, and numbered as follows: ———.

No plate for shell or other part of boiler subject to tensile strain, other than herein specified, will be used in the construction of said boiler–, the lowest tensile strength stamped thereon being ——— pounds.

The dimensions of the boiler– will be: Length, ———; diameter, ———.

	Material	Number	Length	Thickness	Diameter
Tubes					
Flues					
Furnaces					

Kind of furnaces, ——; round, ——; corrugated, ——; flat sides, ——. Thickness of plates of cylindrical shell of boiler, ——; thickness of side sheets in flat side of furnace, ——; thickness of flat top sheet of back connection, ——; thickness of plates of cylindrical shell of back connection, ——; thickness of material of boiler heads, ——; thickness of tube sheets, ——; thickness of plates of shell of steam chimney, ——; thickness of plates in lining of steam chimney, ——; thickness f side sheets, ——. Kind of rivets (iron or steel), ——; diameter of rivet holes, ——; pitch of rivets, ——. All rivet holes of boiler–, including steam and mud drums, will be fairly drilled and no part of such holes will be punched, ———; or punched, ———. All tubes used in the construction of said boiler– are of the thickness and material required by, and have met all the other requirements of, the rules of the Board of Supervising Inspectors, as shown by statement of the manufacturer of the tubes. Steam pressure for which boiler —— to be inspected, —— pounds. Style of boiler, ——. Boiler– to be installed upon the steamer ———.

Signature: ——— ———.

Subscribed and ——— to before me this —— day of ———, 19—.
(Sworn or affirmed.)

[NOTARY'S SEAL.]

——— ———,
Notary Public.

NOTE.—Inspectors will not accept this affidavit without the data required, unless accompanied by a satisfactory explanation in writing, to be filed with the affidavit.

[Form 936]

Affidavit of Manufacturer of Marine Steam Boilers

DEPARTMENT OF COMMERCE,
STEAMBOAT-INSPECTION SERVICE.

STATE OF ———, *County of* ———.

On this —— day of ———, A. D. 19—, personally appeared before me, ———

———, a notary public in and for the county of ——— and State of ———, Mr. ——— ———, who deposes and says that he is ——— ———, of ——— ———, boiler manufacturer, and that the accompanying samples of ———, manufactured by ——— ———, of ——— ———, were cut from plates which are to be used in the construction of —— marine boiler– for ———, of ———, and no plate for shell or other part of boiler subject to tensile strain of less tensile strength or quality than that determined by the United States inspector who made the tests, which lowest tensile strength found was ——— pounds, will be used in the construction of said boiler–, the dimensions of which will be: Length, ———; diameter, ———.

	Material	Number	Length	Thickness	Diameter
Tubes.....	.....	.....	.....	.....	.....
Flues.....	.....	.....	.....	.....	.....
Furnaces.....	.....	.....	.....	.....	.....

Kind of furnaces, ——; round, ——; corrugated, ——; flat sides, ——. Thickness of plates of cylindrical shell of boiler, ——; thickness of side sheets in flat side of furnace, ——; thickness of flat top sheet of back connection, ——; thickness of plates of cylindrical shell of back connection, ——; thickness of material of boiler heads, ——; thickness of tube sheets, ——; thickness of plates of shell of steam chimney, ——; thickness of plates in lining of steam chimney, ——; thickness of side sheets, ——. Kind of rivets (iron or steel), ——; diameter of rivet holes, ——; pitch of rivets, ——. All rivet holes of boiler—, including steam and mud drums, will be fairly drilled and no part of such holes will be punched, ———; or punched, ———. All tubes used in the construction of said boiler– are of the thickness and material required by, and have met all the other requirements of, the rules of the Board of Supervising Inspectors, as shown by statement of the manufacturer of the tubes. Steam pressure for which boiler —— to be inspected, —— pounds. Style of boiler, ——. Boiler to be installed upon the steamer ———.

Signature: ——— ———

Subscribed and ——— to before me this —— day of ———, 19—.
(Sworn or affirmed.)

[NOTARY'S SEAL.]

——— ———,
Notary Public.

NOTE.—Inspectors will not accept this affidavit unless the data required are given, unless accompanied by a satisfactory explanation in writing, to be filed with the affidavit.

Inspectors may make requisition on the department for the necessary supply of blank affidavits for the use of boiler manufacturers.

Foreign-built Boilers. Boilers of foreign-built vessels admitted to American registry shall be inspected in the same manner as boilers on American vessels.

Furnaces. The tensile strength of steel used in the construction of corrugated or ribbed furnaces shall not exceed 67,000 and be not less than 54,000 lbs.; and in all other furnaces the minimum tensile strength shall be not less than 58,000 and the maximum not more than 67,000 lbs. The minimum elongation in 8 inches shall be 20 per cent.

Lapwelded Boiler Tubes up to and Including 4 Inches in Diam.

All lapwelded tubes shall be made of charcoal iron, or mild steel, made by any process.

Surface Inspection. Tubes shall be free from defective welds, cracks, blisters, scale, pits, and sand marks.

Tests. The following tests shall be made before shipment by the manufacturer:

(*a*) A test piece 2 in. in length cut from a tube shall stand being flattened by hammering until the sides are brought parallel and separated by a distance not greater than one-third the outside diam. of the tube, without showing cracks or flaws, the weld being 45 deg. from the bend.

(*b*) A second tube shall have a flange turned over at right angles to the body of the tube and shall have a width equal to $\frac{3}{8}$ of an in. Tubes less than 3 in. in diam. shall only be required to have a flange turned equal to $\frac{1}{8}$ of the diam. of the tube. This test shall not be applied to threaded stay tubes.

All the work shall be done cold.

Each tube shall stand an internal hydrostatic pressure of 1000 lbs. per sq. in. and shall be struck near both ends while under pressure with a 2-lb. hammer or its equivalent without showing signs of weakness or defects.

All steel tubes, except those made of open-hearth steel, shall have the ends properly annealed by the manufacturer before shipment.

All steel tubes shall stand expanding, flanging over on the tube plate, and beading without flaws, cracks, or opening at the weld; except threaded stay tubes.

Lapwelded Boiler Tubes over 4 Inches up to and Including 30 Inches in Diameter

All lapwelded boiler tubes over 4 in. in diam., up to and including 30 in. in diam., shall be made of wrought iron or mild steel, made by any process.

(*a*) A test piece, 2 in. in length, cut from a tube shall stand being flattened by hammering until the sides are brought parallel and separated by a distance not greater than one-third the outside diam. of the tube, without showing cracks or flaws, with the weld being 45 deg. from the bend.

Each tube shall stand an internal hydrostatic pressure of 500 lbs. per square inch and shall be struck near both ends while under pressure with a 2-lb. hammer or its equivalent without showing signs of weakness or defects.

All steel tubes except those made of open-hearth steel shall have ends properly annealed by the manufacturer before shipment.

All steel tubes shall stand drilling, riveting, and calking, and work necessary to install them into the tube head without showing any weakness or defects.

No tube increased in thickness by welding one tube inside of another shall be allowed for use, but the ends of boiler tubes may be welded on for the purpose of making repairs or new tubes may be welded for the purpose of making them longer.

Seamless Steel Boiler Tubes

Material. The steel shall be made by the open-hearth process.

Surface Inspection. Tubes shall be free from all surface defects. The defects to be particularly avoided in seamless tubes are tears, snakes, checks, slivers, scratches, laps, pits, rings, and sinks.

All seamless cold-drawn tubes shall be annealed as a final process. One or more tubes shall be selected at random from each charge of annealing furnace, and coupons cut from same for testing.

(*a*) A piece 3 in. long cut from the first tube shall stand being flattened by hammering until the sides are brought parallel and separated by a distance

not greater than one-third the outside diam. of the tube, without showing cracks or flaws.

(*b*) A flange shall be turned around the end of the tube to a width equal all to $\frac{3}{8}$ of an in. beyond the outside body of the tube. Tubes less than 3 in. in diam. shall only be required to have a flange turned equal to $\frac{1}{8}$ of the diam. of the tube. This test shall not be applied to threaded stay tubes.

Tests (*a*) and (*b*) shall be done cold.

Where hot-finished tubes are furnished, the tubes shall pass the same manipulating tests as cold-drawn tubes and shall be subject to the same conditions as to gage, but do not have to be annealed.

Each tube shall be subject to an internal hydrostatic pressure of 1000 lb. per sq. in. without showing signs of weakness or defects.

All tubes, except threaded stay tubes, shall stand expanding, flanging over on the tube plate, and beading without flaw or crack.

All tubes shall be carefully gaged and shall not be less than the specified thickness.

Statement or Certificate of Manufacturer of Boiler Tubes. When the manufacturers of boiler tubes file with the Supervising Inspector General a certificate duly sworn to that all tubes manufactured by them will stand the tests and meet all the requirements of the foregoing rule, they shall be accepted by inspectors of this service and no other affidavit will be required.

Tests of Bars for Stays and Braces

All steel bars to be used as stays or braces in marine boilers and allowed a stress of 7000, 8000, or 9000 lbs. per sq. in. of section, tested by the United States inspectors at the mills where the material is manufactured, shall be tested in the following manner: There shall be taken from each heat two pieces for tensile tests and two pieces for bending tests. The full-size bars within the capacity of the testing machine may be used for tensile tests. Where the full size of the bar is too large for the capacity of the testing machine, the bar may be reduced in size to meet such capacity. To facilitate and insure accurate tests, all bars for tensile and bending tests may be reduced in size. The minimum tensile strength of each test piece shall be not less than 58,000 lbs. per sq. in. of section and each test piece that has been reduced in size shall show an elongation of at least 28 per cent in 2 in. Where the full size of the bar has been used for testing, the test piece shall show an elongation of at least 25 per cent in 8 in. When the tensile strength of the test piece is more than 63,000 lbs. per sq. in. of section, each test piece that has been reduced in size shall show an elongation of at least 26 per cent in 2 in. Where the full size of the bar has been used for testing, each test piece shall show an elongation of at least 22 per cent in 8 in. The pieces for the bend test shall be bent cold to a curve, the inner radius of which is equal to $1\frac{1}{2}$ times the diam. of the bar without flaws or cracks. Should any such test bar fail in either the tensile or bending test, no bars from such heat shall be allowed to be used in the construction of any marine boiler. Where a heat of steel bars has been passed by an inspector, separate lots of bars from such heat may be furnished to different boiler manufacturers upon a certificate from the mill that the bars were made from such accepted heat.

Boiler manufacturers desiring to use tested steel stays or braces shall be

required to furnish the inspectors with the following form of affidavit, duly filled in:

[Form 937]

STATE OF ———, *County of* ———, *ss:*

Personally appeared before me, a notary public for and in the county of ——— and State of ———, Mr. ——— ———, who, being first duly sworn, deposes and says that he is the ——— of the steam boiler works situated at ———, and known as the ———, and that the lot or lots of steel bars from which the test bars were taken and tested by the inspector on the —— day of ———, 19—, and allowed for use in the steam boiler– to be constructed for the steamer ———, and to be allowed a strain not to exceed —— pounds per square inch of section as a working steam pressure, will be used in the construction of the boiler– for the steamer ———, and no material for any braces, stays, or stay bolts required to carry a strain equal to ——— pounds per square inch of section will be used as braces, stays, or stay bolts in the construction of the boiler– for the said steamer unless tested by the inspector and approved by him in accordance with the requirements of law.

——— ———,

——— ———.

Sworn to and subscribed before me this —— day of ——— 19—.

[NOTARY'S SEAL.] ——— ———,

Notary Public.

SECTION 5

TURBINES

AND

MECHANICAL REDUCTION GEARS

BY

A. G. CHRISTIE, M. E., University of Toronto; Assoc. Professor Mechanical Engineering, The Johns Hopkins University; Mem. A.S.M.E., Amer. Gas Assoc.

J. A. DAVIES, Ass't Gen. Supt., Essington Wks., Westinghouse Electric & Machine Co.; Mem. Soc. Nav. Arch., & Mar. Eng. and A.S.N.E.

CONTENTS

TURBINES

By A. G. CHRISTIE	Page
Properties of Steam	528
Adiabatic Expansion	529
Actual Expansion	533
Nozzles	538
Impulse Blading	543
Curtis Blading	546
Parsons Blading	548
Clearances and Leakage	555
Rotational Losses	561
Provisional Proportions of Turbine:	
Wheel Efficiencies	563
The Limiting Factors of Capacity	563
Provisional Proportions of an Impulse Turbine	564
Provisional Proportions of a Parsons Turbine	566
Gaging	570
Provisional Proportions of Direct-coupled Parsons Marine	572
Provisional Proportions of Astern Turbines	572
Provisional Proportions of Curtis Rateau Turbines	572
Turbine Design:	
Design of a Simple Impulse Turbine	573
Design of a Simple Curtis Turbine	575
Design of a Rateau Turbine	577
Design of a Curtis-Rateau Turbine	580
Design of a Parsons Turbine	581
Design of a Curtis-Parsons Turbine	588
Miscellaneous Thermodynamic Considerations	589
Martin's "New Theory of the Steam Turbine"	590
Mechanical Design:	
Design of Turbine Discs	593
Critical Speed of Rotors	605
Design of Rotor Shafts	609
Drum Rotors	609
Shaft Couplings	609
Blading	611
Cylinder Construction	624
Turbine Drains	628
Dummy Pistons, Glands, Labyrinths	631
Casing Glands	633
Bearings	635
Balancing of Rotors	640
Miscellaneous Mechanical Equipment	646
Micrometer Gear	649
Bridge Gages	650
Governors and Valve Gears	653
Lubrication System:	
Bearings	661

	Page
Lubricating Oils	662
Types of Lubricating Systems	663
Oil Pumps	667
Strainers	669
Oil Filters	671
Oil Coolers	677
Tanks	678
Oil Piping	679
Piping Connections:	
Steam Lines	679
Traps and Drains	681
Gland Piping	682
Foundations	683
Lifting Gear	687
Mechanical Reduction Gearing	692
Electrical Gearing	692
U. S. S. Jupiter	693
U. S. S. New Mexico	694
U. S. S. Tennessee	701
S. S. Mjölner	703
S. S. Wulsty Castle	704
Hydraulic Gearing	707
Performance:	
Effects of Variation in Operating Conditions	710
Turbine Testing	712
Test Data and Trials	715
Trial Performance of U. S. S. Wadsworth	720
Electric Drive Performance	720
Performance of Small Steam Turbines	722
Comparison of Marine Steam Turbines and Other Prime Movers	723
Types of American Steam Turbines	732
Westinghouse Marine Turbines	732
General Electric Turbines	745
Delaval Steam Turbines	755
Kerr Turbines	762
Poole Marine Turbines	767
Direct Drive Marine Propelling Turbines	774
The Ljungstrom Turbine	779
Sturtevant Turbines	784
Terry Steam Turbines	788
Engine Room Layouts	794
MECHANICAL REDUCTION GEARS	
By J. A. DAVIES	
Types	806
Pinion	808
Gear Wheels	813
Testing	815
Care and Operation	815

STEAM TURBINES

BY

A. G. CHRISTIE

REFERENCES: Goudie, "Steam Turbines," Longmans, Green & Co.; Martin, "Steam Turbines," Longmans, Green & Co.; Sothern, "The Marine Steam Turbine," Crosby, Lockwood & Sons; Meyer, "Steam Turbines," U. S. Naval Institute; Stodola, "Die Dampfturbinen," Springer; Thomas, "Steam Turbines," Wiley; Moyer, "Steam Turbines," Wiley; Peabody, "The Steam Turbine," Wiley; Journal, American Society of Naval Engineers; Engineering, London.

SYMBOLS

V_0 = theoretical steam velocity, ft. per sec.
V_1 = absolute velocity from nozzle, ft. per sec.
V_2 = relative velocity entering blades, ft. per sec.
V_3 = relative velocity leaving blades, ft. per sec.
V_4 = absolute velocity leaving blades, ft. per sec.
U = blade speed, ft. per sec.
N = R.P.M.
d = mean diameter in in.
D = mean diameter in ft.
α = nozzle angle or angle of Parsons blades.
β = entering angle of blades.
γ = leaving angle of blades.
g = 32.2.
R = ratio of blade speed to steam speed.
F = reheat factor.
$(H_1 - H_2)$ = heat drop.
E_s = stage efficiency.
v_s = specific volume.
s = ratio of clear area to annular area.
w = lbs. of steam per H.P.
W = lbs. of steam per sec.
l = blade length in in.
r = ratio.
$\left.\begin{matrix} r \\ R \end{matrix}\right\}$ = also used for radius in in. and in ft.
t = thickness in in.
F = centrifugal force.
m = area.
ν = Poissan's ratio.
R_s = radial stress.
T = tangential stress.

A steam turbine is a prime mover in which a portion of the heat in the steam is transformed by a drop in pressure in suitably designed passages into kinetic energy in the issuing jet which in turn is converted into mechanical work on impinging or reacting upon suitably designed moving blades.

Simple Impulse Type. When the turbine contains a single set of expanding elements or nozzles (*a*) in which the complete pressure drop takes place and where the resulting jet acts on the blades of a single revolving disk (*b*) as in Fig. 1, the turbine is known as a "simple impulse," or generally as the **DeLaval type.** It is characterized by a decrease of relative velocity of the steam in crossing the blade and also by the fact that the pressure on the two sides of the blade is the same. The methods of designing nozzles and blades which will produce as small a shock as possible on the steam when it enters the blades are shown on p. 543.

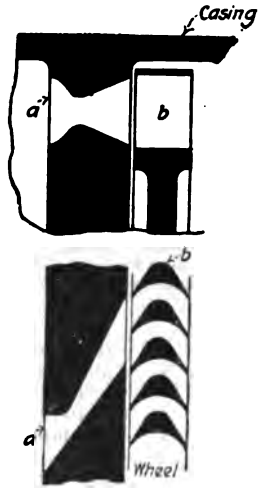


FIG. 1.—Simple Impulse or DeLaval Type.

Velocity Stage Type. Sometimes the velocity of the steam from the nozzle is so great relatively to the blade speed, that the velocity leaving the blades is also relatively high. If this is allowed to pass away unused the efficiency of the turbine would be very low. A portion of this may be recovered by inserting stationary blades (*c*) which reverse the direction of the steam issuing from the first row of blades (*b*) and direct it against a second row of blades (*d*), as shown in Fig. 2*a*. This construction is known as a "velocity stage" or generally as a "**Curtis Stage.**" The earlier Curtis turbines of the General Electric Co. employed this construction. Fig. 2*b* shows an extension of this type.

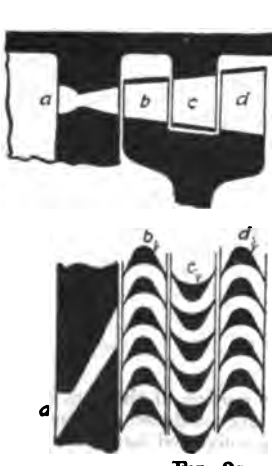


FIG. 2*a*.

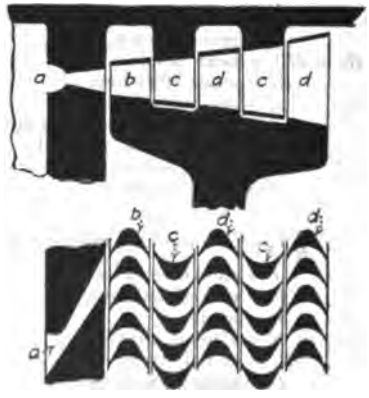


FIG. 2*b*.

FIG. 2.—Velocity or Curtis Stage.

Multiple Impulse Type. If, however, the velocity of the jet is so chosen that the transformation of energy into work in crossing the blade approaches a maximum, then it will be generally found that the total pressure drop has not been utilised. Further work can be done by a second or third, or even

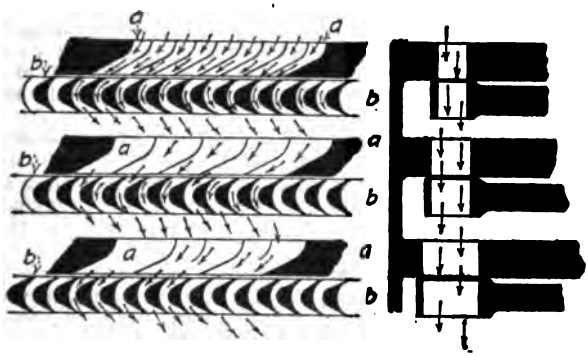


FIG. 3.—Multiple Impulse or Rateau Type.

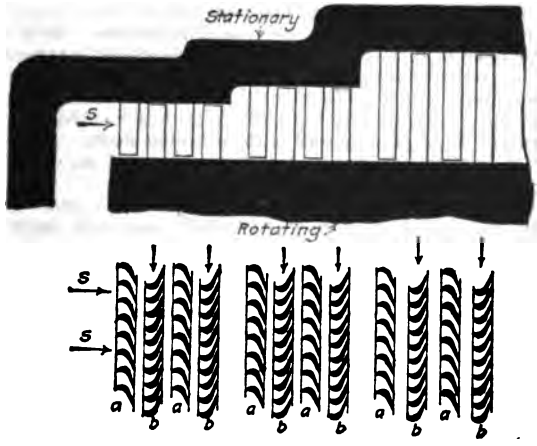


FIG. 4.—Impulse-reaction or Parsons Blading.

more, pressure drops in succeeding sets of nozzles and blade wheels. In other words, the steam may be made to pass through a number of impulse wheels in series, as shown in Fig. 3, where (*a*), (*a*) and (*a*) are the nozzles and (*b*), (*b*) and (*b*) the revolving disks carrying the blades. This type of turbine is known as the "Multiple Impulse" or generally as the **Rateau type.**

Impulse-reaction Type. The drop in pressure and consequent expansion of the steam occurs wholly in the nozzles in each of the preceding types of

turbines. There is still another class of turbines in which there is a pressure drop in passing through the moving blades, as well as through the stationary or nozzle blades, as shown in Fig. 4 in which (*a*) and (*a*) are stationary blades or nozzles and (*b*) and (*b*) are the revolving blades. The pressure drop in (*a*) produces an increase in kinetic energy which is absorbed on impinging on (*b*) as in the impulse turbine. But the pressure drop in the revolving blade (*b*) produces an increase in the velocity of the steam at exit and a consequent reaction on the blade with a large component in the direction of rotation. This turbine is thus an "impulse-reaction" turbine, generally called the **Parsons type.**

Each of these types has certain advantages over other types for specific purposes. In practice, therefore, many combination turbines are used, such as Curtis-Parsons, Curtis-Rateau, while almost all marine turbines use Curtis wheels for the reversing section of the unit, on account of their compactness and low weight for large power capacity. Complete designs of all these types will be shown later.

Three different classes of marine turbines are now being built and used. The first class consists of turbines of slow rotative speed **direct connected** to the propeller shaft. In order to get satisfactory blade and steam speeds, the disks and drums of these turbines must be of large diameter resulting in very heavy turbines. Usually a compromise between turbine and propeller speeds is assumed. Neither turbine or propeller then operates at its maximum efficiency. The steam consumption is consequently high.

The second class consists of turbines of high rotative speed **connected through reduction gearing** to slow speed propellers. Both turbine and propeller can operate at their best speed and hence at their maximum economy. The gear losses are small. This combination is quite economical in both steam consumption and weight.

The third class consists of turbines of high rotative speed **driving turbo-generators** with slow speed motors on the propeller shaft. This combination offers certain distinct advantages for some ships and its use is increasing. It also is quite economical.

The two latter classes have very distinct advantages over the first class, particularly in regard to economy and space, and with improvements in reduction gears and electric drive will gradually supersede the first class. Practically all auxiliaries may be driven by high-speed turbines sometimes using reduction gears.

The following notes therefore refer more specifically to the last two classes, though they may also be applied to the first class.

PROPERTIES OF STEAM

The properties of steam are given in full on pp. 209 to 217 in this book. Computations on steam turbines involve frequent reference to these tables and use of the heat entropy diagram which is shown on pp. 214 and 215.

Use of the Steam Tables. All pressures are absolute. Hence, barometric pressures must be added to all stated gage pressures for turbines before referring to tables.

Pressures below atmosphere are also stated in steam tables in pounds absolute. Such pressures are usually read as **vacuum** in inches of mercury. The absolute pressure can be found approximately by subtracting the vacuum from the barometric pressure and multiplying the difference by 0.490. Thus vacuum 27.9 in.; barometric 29.7 in.:

Abs. press. = (29.7 − 27.9) × 0.490 = 0.88216 lb. per sq. in. When

accurate tests are being made the temperatures of the mercury columns must be read and a correction made for this factor. Mercury expands 0.0001 in. per in. of length per deg. One inch of mercury at 32 deg. F. is equivalent to 0.4912 lb. per sq. in. **Temperatures** are always in deg. F. **Superheat** may be indicated by the number of degrees superheat as 100 deg. F. at 165 lbs. absolute pressure, or may be indicated as the total temperature, thus, 165 lbs. abs., 466 deg. F. temperature.

All **heat quantities** are given in B.t.u., *i.e.*, $\frac{1}{966}$ of the heat required to raise 1 lb. of water from 32 deg. to 212 deg. F.

Heat of the liquid is the heat required to raise the liquid from 32 deg. F. to the boiling point.

Latent heat is the heat required to vaporise the liquid once it has been raised to the boiling point.

The **total heat** is the sum of the latent heat and the heat of the liquid. In the case of superheat it also includes the heat required to superheat the steam after it has been evaporated.

Specific volume of the steam is needed for many steam turbine calculations and is the volume in cu. ft. per lb. of steam.

The calculations and diagrams on the following pages are based on Marks and Davis Steam Tables.

ADIABATIC EXPANSION

A vapor or gas is said to expand adiabatically when all the work is done at the expense of its internal heat energy. During such an expansion the substance receives no heat from any outside source nor does it give up heat to any outside object except in the form of work. It is therefore evident that such expansion produces the maximum amount of work from an expanding gas for a given expenditure of heat. Adiabatic expansion is also characterised by a drop in temperature. This fact is **utilized in turbine design** by expanding the steam nearly adiabatically to a low pressure in a nozzle and thus lowering the temperature of the steam before passing into the turbine casing itself. This lessens the liability to trouble from distortion under heat.

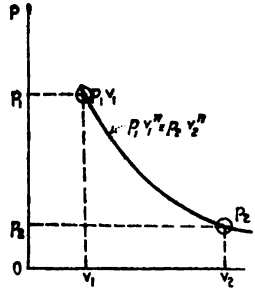


FIG. 5.—Adiabatic Expansion on the PV Diagram.

On the **pressure-volume (PV) diagram** such an expansion may be represented by a curve $P_1V_1^n = \text{const.} = P_2V_2^n$ as in Fig. 5 where the exponent n has values for adiabatic expansion differing for various substances. For value of n for water vapor see p. 223.

It is more convenient to deal with the adiabatic expansion of steam on the **temperature-entropy** or **heat-entropy diagrams.** The heat-entropy diagram is shown on pp. 214 and 215, where the coördinates are total heat and entropy. The lines of constant pressure, constant quality and constant superheat, are shown on the diagram. Specific volumes may be obtained from steam tables. The total entropy of the steam remains constant during adiabatic expansion from the initial pressure and conditions to the final pressure. Hence on these diagrams adiabatic expansion may be represented by a line of constant entropy drawn from the initial condition point to the

final pressure line, as shown in Fig. 6, which is a temperature-entrophy diagram, and Fig. 7, which is a portion of the total heat-entropy diagram.

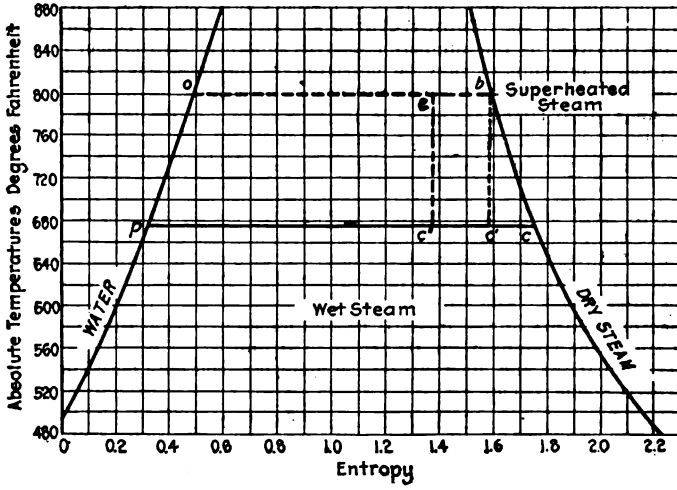


FIG. 6.—Adiabatic Expansion of Steam on Temperature-entropy Diagram.

Practically all thermodynamic calculations with turbines are based on such expansions as shown in Fig. 7. From this diagram it is possible to determine the total heat contents of the steam at any point in the expansion. As the expansion proceeds with decreasing pressure the total heat obviously decreases. The difference between the initial total heat and that at any point in the expansion is called the **heat drop** and with adiabatic expansion this heat must be turned into work. The quickest way to determine such heat drop is to take it from a heat-entropy diagram as shown on p. 214.

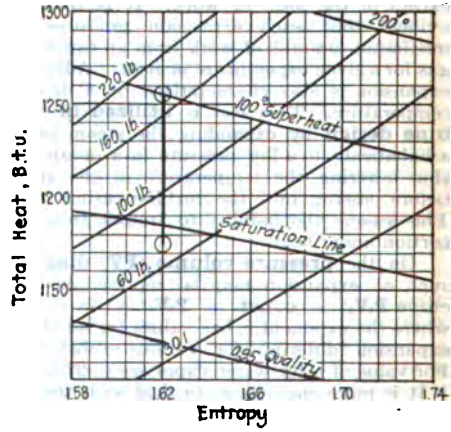


FIG. 7.—Adiabatic Expansion on the Heat-entropy Diagram.

Heat-entropy diagrams accompany Marks & Davis' Steam Tables which have been used in this book. In turbine design, Ellenwood's "Steam Charts" have proven very useful and will be used for the solution of problems. These charts include values of specific volumes which

are most convenient. Goodenough's "Properties of Steam and Ammonia" contains one of the best heat-entropy charts that have been published and this diagram is now being very widely used by engineers.

Example. Steam expands adiabatically from 180 lbs. abs. 100 deg. superheat, to 70 lbs. abs. What heat is available for work?

From diagram; Fig. 7, total heat at 180 lbs. and 100 deg., $H_1 = 1254$ B.t.u.

Following line of constant entropy on chart from condition representing 180 lbs. and 100 deg. superheat, it cuts the 70 lb. line at a total heat of $H_2 = 1171.6$, therefore **heat available** for work.

$$H_1 - H_2 = 1254 - 1171.6 = 82.4 \text{ B.t.u.}$$

The **quality** at the end of expansion is 0.991 and the specific volume 6.14 cu. ft. per lb.

Steam Volume Calculations. It is necessary to know the **volume of the steam at any condition** in order to calculate the areas of passages in steam turbines. This information can also be deduced from the heat-entropy diagram. The quality or superheat can be read off from the curves and the specific volume calculated from the steam tables. This method is somewhat cumbersome and tedious and may be greatly simplified by the use of Ellenwood's "Steam Charts," already referred to, where total heat and specific volume are the coördinates and constant entropy, constant pressure, etc., are represented by series of curved lines. These charts permit more rapid work than the usual heat-entropy diagrams. **To calculate this volume from** the steam tables, as is sometimes necessary, proceed as follows:

The initial total heat and total entropy are found from the steam tables for the given initial conditions.

Taking the preceding problem, the steam tables give total heat $H_1 = 1254.3$ B.t.u., total entropy 1.6201. At 70 lbs. the entropy of the water = 0.4411. Therefore entropy of evaporation to condition at end of expansion = 1.6201 − 0.4411 = 1.1790. At final pressure, entropy of complete evaporation = 1.1896. Therefore quality at end of expansion = $\frac{1.1790}{1.1896} = 0.991$ and total heat at end of expansion $H_2 = 272.6 + 0.991 \times 907.2 = 1171.6$.

$\therefore H_1 - H_2 = 1254.3 - 1171.6 = 82.7$ B.t.u. The specific volume of saturated steam at low pressure is 6.2; therefore, specific volume at final conditions $= 0.991 \times 6.2 = 6.155$ cu. ft.

A very convenient diagram for finding approximate steam volumes is shown on a chart in Goudie's "Steam Turbines" and reproduced to small scale in Fig. 8.

Relation of Heat Energy and Work. 1 B.t.u. = 777.54 ft. lbs. 1 horsepower = 33,000 ft. lbs. per min. = 1,980,000 ft. lbs. per hr. = $\frac{1,980,000}{777.54}$ or 2546.5 B.t.u. per hr.

The **"heat available"** for work per pound of steam $(H_1 - H_2)$ can be determined by the preceding methods. The **ideal water rate** of a turbine in lbs. per hr. (W) can be calculated from this heat drop as follows:

$$W = \frac{2546.5}{H_1 - H_2}$$

When steam flows from a high to a lower pressure, as in a steam nozzle, assuming there is no friction losses, the heat available for work is expended in increasing the velocity and the kinetic energy of the jet. This increase in kinetic energy is equivalent to the heat drop. Thus $777.6\ (H_1 - H_2) = \frac{V^2}{2g}$,

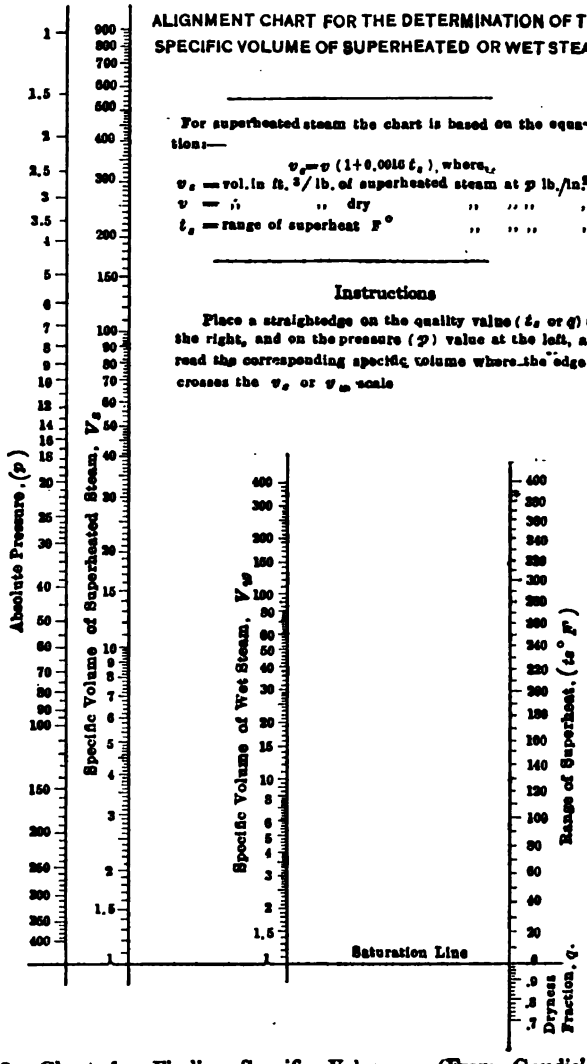

ALIGNMENT CHART FOR THE DETERMINATION OF THE SPECIFIC VOLUME OF SUPERHEATED OR WET STEAM
For superheated steam the chart is based on the equation:—
$v_s = v\,(1+0.0016\,t_s)$, where,
v_s = vol. in ft.^{3}/lb. of superheated steam at p lb./in.^{2} abs.
v = " " dry " " " "
t_s = range of superheat F°
Instructions
Place a straightedge on the quality value (t_s or q) at the right, and on the pressure (p) value at the left, and read the corresponding specific volume where the edge crosses the v_s or v_w scale
Absolute Pressure, (p)
Specific Volume of Superheated Steam, V_s
Specific Volume of Wet Steam, V_w
Saturation Line
Range of Superheat, (t_s ° F.)
Dryness Fraction, q.

FIG. 8.—Chart for Finding Specific Volumes. (From Goudie's "Steam Turbines.")

where $(H_1 - H_2)$ = the heat drop as before, V = the resultant increase in velocity from the expenditure of this heat drop, $g = 32.2$.

$$\therefore V = \sqrt{2g(H_1 - H_2) \times 777.6} = 223.7\sqrt{(H_1 - H_2)}$$

Fig. 9 shows the velocities resulting from given heat drops.

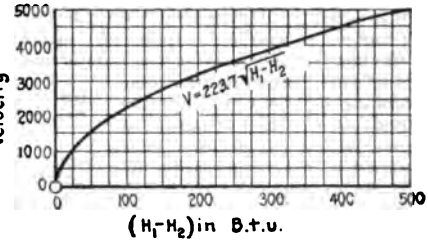


FIG. 9.—Relation Between Steam Velocity and Heat Available.

ACTUAL EXPANSION

Only theoretical expansion is considered in the previous paragraphs. In actual machines this is altered by several factors among which are throttling, leakage, radiation and friction.

Throttling. Some throttling usually occurs in a steam turbine, particularly at the throttle valve and in governing valves when such are used. The steam does no useful work when throttled from the initial to the final pressures. The total heat content of the steam will therefore be the same neglecting radiation at the end of such an expansion as at the beginning.

Throttling can best be **represented on a heat-entropy diagram** as in Fig. 10. Here it is assumed that the steam first expands adiabatically along the line *AB* and the heat drop is converted into kinetic energy. Later on the steam is practically brought to rest at the lower pressure and with the velocity reduced to zero the kinetic energy is reconverted into heat along the constant pressure line *BC*. The total heat at *C* is the same as at *A*. Throttling can therefore be considered in one step and as "**a change of pressure at constant total heat.**" The specific volume and entropy increase, but the temperature decreases with such expansion.

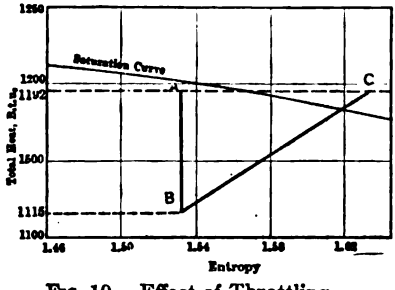


FIG. 10.—Effect of Throttling.

Leakage occurs over the tips of blades in Parsons turbines and through labyrinth packings between blade wheels in impulse turbines. **The effect of leakage** is practically the same as that of throttling, for the steam passes from the higher to the lower pressure without doing useful work. Leakage may therefore be considered as a constant heat change just as in Fig. 10.

Radiation decreases the amount of total heat present at the beginning of each expansion in a turbine. However in modern turbines, particularly those of moderate and large size, where the cylinder is well lagged, the radiation loss is so small as compared with other losses less definitely known that it may be neglected in the calculations. Radiation losses seldom exceed 0.5 per cent of the total heat.

Friction. When steam flows across a surface at any velocity, heat is produced by friction which, in the case of a jet, is at the expense of its kinetic energy. Since such heat cannot be radiated to the outside beyond the normal radiation loss, it is returned to the steam and thus increases its heat content. The final kinetic energy of the steam is decreased by this conversion of a portion into heat. This is shown on a heat-entropy diagram in Fig. 11, where *AB* represents adiabatic expansion in a nozzle. However, if there is friction in the nozzle itself some of the kinetic energy is converted into heat so that at the lower pressure the steam contains the heat units represented at *C* rather than at *B*. Hence actual expansion may be assumed to take place along line *AC*. In crossing the blades the further amount of heat represented by *CD* is returned by friction to the steam from the kinetic energy.

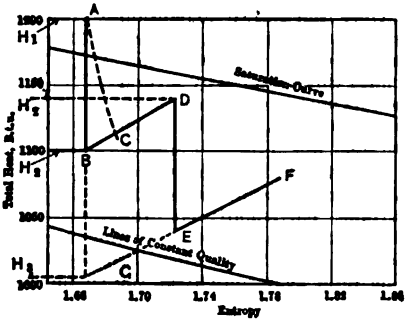


FIG. 11.

$(H_1 - H_2)$ is heat drop *AB*, but $(H'_2 - H_2)$ is equivalent friction loss *BD*. Hence energy from jet doing actual work is $(H_1 - H'_2)$ B.t.u.

Friction **produces some important effects.** It increases the total heat of the steam at the end of expansion and hence increases the dryness, the total entropy and the specific volume; larger passage ways must be provided for the flowing jet. On the other hand, it decreases the kinetic energy available for work.

It is well known that if a jet of steam is allowed to form an eddy, its velocity will be dissipated by the friction of the particles upon themselves. Steam passages must therefore be so designed that **eddies** do not form. Still another form of friction is that due to whirling disks or drums in an atmosphere of steam. (See p. 561.)

Efficiency Ratio. Referring to Fig. 11, it will be noted that after allowing for all losses due to leakage, friction, windage, etc., amounting to $(H'_2 - H_2)$ the net heat available to produce work is $(H_1 - H'_2)$. The ratio of this net heat to the total heat drop or $\frac{H_1 - H'_2}{H_1 - H_2}$ is called the **"stage efficiency."** The efficiency ratio of the whole turbine may be defined as the heat actually converted to work in the turbine to the heat available from adiabatic expansion from initial conditions to exhaust pressure.

The efficiency ratio is the best measure of a turbine's efficiency for it expresses the degree to which the actual machine approaches the ideal as represented by pure adiabatic expansion. Every engineer should therefore be familiar with the methods of calculating the efficiency ratio.

Care must be taken to note the basis on which efficiency ratio is expressed. The heat converted to work per lb. of steam is the heat equivalent to one horsepower divided by the lbs. of steam per H.P.-hr., *i.e.*, $\frac{2546.5}{W}$, where W is the pounds of steam per H.P.-hr. Hence, if $(H_1 - H_2)$ is the heat available from

adiabatic expansion from initial conditions to final pressure, the **efficiency ratio** $= \frac{2546.5}{W(H_1 - H_2)}$. There are, however, several kinds of horsepower. In steam turbine work there is the brake horsepower, the shaft horsepower, which may be measured beyond the gears, and in electrical units the electrical horsepower. Where the generator efficiency is not known it is frequently necessary to express the efficiency ratio as "based on electrical horsepower." In steam engine work there is also the indicated horsepower, see p. 1403. Hence care must be taken to state the horsepower on which the efficiency ratio is figured; it is best to state this as "efficiency ratio based on ——horsepower."

Examples. A 3000-S.H.P. turbine with 200 lbs. per sq. in. gage press., 100 deg. F superheat, and 28 in. vacuum develops a S.H.P. on 11 lbs. of steam per hour. Find the efficiency ratio based on S.H.P.

Total heat at 214.7 lbs. per sq. in. abs., and 100 deg. F., $H_1 = 1259$
Total heat after adiabatic expansion to 28 in. vacuum, $H_2 = 896$

Heat available from adiabatic expansion, $H_1 - H_2 = 363$

Efficiency ratio based on S.H.P. $= \frac{2546.5}{W(H_1 - H_2)} = \frac{2546.5}{11 \times 363} = 63.7$ per cent.

The 50 kw. electric lighting set on the same ship takes dry steam at 125 lbs. per sq. in. gage, exhausts against 5 lbs. per sq. in. gage back pressure and uses 55 lbs. per kw.-hr. Find the efficiency ratio based on electrical H.P.

Total heat at 139.7 lbs. per sq. in. abs. dry steam, $H_1 = 1192$
Total heat after adiabatic expansion to 19.7 lbs. abs., $H_2 = 1048$

B.t.u. available from adiabatic expansion, $H_1 - H_2 = 144$
Lbs. of steam per electrical H.P.-hr. $= 55 \times 0.746 = 41.03 = W$

Efficiency ratio based on electrical H.P. $= \frac{2546.5}{W(H_1 - H_2)} = \frac{2546.5}{41.03 \times 144} = 43.1$ per cent.

Reheat Factor. Since the total heat and entropy are increased by the effect of friction as in Fig. 11, it can be shown that, if the steam were further expanded adiabatically from this final condition D at P_2 to E at a third pressure P_3, the amount of heat available is greater than if the original expansion from A to B had been carried on from B to the same pressure P_3 at G. Similar effects will be noted if further stages are added.

Let H_{r1}, H_{r2}, H_{r3}, etc., be the individual heats available from adiabatic expansion in each of the various stages considering reheating from friction as in Fig. 11. $\Sigma H_r = H_{r1} + H_{r2} + H_{r3}$, etc. Then $\Sigma H_r > (H_1 - H_2)$ where $(H_1 - H_2)$ is heat drop in a single expansion from initial conditions to final pressure.

The ratio $\frac{\Sigma H_r}{(H_1 - H_2)}$ is known as the "reheat factor."

Let H_{a1}, H_{a2}, H_{a3}, etc. = heat converted into work per stage after allowing for reheating and let $H_1 - H_2$ = heat drop from initial conditions to exhaust pressure in a single expansion. Total heat to work is $H_{a1} + H_{a2} + H_{a3}$, etc. $= \Sigma H_a$. **Then $\frac{\Sigma H_a}{H_1 - H_2}$ = efficiency ratio.**

Goudie in "Steam Turbines" gives the following formula for the "reheat factor" with dry steam with the usual vacua employed and for pressures from 100 to 200 lbs. per sq. in.

$$F = 1 + 0.1(1 - n_s)$$

where n_s is the stage efficiency of the turbine as defined in the following paragraph and F is the reheat factor. The value of F increases with the superheat.

Fig. 12 from Goudie's "Steam Turbines" gives curves of reheat factors for multi-stage turbines at pressures between 100 and 200 lbs. per sq. in. pressure and 0 deg. F. to 300 deg. F. superheat.

$$\text{The stage effy. ratio in multi-stage turbines} = \frac{\text{Overall effy. ratio}}{\text{reheat factor}}$$

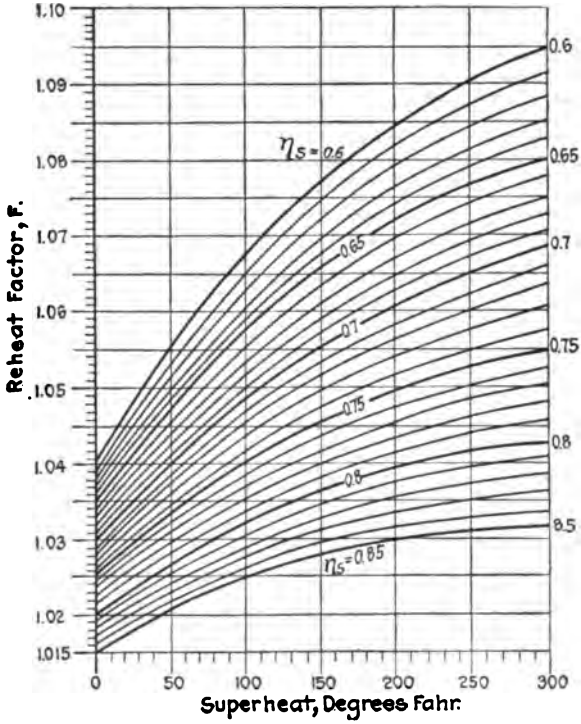


FIG. 12.—Relation Between Reheat Factors, Stage Efficiencies and Superheat. (From Goudie's "Steam Turbines.")

Condition Curve. Sometimes a line is drawn on the heat-entropy diagram connecting the points representing the initial conditions at the beginning of each stage and the final steam condition. This is known as a "condition curve." A condition curve for such expansion is shown in Fig. 13. This curve is of considerable use in the design of turbines but it is not easy or practicable to lay it out point by point on a heat-entropy diagram.

The condition curve is flat and, when the steam is initially superheated, can be drawn with sufficient accuracy when the initial and final conditions of the steam and the point where it crosses the saturation line are known. If the steam is initially dry or in a wet state a straight line joining the initial and final condition points forms the "condition curve."

Goudie suggests the following empirical formula for giving the slope of a straight line joining the initial point in the superheat region to the point where the condition curve crosses the saturation line.

$$\phi_s = 0.055 + 0.8f^2 - 0.00006\,(P_1 + t_{s1}).$$

where ϕ_s = entropy increase per 100 B.t.u. heat drop.
P_1 = initial pressure absolute in lbs. per sq. in.
t_{s1} = initial superheat at P_1 in deg. F.
$f = (1 - n_s)$ = stage friction factor.

To apply this formula calculate ϕ_s and lay off on heat-entropy diagram AD, Fig. 13, for 100 B.t.u. heat drop. Lay off $DE = \phi_s$, join AE and continue to cut the saturation curve at F. A smooth curve through A, F and C the final condition, gives the condition curve.

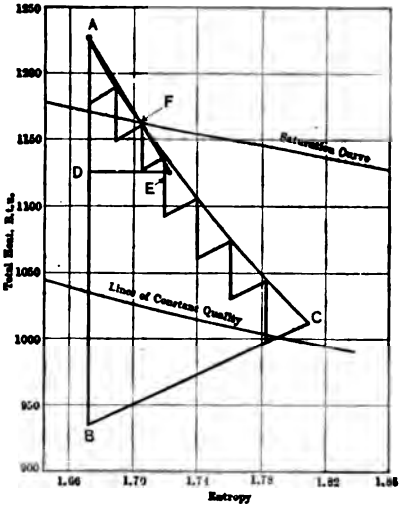


FIG. 13.—Condition Curve.

Several methods have been proposed to determine the pressures at the various stages of multi-stage turbines. One of the simplest is that of plotting a cumulative heat curve which allows for the reheat factor. The cumulative heat H_c at any pressure is equal to the adiabatic heat drop to that pressure divided by the stage efficiency of the turbine. The condition curve can now be plotted along with the cumulative heat curve in Fig. 47, with the logarithm of pressure as a base. The cumulative heat drop scale on the right hand side can be divided into as many sections as there are stages and horizontal lines drawn through these points to the cumulative heat curve give the required pressure at each stage.

Fig. 47 shows a 10-stage Rateau turbine operating between 200 lbs. gage, 100 deg. F. superheat and 28 in. vacuum with reheat factor 1.05 and stage efficiency of 0.68. The heat drop for the first stage turbine is 39 B.t.u. and 38 B.t.u. for all the remaining stages.

When the condition curve and the pressures are determined, the specific volume curve can be drawn. For convenience the lower section of the specific volume curve is drawn ten times scale. This is of great assistance in calculating nozzle heights and blade lengths in turbine design.

Some designers do not care about having equal work done in each stage and make no particular effort to correct for this reheat factor in their pressure distribution. They assume some fixed ratio of wheel speed to steam speed and maintain this ratio throughout the turbine. Toward the exhaust end there will be more heat available per stage due to reheating and hence the steam velocity increases. The wheel diameter must therefore be increased to maintain this constant ratio. Hence the diameters of the wheels constantly increase from the inlet end to the exhaust.

To Calculate Steam Consumption. If the initial condition of the steam entering a turbine and the final pressure together with the efficiency ratio are known then it is possible to find the heat converted to work per lb. of steam and from that the lbs. of steam per H.P.-hr. can be calculated.

Example. A 2500-H.P. turbine is supplied with steam at 200 lbs. gage, 100 deg. F. superheat and exhausts into 28 in. vacuum. What is the probable steam consumption?

Let efficiency ratio be 0.63. From heat-entropy diagram the heat available from adiabatic expansion from 214.7 lbs. abs., 100 deg. superheat to 2 in. abs. as in preceding problem:

$H_1 - H_2 = 363$ B.t.u.

Heat actually turned into work $= 363 \times 0.63 = 228.7$ B.t.u.

Lbs. of steam per H.P.-hr. $= \frac{2546.5}{228.7} = 11.13$ lbs.

The stage efficiency ratio in multi-stage turbines = Overall efficiency ratio / reheat factor.

Thus in the above problem from Fig. 12, the reheat factor at 0.63 overall efficiency ratio is 1.0605; therefore the stage efficiency ratio $= \frac{0.63}{1.0605} = 0.594$.

Fig. 14 is very convenient for estimating steam consumption when the heat available from adiabatic expansion is known.

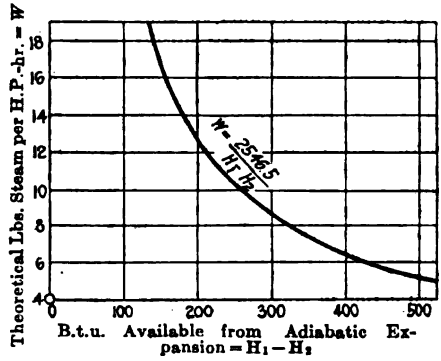


FIG. 14.—Relation of Available Heat and Theoretical Steam Consumption.

The efficiency ratio may also be found from the ratio of the theoretical water rate to the actual water rate.

$$\text{Efficiency ratio} = \frac{\text{theoret. water rate}}{\text{actual water rate}}.$$

Nozzles

Nozzles and orifices are used to convert the heat available from a pressure drop into kinetic energy by increasing the velocity of the steam jet. Let steam flow from pressure P_1 to pressure P_2 through a hole in plate A, Fig. 15, with well-rounded converging sides. The pressure decreases in passing through the hole and the velocity increases. The discharge becomes a maximum when P_2 at the throat becomes about $0.58 \times P_1$ with wet steam and $0.54 \times P_1$ with superheated steam, and does not increase with any further lowering of P_2. The value of P_2 ($= 0.58P_1$) is called the "**critical pressure**" and the velocity produced by the corresponding heat drop is the "**critical velocity.**" (See p. 233.)

In multi-stage turbines where the stage pressure does not exceed 58 per cent of that of the preceding stage, **simple converging** orifices, as shown in Fig. 15, are used to expand the steam.

The steam jet leaving either A or B issues from the orifice in straight lines and generally does not start to diverge until it has passed a short distance beyond the mouth, provided P_2 is the same pressure as P_2.

If, however, the pressure P_2 is decreased below P_2 ($= 0.58P_1$) the discharge of the orifice is not increased, the pressure in the throat still remains P_2 and expansion is completed outside the orifice. As the jet is no longer confined, expansion takes place in all directions. This results in whirls, eddies and

turbulent flow which causes a reconversion of kinetic energy into heat. A **divergent nozzle** added beyond the throat can have its areas so designed that the steam expansion is efficiently controlled, the direction of the jet is maintained in the proper direction and the discharge velocity will be the maxi-

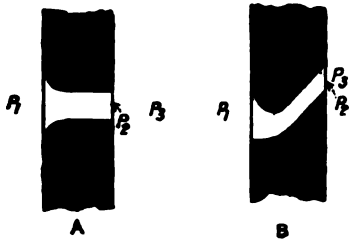


FIG. 15.

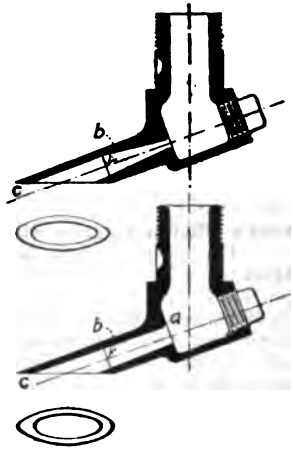


FIG. 16.—Types of Converging-diverging Nozzles.

mum for the complete heat drop. The flow will also be in a straight line. The nozzle then becomes a **converging-diverging nozzle** as shown in Fig. 16. The nozzle diverges from the throat (*a*) to the mouth (*b*) and is parallel to the axis of the nozzle from *b* to *c* to ensure straight-line flow on the blades.

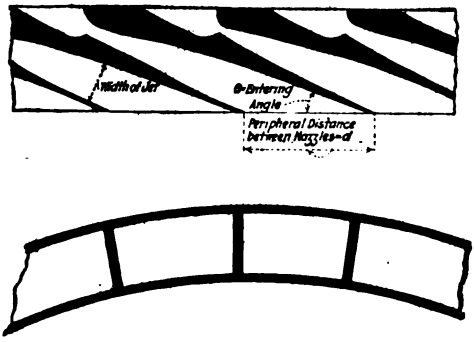


FIG. 17.—Nozzle Design in Curtis Turbines.

Some designers continue the taper from *b* to *c* instead of using the parallel section. This simplifies machining the nozzle. In some turbines these are rectangular in cross-section, as shown in Fig. 17, which allows easy machining.

The **velocity theoretically produced** in a nozzle in expanding from P_1 to P_2 is $V = 223.7\sqrt{H_1 - H_2}$. (See p. 533.) If the friction losses in the nozzle are expressed as y a ratio of the heat drop, then the **actual resultant velocity** of the steam leaving the nozzle is

$$V = 223.7\sqrt{(H_1 - H_2)(1 - y)}$$

Example. Let steam expand in a nozzle from 180 lbs. abs., 90 deg. superheat to 80 lbs. abs. y for nozzle = 6 per cent.

Heat available from adiabatic expansion from heat chart $(H_1 - H_2) = 1249 - 1178 = 71$ B.t.u.

$\therefore$ Velocity of issuing jet $= 223.7\sqrt{(1 - y)(H_1 - H_2)} = 223.7\sqrt{0.94 \times 71} = 1828$ ft. per sec.

It is usually not necessary to calculate more than the throat and mouth velocities since manufacturing considerations require a conical or other straight-sided passage connecting mouth and throat.

Nozzle efficiency varies with the size of nozzle and the condition of the steam. Dr. Stodola found the efficiency of **short convergent-divergent nozzles** to be 92 to 95 per cent with superheated steam. Other tests have shown 90 to 92 per cent efficiency, while with **low pressures and wet steam** 85 to 90 per cent efficiency has been noted.

Theoretically the **convergent orifice** has 100 per cent efficiency. But as usually constructed it consists of rectangular passages with considerable curved-wall surface as in B, Fig. 15. Losses occur due to certain compressive action in making the steam jet turn through this arc and to surface friction. In the absence of complete test data the efficiency of such orifices may be taken as 92 to 95 per cent **for superheated steam** and 90 per cent for **dry steam.** Lower values must be assumed for wet steam. The finish of the passage also affects efficiency; smooth highly polished surfaces give high efficiencies, rough-cast surfaces give low efficiencies.

Steam is said to be **"completely expanded"** when it issues from a nozzle at a pressure equal to that of the lower stage P_2. If, however, the exit area is made so small that the steam leaves the nozzle at a higher pressure than that in the stage into which it expands, it is said to be **"under-expanded."** This practice is frequently resorted to by designers and involves no serious consequences provided the degree of under-expansion is moderate. The expansion is completed as a rule with no spreading of the jet during its passage across the clearance space between the end of the nozzle and the blade. This is illustrated in Fig. 18.

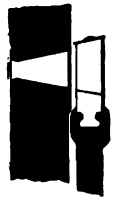


FIG. 18.

The inlet side of the blades is made long enough to receive the fully expanded jet. This practice improves efficiency at partial loads and when low or throttled steam pressures are used. If the nozzle were designed so that the steam was expanded to the lower-stage pressure before reaching the mouth of the nozzle, eddies, whirls and other disturbances would be set up and the efficiency of the nozzle would be decreased very rapidly. The steam is then said to be **"over-expanded."** While moderate under-expansion is desirable for turbines operating under varying loads and steam pressures, over-expansion is detrimental under all conditions and should be avoided.

The effect of under-expansion and over-expansion are presented in Tables 1 and 2, which were taken from curves in Goudie's "Steam Turbines."

Let A'_0 be the actual area of the mouth of the nozzle and A_0 be the correct area necessary for complete expansion. Then the amount of under- or over-expansion can be expressed by the ratio $\frac{A'_0}{A_0}$.

Also if V'_0 is the actual velocity striking the blades in the under- or over-expanded condition and V_0 the velocity that would have been developed in a properly designed nozzle or orifice, then $V'_0 = KV_0$ where K is velocity coefficient.

Table 1. Under-expansion

$\frac{A'_0}{A_0}$	K in per cent	Loss in velocity $(1 - K)$ in per cent	Additional energy loss $y' = (1 - K^2)$
0.95	99.92	0.08	0.20
0.90	99.70	0.30	0.60
0.85	99.30	0.70	1.35
0.80	98.80	1.20	2.30
0.75	98.23	1.77	3.40
0.70	97.55	2.45	5.00

Table 2. Over-expansion

$\frac{A'_0}{A}$	K in per cent	Loss in velocity $(1 - K)$ in per cent	Additional energy loss $y' = (1 - K^2)$
1.05	99.60	0.40	0.75
1.10	98.80	1.20	2.30
1.15	97.50	2.50	4.80
1.20	95.80	4.20	7.20
1.25	93.75	6.25	12.10

It will be at once noted that the losses due to over-expansion as shown in the above table are of far greater seriousness than those due to under-expansion. A small degree of under-expansion causes no serious loss at full load and full boiler pressure and, by preventing over-expansion when the boiler pressure drops slightly, on the whole it adds to rather than decreases the efficiency of the turbine considered over a range of practical operating conditions.

In a multi-stage turbine the steam entering each nozzle after the first nozzle usually does so with an initial velocity approaching the absolute velocity leaving the preceding blade. This energy is sometimes considered as available for increasing the heat drop in the succeeding nozzle, thus:

Heat available $= \frac{V_4^2}{2g} + (H_1 - H_2)$ when V_4 is the velocity of approach or absolute velocity leaving the preceding blade. Usually, however, it is assumed that the absolute velocity of steam from the blades is reconverted into heat and increases the total heat only of the steam entering the next nozzle. In other words H_1 is increased by the amount $\frac{V_4^2}{2g}$ and becomes H'_1. The heat available now becomes $(H'_1 - H'_2)$. This method will be used in the following notes, H'_1 being taken from the condition curves.

Length and Angle of Nozzles. The axis of a divergent nozzle usually makes an angle of 17½ to 22½ deg. with the axis of the blades on which it discharges. Diaphragm orifices usually are so designed as to have discharge angles of about the same amount, though these angles may be increased at the expense of efficiency to keep down the blade length in the last stages of a turbine. The **length of the diverging section** of a nozzle is given by the formula $\mathbf{L} = \frac{d_m - d_t}{2 \tan \theta/2}$, where L = length in inches, d_m = diameter at mouth,

d_t = diameter at throat and θ = included angle of diverging side. θ may vary considerably but usually lies between 12 deg. and 15 deg. It should not exceed 20 deg. because above this limit the expansion is too rapid and eddy disturbances cause a serious decrease in the efficiency of the nozzle.

Flow of Steam Through Orifices and Nozzles. The **weight of steam** passing through a nozzle is found from the equation:

$$W = \frac{AV}{144v_s}$$

where W = lbs. of steam per sec.
A = area of throat in sq. in.
V = velocity at throat in ft. per sec.
v_s = specific volume of steam at quality and pressure of throat.

The **area of the throat** can be determined for a given steam flow from the equation

$$A = \frac{144Wv_s}{V}$$

If, however, the orifice makes an angle with the axis of the blades, as in B, Fig. 16, then while the true throat area is ce, the projected throat area is cd and this must be calculated to find if the orifices can be all got in the annular ring. Let α be the angle the orifice makes with the axis of the blade disk.

$$\text{Projected area of throat } A' = \frac{144Wv_s}{V \sin \alpha}$$

Example. Twenty-four pounds of steam per sec. enter a certain low-pressure stage of a Rateau turbine at 10 lbs. abs. press., 18 deg. F. superheat and expand to 5.5 lbs. abs. press. γ for orifice = 5 per cent; angle of nozzle = 20 deg.

From chart, $H_1 - H_2 = 1151.5 - 1110 = 41.5$ B.t.u.; therefore $V = 223.7\sqrt{0.95 \times 41.5} = 1405$ ft. per sec.

Specific volume at 5.5 lbs. and condition at end of expansion = 65.6 cu. ft. Therefore area of throat $A = \frac{144 \times 24 \times 65.6}{1405} = 161$ sq. in. and projected annular free area $A' = \frac{A}{\sin 20 \text{ deg.}} = 471$ sq. in.

If these orifices fill the whole annular ring then allowance must be made for the thickness of the walls forming the sides of the nozzle, as in Fig. 19. The projected thickness $t' = \frac{t}{\sin \alpha}$. This may be taken care of by a coefficient C which represents the ratio of the annular ring area to the clear nozzle mouth. C varies from 1.02 to 1.08 in commercial designs.

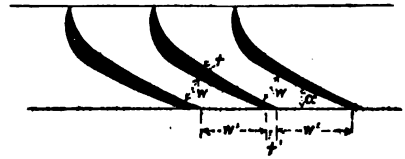


FIG. 19.—Orifice Design in Diaphragms of Impulse Turbines.

In the preceding problem, assuming $C = 1.05$, annular ring area $= 1.05 \times 471 = 494.5$ sq. in. Assume mean diameter of 38 in. for blades, then circumference $= 3.141 \times 38 = 119.38$ in. Hence radial height of orifices must be $\frac{494.5}{119.38} = 4.14$ in. The number of orifices that will be used will be determined by experience. In general the orifices should not have wide throats as these give the direction.

In designing a multi-stage turbine with a series of nozzles not occupying the full periphery of the wheel, allowance must be made for "**lead.**" The steam while crossing moving blades is carried around with the blades and leaves them further along in the direction of rotation than the point of entry. The succeeding nozzles must be placed forward to receive this steam. The amount of lead is influenced by steam velocities and the clearances on the two sides of the blade.

Impulse Blading

As previously stated, the steam leaves the nozzle at a certain absolute velocity and flows in practically straight lines at an angle of from 17½ deg. to 22½ deg. with the centre line of the blade wheel. This steam strikes a blade moving at a certain velocity in the same general direction as the steam. It therefore enters the blade with a relative velocity, as shown on the velocity diagram, Fig. 20, which is drawn for 1 lb. of steam.

Let V_1 = absolute velocity leaving nozzle at angle α, ft. per sec.
U = mean peripheral velocity of blades, ft. per sec.
V_2 = relative entering velocity of steam at angle β, ft. per sec.
V_3 = relative leaving velocity at angle γ, ft. per sec.
V_4 = absolute leaving velocity, ft. per sec.

It has already been shown that the vel. $V = 223.7\sqrt{H_1 - H_2}$ where $H_1 - H_2$ = heat available to produce kinetic energy; therefore $H_1 - H_2 = \left(\frac{V}{223.7}\right)^2$. Hence if the vel. is known, its equivalent heat value can be calculated.

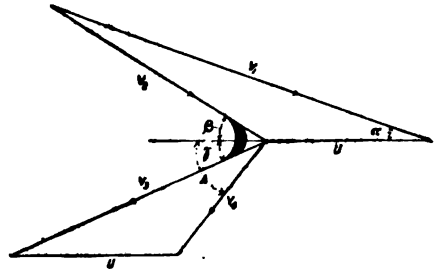


FIG. 20.—Velocity Diagram for Simple Impulse Turbines.

Hence for every lb. of steam in Fig. 20.

$$\text{Energy entering blade} = \left(\frac{V_1}{223.7}\right)^2$$

$$\text{Energy leaving blade} = \left(\frac{V_4}{223.7}\right)^2$$

$$\text{Neglecting friction, work done in blades} = \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2 \text{ in B.t.u.}$$

If there is no friction loss crossing the blade the peripheral vel. U for max. effy. should be chosen as $\frac{1}{2}V_1 \cos \alpha$. The friction loss in B.t.u. in crossing the blade may be represented by the fraction y of the relative entering vel. V_2. Hence relative vel. of outlet $V_3 = V_2\sqrt{(1 - y)}$. With friction the

peripheral speed U for max. effy. is less than $\frac{V_1}{2}\cos\alpha$ as above, and decreases as y increases; the new value of U is given by the formula $U = \frac{V_1 \cos\alpha}{1+\frac{1}{\sqrt{1-y}}}$

$\frac{U}{V_1}$ is called the **"velocity ratio"** and is generally assumed for each design. A suitable value of U can be assumed and the value of V_1 found. After allowing for nozzle losses, the heat drop per stage is easily determined.

Example. Assume a blade speed of 615 ft. per sec. and a nozzle angle of 18 deg., assume $y = 0.26$:

Then $615 = \frac{V_1 \cos 18}{1+\frac{1}{\sqrt{0.74}}} = \frac{0.951 V_1}{2.163}$, and $V_1 = 1400$ ft. per sec.

Let nozzle loss = 3 per cent of the theoretical velocity.

Then V_0 (theoretical vel. in nozzle) $= \frac{V_1}{0.97} = 1443$ ft. per sec. and

Heat drop per stage $(H_1 - H_2) = \left(\frac{V_0}{223.7}\right)^2 = 41.6$ B.t.u.

Usual velocity ratios on simple impulse turbines are from 0.3 to 0.35. On multiple stage impulse machines from 0.40 to 0.45.

The friction loss crossing the blade varies with the relative entering vel. V_2. In Marks' "Mechanical Engineers' Handbook," Lowenstein gives values for this loss as shown in Table 3.

Table 3. Friction Loss in Blades

Velocity (V_2) relative to the blade, ft. per sec.	200	400	600	800	1000	1500	2000	2500
Blade velocity coefficient $\sqrt{1-y}$.	0.953	0.918	0.888	0.863	0.841	0.801	0.774	0.754

These coefficients were derived from tests and include not only the friction loss in the moving blades but also the total losses from the nozzle mouth to the entrance of the succeeding intermediate or stationary blades if any. These losses are the loss in the clearance space between the nozzle and moving buckets, the loss due to the bluntness of the blade at inlet, steam friction losses in passing through the moving buckets, and finally the loss in the clearance space between the moving buckets and the succeeding intermediates.

Energy loss in passing across blade in B.t.u.

$$= \left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2$$

Effective energy to produce work on blade

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right)^2 + \left(\frac{V_3}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2$$

The efficiency of the nozzle and blade (where H is the total heat drop per stage)

$$= \frac{\left(\frac{V}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right) + \left(\frac{V_3}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2}{H}$$

End Thrust. The absolute entering vel. can be resolved into components at right angles one of which is along the axis of the shaft. The resultant axial thrust amounts to $\left(\frac{w}{g} V_1 \sin \alpha\right)$ for w lbs. of steam. Similarly the absolute leaving vel. produces a reaction equal to itself and in the opposite direction amounting to $-\frac{w}{g} V_4 \sin \Delta$ or more conveniently $-\frac{w}{g} V_3 \sin \gamma$. The **total axial thrust** is the sum of these two and amounts to

$$\frac{w}{g} (V_1 \sin \alpha - V_3 \sin \gamma)$$

The discharge angle of the blade is frequently made less than the inlet angle so as to decrease the loss due to V_4 and at the same time to partially counterbalance the friction loss. This angle may be 5 to 10 deg. smaller than the inlet angle.

Example. Let steam expand in a Rateau stage from 85 lbs. abs., 37 deg. superheat to 53 lbs. abs. y in nozzle = 0.06, nozzle angle = 20 deg., wheel speed = 550 ft. per sec.

$$V_1 = 223.7 \times \sqrt{0.94(H_1 - H_2)} = 223.7 \times \sqrt{0.94 \times 39.1} = 1355 \text{ ft. per sec.}$$

Construct velocity diagrams, Fig. 21, with equal blade angles and Fig. 22, with the exit angle closed 7½ deg.

The effect of closing the exit angle is clearly indicated by the decrease in the value of V_4 from 400 ft. per sec. in Fig. 21 to 332 ft. per sec. in Fig. 22. The energy loss decreases from 3.2 B.t.u. to 2.2 B.t.u. a saving of 1 B.t.u. or an improvement of the stage efficiency of 2½ per cent.

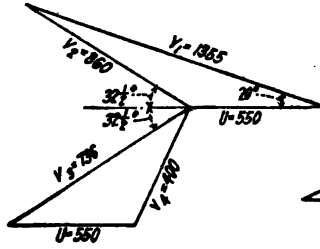


FIG. 21.

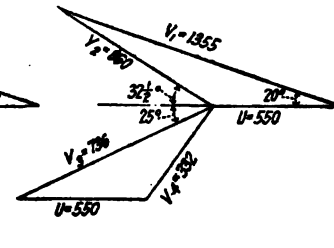


FIG. 22.

Effect of Closing Discharge Angle.

Work done on blade in Fig. 22

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2 - \left\{\left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2\right\}$$
$$= 36.75 - 2.2 - (14.8 - 10.85)$$
$$= 30.6 \text{ B.t.u. and}$$

Blade efficiency $= \frac{30.6}{39.1} = 78.3$ per cent.

It is generally assumed that the volume of steam passing across the blades remains the same. However, velocity is decreased. The inlet edge of a blade is theoretically the same height as the outlet of the nozzle. Hence the outlet height of the blade must be increased over the inlet height by the ratio $\frac{V_1 \sin 20 \text{ deg.}}{V_2 \sin 32\frac{1}{2} \text{ deg}}$ or 1.175 in Fig. 21 and in Fig. 22 by $\frac{V_1 \sin 20 \text{ deg.}}{V_2 \sin 25 \text{ deg.}} = 1.46$.

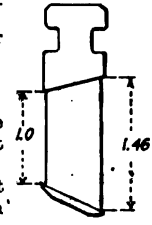


FIG. 23.

This amount of lengthening can only be done on the shorter blades as in Fig. 23. On longer blades the exit angle cannot be closed as much as indicated.

The axial thrust is slightly increased by closing the outlet angle. Thus in Fig. 21 the end thrust per lb. of steam $= \frac{V_1 \sin 20 - V_2 \sin 32.5}{32.2} = 68.5$ ft.-lbs. toward the exhaust; in Fig. 22 this is increased to 146 ft.-lbs.

The details of blading design and methods of laying out impulse blades are given on p. 715.

Curtis Blading

In the velocity compounded or Curtis stage the velocity leaving the nozzle is too high to be efficiently utilised in one row of revolving blades. Stationary guide blades are placed beyond the first row to reverse the direction of the steam and to direct the jet against a second set of revolving blades.

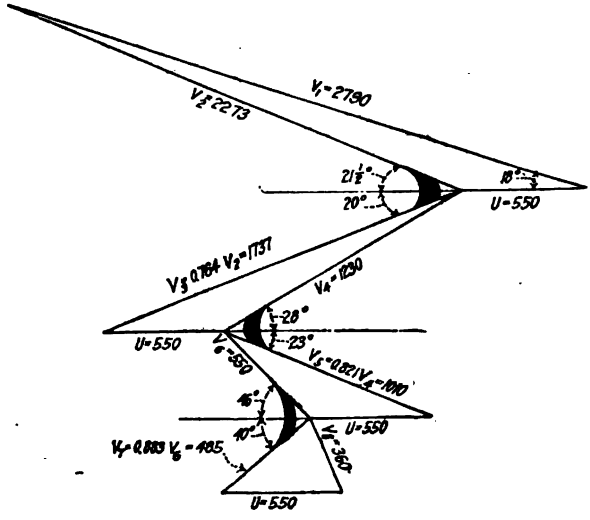


FIG. 24.—Velocity Diagram for a Curtis Stage.

Example. Let steam expand from 205 lbs. abs. 90 deg. superheat, to 26 lbs. abs. in a Curtis stage. $H_1 - H_2 = 165.5$ B.t.u. Allow y for nozzle = 6 per cent.
Then vel. of jet $V_1 = 223.7\sqrt{0.94 \times 165.5} = 2790$ ft. per sec.
Let nozzle angle = 18 deg. and wheel speed = 550 ft. per sec.

Construct velocity diagram, Fig. 24, allowing friction losses crossing blades as indicated in Table 3.

The work done on the blade

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_8}{223.7}\right)^2 - \left\{\left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2\right\} -$$

$$\left\{\left(\frac{V_4}{223.7}\right)^2 - \left(\frac{V_5}{223.7}\right)^2\right\} - \left\{\left(\frac{V_6}{223.7}\right)^2 - \left(\frac{V_7}{223.7}\right)^2\right\}$$

$$= 155.6 - 2.6 - (103.2 - 60.3) - (30.2 - 20.3) - (6.05 - 4.7) = 98.85 \text{ B.t.u.}$$

The efficiency of the blades $= \frac{98.85}{165.5} = 59.7$ per cent.

If the reversing blade and second revolving blades had not been used the heat converted to work would have been [155.6 − 30.2 − (103.2 − 60.3)] or 82.5 B.t.u. Hence the reversing blade effects an increase of work per stage of 19.7 per cent.

As before the outlet length of the various blades can be found. Assuming nozzle outlet = 1 in., this length is:

For the first blade $= \frac{V_1 \sin 18}{V_3 \sin 20} = \frac{2790 \times 0.309}{1737 \times 0.342} = 1.45$ in.

For the intermediate blade $= \frac{V_1 \sin 18}{V_3 \sin 23} = 2.10$ in. This blade can be expanded on outlet in both directions as the steam is not subject to centrifugal force in crossing it. It is usually ¼ in. narrower than first blade.

For the second revolving blade $= \frac{V_1 \sin 18}{V_7 \sin 40} = 2.77$ in. This blade does not have a sharp angle and is usually made ½ in. narrower than the first revolving blade. A usual construction for the first stage of a Curtis-Rateau unit is to have blades 1½ in., 1¼ in. and 1 in. wide, respectively for the three sets.

Some early impulse turbines were built without shrouding on the outer ends of the blades. It was found that the steam tended to "spill over" the ends of the blades. Shrouding forms a closed passage across the blade. However there is apparently reason to believe that a portion of the friction loss in crossing a blade is due to the centrifugal action of the steam in the blade.

When steam passes across the clearance space between the moving blade to the stationary blade and from this reversing blade to the second moving blade, the jet tends to spread. Hence the inlet sides of each of these blades are made about 1/16 in. longer than the outlet of the preceding blade. An-

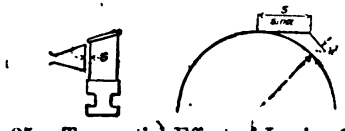


FIG. 25.—Tangential Effect of Issuing Steam Jet.

other reason for this lengthening of blade inlet side is as follows: The steam is flowing tangentially when crossing the distance between blades, or blade and nozzle. Hence it strikes the blades farther from the axis of the blade wheel than the mouth of the preceding nozzle or blade. This factor is of importance when the blade wheel is small. Fig. 25 illustrates how to find this blade height allowance. Path of steam $= \frac{S}{\sin \alpha}$ where α = nozzle angle. h' = necessary increase in blade length.

Parsons Blading

Blading. The Parsons type of turbine has already been defined as the impulse and reaction type. The blading is so designed that expansion takes place in the moving as well as in the stationary blades. All blades are of the converging orifice type and can be considered as such in the following discussion. There is a drop in pressure across each separate row of blades. Hence Parsons blading must have full peripheral admission. In certain sizes this will require short blades at the H.P. end, or else a small diameter of shaft, or both.

This H.P. section of the Parsons turbine is frequently replaced by a Curtis wheel with a few nozzles, which combination is discussed later. The velocity diagrams for Parsons blading are shown in Fig. 26.

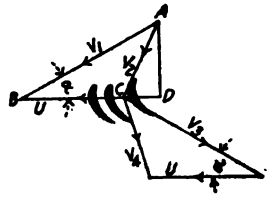


FIG. 26.—Velocity Diagram for Parsons Turbine.

It is generally assumed that the discharge angles of both stationary and moving blades are the same and this angle is usually 20 deg. on all normal blading.

The expansion in the first stationary blades must be such as to produce the velocity V_1. Neglecting friction the heat required to produce this velocity $H'_s = \left(\frac{V_1}{223.7}\right)^2$ B.t.u.

The relative velocity of the steam entering the moving blade is V_2 and in crossing this blade the pressure drops a certain amount and some heat H_m is made available for work.

$$H_m = \left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2$$

The absolute velocity leaving V_4 constitutes a loss if only a single moving and stationary blade is considered, and equals $\left(\frac{V_4}{223.7}\right)^2$ B.t.u.

The total work done on the blades

$$= H_s' + H_m - \left(\frac{V_4}{223.7}\right)^2$$

$$= \left(\frac{V_1}{223.7}\right)^2 + \left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2$$

The absolute leaving velocity V_4 from the moving blades becomes velocity of approach for the next set of stationary b.ades, so that for the second and following stages:

$$H_s = \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2$$

and the total work per stage $= H_s + H_m$

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2 + \left(\frac{V_3}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right)^2$$

If H is the total heat drop per stage then $\frac{H_s}{H}$ is the "**degree of reaction**" which is usually equal to 0.5. Hence equal work is developed by the impulse jet discharging from the stationary blade and by the reaction jet leaving the moving blade. As the exit angles are also equal, the triangles are equal and therefore $V_1 = V_3$ and $V_2 = V_4$.

Therefore the work done per stage

$$H_w = 2\left\{\left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right)^2\right\}$$

$$= \frac{2}{(223.7)^2}\left(V_1^2 - V_2^2\right)$$

But from construction $V_2^2 = AC^2 = AD^2 + DC^2$

$$AD = V_1 \sin\alpha \text{ and } DC = V_1 \cos\alpha - U$$

$$\therefore \quad V_2^2 = V_1^2 \sin^2\alpha + (V_1 \cos\alpha - U)^2$$

$$= V_1^2 \sin^2\alpha + V_1^2 \cos^2\alpha - 2UV_1 \cos\alpha + U^2$$

$$= V_1^2 - 2UV_1 \cos\alpha + U^2.$$

$$\therefore \quad H_w = \frac{2}{(223.7)^2}(V_1^2 - V_1^2 + 2UV_1 \cos\alpha - U^2)$$

$$= \frac{2}{(223.7)^2}(2UV_1 \cos\alpha - U^2)$$

$$= 2\left(\frac{V_1}{223.7}\right)^2\left(2\frac{U}{V_1}\cos\alpha - \frac{U^2}{V_1^2}\right)$$

$$= 2\left(\frac{V_1}{223.7}\right)^2 (2R\cos\alpha - R^2)$$

where $R = \frac{U}{V_1}$ the ratio of blade speed to steam speed, which is generally assumed for each turbine. The efficiency of the Parsons stage is found by assuming that theoretically $\left(\frac{V_1}{223.7}\right)^2$ is available for work in each row. Hence

$$\text{diagram efficiency } E = 2\frac{\left(\frac{V_1}{223.7}\right)^2\left(2R\cos\alpha - R^2\right)}{2\left(\frac{V_1}{223.7}\right)^2}.$$

$$= 2R\cos\alpha - R^2.$$

The highest economy with Parsons blading is shown in Fig. 27 to be about 90 per cent when $R = 0.94$ and $\alpha = 20$ deg. and decreases as R is increased or decreased from this amount.

On the land types of reaction turbine R varies from 0.5 to 0.7. Some large high efficiency turbines in this country have been built with $R = 0.80$. Similar ratios are used on the turbines with reduction gears in marine service.

Where direct coupled marine sets are used the wheel speed must be low and in order not to have too long a turbine or too large a rotor, low ratios of R are used, varying from 0.35 to 0.5. In some torpedo boats and other vessels requiring light weight, ratios as low as 0.3 have been used. Direct connected marine turbines with low values of R do not operate at high economy.

The equations given above do not consider any friction loss crossing the blade. Parsons blading is made with the entrance angle approximately tangent to V_1 or a few degrees larger than this angle. The exit angle is made equal to α. The efficiency of the blading is not as much affected by variation of the entrance angle as is impulse blading particularly as the steam velocity

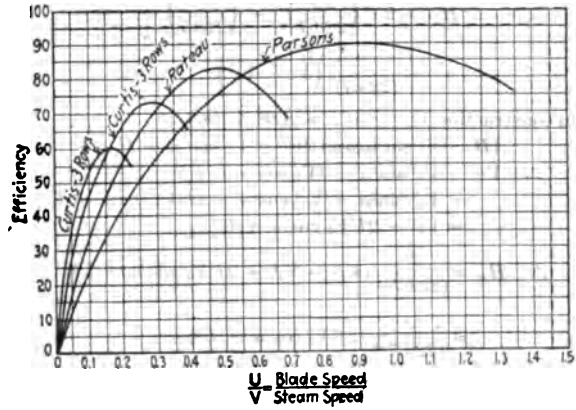


FIG. 27.—Relation Between Turbine Efficiency and Speed Ratio.

is much less. However, the inlet angle is usually increased 5 deg. to 15 deg. over the diagram angle B, particularly in marine turbines. This gives better economy at variable speeds.

The steam enters the blade at velocity V_2, turns through an angle and is discharged at velocity V_3. In passing over the blade surfaces a certain amount of kinetic energy is converted into heat by friction. Hence the heat drop per stage exceeds the work done per stage by a certain amount, depending on the friction factor of the blading.

Considering this friction loss, the necessary heat drop per stage H_d can be calculated from the formula.

$$H_d = 2\left(\frac{V_1}{223.7}\right)^2 \gamma \left\{ \left(2R\cos\alpha - R^2\right) + E_s \right\}$$

where E_s = the stage efficiency which increases from the h.-p. to the exhaust and γ = the ratio of working steam to total steam considering clearance. γ is discussed later.

This formula can be further simplified if desired.

$E_s = \gamma(2R\cos\alpha - R^2) \times$ nozzle efficiency of rows.

Substituting in the formula for heat drop per stage, the heat assignment per pair of rows becomes

$$H_d = \frac{2\left(\frac{V_1}{223.7}\right)^2}{\text{nozzle efficiency of rows}}.$$

If the nozzle efficiency of the Parsons rows is known an easy method of calculating the heat assignment is available. This nozzle efficiency varies from 85 to 90 per cent.

The frictional losses through the blades depend on velocity of the steam, on density, on surface area of blading and to some extent on smoothness of surface. In general frictional losses as above calculated will decrease the efficiency of the unit from 10 to 15 per cent and this figure includes the nozzle losses of reaction blading.

When the heat allotment per stage per lb. of steam H_d has been determined as above, the **number of rows on the Parsons drum** can be found by the equation $N = \frac{(H_1 - H_2)F}{H_d}$, where $(H_1 - H_2)$ is the adiabatic heat drop available per pound of steam and F is the reheat factor.

As the steam expands passing through a turbine, its volume increases. Hence to maintain the same steam speed through the blades the clear area through them must increase for the steam jet obeys the law $V = \frac{Q}{A}$ where V = the velocity in ft. per sec., Q = cu. ft. per sec. flowing and A = area of passage in sq. ft. at right angles to the jet.

To secure this constant steam velocity, either each row of blades must be longer than the preceding one, or the outlet angle must be increased. The first is impractical construction. Hence some builders maintain the same length of blade at constant discharge angle for a series of rows on the same diameter of cylinder and then lengthen the blades for a second series of rows, also with constant discharge angle, then a third set, etc. Each such set of blade rows of constant diameter is called a "**barrel**" and each series of barrels of approximately the same mean diameter is called a "**cylinder.**"

Two methods of construction are followed in a barrel. In the one described above the length of blade is so chosen that the desired steam velocity is secured with the chosen discharge angle in the central rows of the barrel. The steam velocities are below this normal value in the first rows and exceed it in the last rows. This method simplifies design but does not secure highest efficiency.

The second method is to construct the turbine with a series of barrels as before, but to alter the outlet angle of the blades by twisting them slightly. This is known as "**gaging**" **the blades.** The **per cent gaging** is the ratio of the clear opening at right angles to the jet to the full annular opening between drum and casing. This gaging varies from 17 to 35 per cent with normal blades. On large condensing turbines with high vacua the last rows of blades at the exhaust end become excessively long. The outlet angle must then be greatly increased above normal to keep the blade length down to the desired amount. These flattened blades are known as "**semi-wing**" when the gaging is from 35 to 50 per cent and "**wing**" when from 50 to 80 per cent. Naturally the final leaving velocity V_4 from the last row is greatly

increased by this construction, and hence the leaving loss in the exhaust $\left(\frac{V_4}{223.7}\right)^2$ is also increased.

Leakage over Blade Tips. The steam pressure is the same on both sides of the moving blade in **impulse turbines.** Hence there is no leakage over the ends of the blades from difference in pressure and the radial clearances may be large.

In **Parsons turbines** a difference of pressure exists across each stationary and moving blade and hence there is leakage of steam through the clearance at the ends of the blades. The clearance must be kept at a certain amount for mechanical reasons, and in proportion to the length of the blades is therefore greater at the H.P. end than at the L.P. end. In Parsons design all tip

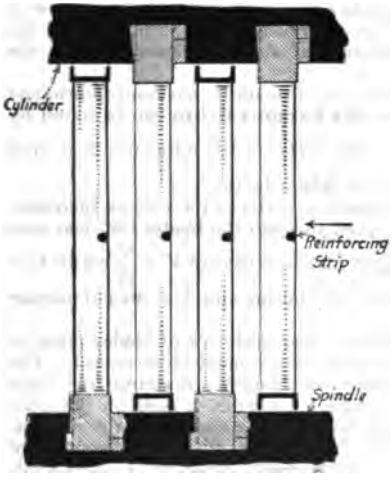


Fig 28.

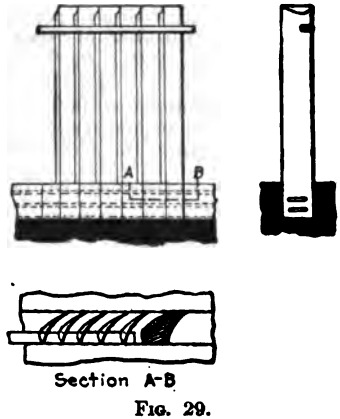


Fig. 29.

clearances should be kept as small as possible. Various schemes are used to secure this desired result. Allis-Chalmers use a channel shaped shroud with fine edges, as shown in Fig. 28. The usual construction is to thin off the tip of each blade to a fine knife edge, as in Fig. 29.

If from any cause the tips of the blades should rub the casing or drum when in operation, this fine edge will wear down with very little heating from friction and thus with little or no consequent extension of the blade. Hence a running clearance is secured again instantly, without danger of a "blade strip." These blade strips occur when the whole cross-section rubs and the blades heat up and expand so that they rub harder than ever; then something has to let go

It is necessary to be able to calculate the leakage through this clearance and to make allowance for it. This can be found by first calculating the total flow through a row of blades and the clearance as follows:

$$W_T = \pi d \frac{\left(l + \frac{c}{M \sin \alpha}\right) V_1 M \sin \alpha}{144 v_1}$$

where W_T = the total flow of steam through blades and clearance in lbs. per sec.; d = the mean diam. of the blades in in. (found by calculating the diam. from the middle of blades on each side of the axis); l = the length of blades in in.; c = the clearance of blades in in.; α = the discharge angle of the blade; V_1 = the steam velocity crossing the blade in ft. per sec. and v_1 = the specific volume of the steam leaving the blade. M = thickness coefficient for Parsons blading = 0.73 to 0.8 for calculating steam flow.

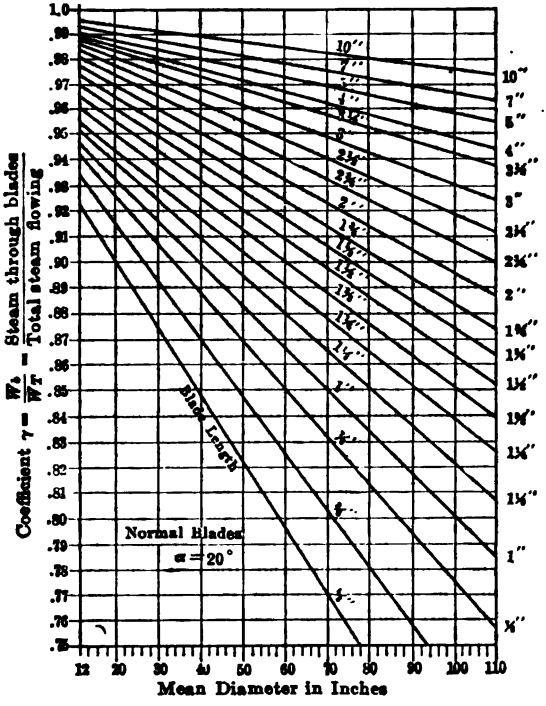


Fig. 30.—Ratio of Working Steam Through Blades to Total Steam in Turbine. $\alpha = 20°$.

The weight of working steam W_b in lbs. per sec. passing through the blade is

$$W_b = \pi d \frac{l V_1 M \sin \alpha}{144 v_1}$$

Hence the per cent loss from leakage through clearance

$$L = \frac{\frac{c}{M \sin \alpha}}{l + \frac{c}{M \sin \alpha}} = \frac{c}{l M \sin \alpha + c}$$

and the **per cent of the total steam passing through the blades**

$$\gamma = \frac{l}{l + \frac{c}{\mathbf{M} \sin \alpha}} = \frac{l\,\mathbf{M} \sin \alpha}{l\mathbf{M} \sin \alpha + c}$$

Clearances over the ends of Parsons blading varies with the diam. of the drum according to the following equation

$$c = 0.015 + 0.003D + 0.005\, l_4.$$

where c is the clearance in in. and D the mean diam. of the blade ring in ft. and l_4 = actual height of blade in in.

It is convenient to plot up the ratio γ of the working steam to total steam passing through the turbine for the various blade angles, blade lengths and clearances used in turbine design. This can be done in each design office for the data on hand for their design practice.

An example of such a diagram is shown in Fig. 30 which is reproduced from Goudie's "Steam Turbine" and is based on a formula by Speakman for normal blades with an outlet angle of 20 deg. $c = 0.01 + 0.008D$.

Leakage losses become excessive with reaction blades less than $\frac{3}{4}$ in. long. In Curtis-Parsons combination turbines the drum diam. are usually such that no clearance is less than 0.025 in.

Effect of "Spilling Over." As the steam crosses a moving Parsons blade it is subject to centrifugal force which throws a portion of this steam against the casing before it has completely crossed the blade. Evidence of this can be noted in the L.P. section of turbines where the steam is wet and where the casing opposite the blades shows considerable signs of pitting. This action has received little study and cannot be prevented in ordinary reaction blading. A few manufacturers like Allis-Chalmers provide a channel shaped shroud on the ends of the blades to prevent this action. It has been a question however whether the additional whirling friction loss of this shroud does not absorb the saving from preventing the spilling-over of the ordinary turbine reaction blades.

Axial Clearance. The blades of impulse wheels should be set comparatively close to the nozzles to prevent spreading of the high velocity steam jets. In reaction turbines on the other hand, the exit velocity from the blades is much lower and the jet fills the whole periphery. There is therefore less chance for lateral spreading and the axial clearances may therefore be more liberal. In general axial clearance is fixed by consideration of the differential expansion that takes place in a long rotor and casing. *J. M. Newton* has suggested the following formula.

$$\mathbf{P}_a = \mathbf{2B} + \frac{1}{4} + \frac{l}{16}$$

where P_a = the axial pitch of the blade rows in in.; B = width of blades in in. and l = length of blades in in.

Several attempts are being made to substitute fine axial clearance and various forms of labyrinth for fine radial clearance to prevent leakage losses have been tried. These have not been widely adopted at the present time.

Blade Lengths. The formula for heat assignment per stage given above depends on the value of V_1, the steam speed across the blade and R the ratio of blade speed to steam speed. Usually R is assumed for the design. The larger the value of R the greater the number of rows of blades the longer the turbine and the higher the efficiency. With R small the turbine is short, compact but less efficient.

If the blade speed U and the ratio R are known, the heat assignment for each stage can be calculated.

Either the first blades would be too short or the last blades too long if the full expansion were assumed to take place on one drum. Hence it is usual to step down the drum of Parsons turbines so that the first row of blades are not less than ¾ in. long.

The usual construction calls for three steps with ratio of diam. $1:\sqrt{2}:2$.

The velocity of the steam varies as $\sqrt{\text{B.t.u.}}$ and also as the mean diam. The number of rows varies as the B.t.u. The number of rows therefore on a reduced diam. will vary as the square of the mean diam.

Hence one row on the L.P. large diam. section will require two rows to replace it in the intermediate, or four rows on the H.P. section. This makes a long H.P. section and increases the necessary clearance of the blades. Hence it is frequently replaced by a Curtis wheel.

The work is generally divided between the three drums of a standard Parsons turbine in the ratio of 1:1:2, the greater portion done on the L.P. section.

Parsons blading is made in standardized sections varying from ¼ in. wide to 1½ in. wide. The width of blade varies with the length of blade, the blade discharge angle, the density of the steam, the velocity of the steam and the blade speed. The maximum value of the ratio of blade length to blade width should not exceed 15 on slow speed and 12 on high speed turbines. Usually it does not exceed 10 on high speed turbines.

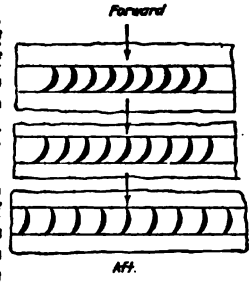


FIG. 31.—Parsons "Semi-wing" and "Wing" Blading.

In some units it is found impossible to get the full volume of steam through the last few rows of blades if the normal blade angle of 20 deg. is maintained. Hence the angles of these blades are increased. Two standard forms of such blades are in use, the "semi-wing" with an angle of about 35 deg. and the "wing" making an angle of 50 deg. These are shown in Fig. 31.

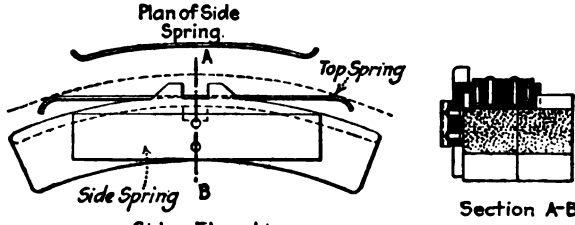
Increasing the angles in this way decreases R and decreases the efficiency of the last few rows, but allows them to utilise the vacuum to the fullest extent.

In Fig. 26 it is evident that there is no axial velocity for $V_1 = V_2$ and both discharge at the same angle α. The calculation of end thrust due to pressure drop across blades and to increases in drum diam. is fully treated on p. 631.

CLEARANCES AND LEAKAGE

A gland may be described as a special casing placed (1) around the shaft where it leaves the cylinder or (2) where a diaphragm between stages encircles the shaft in impulse turbines. The two forms will be discussed separately. In both cases rings or serrated teeth are provided to cause a number of wire-drawings of the steam during its passage through the clearance area of the gland and thus to reduce the leakage. The clearance for mechanical reasons cannot be avoided and hence leakage is bound to occur.

Shaft Glands generally consist of bronze or carbon rings closely fitting the shaft and also fitting special grooves in the gland casing. At the high-pressure end the pressure drop through the glands generally exceeds the critical value, that is, the lower absolute pressure—often atmospheric or



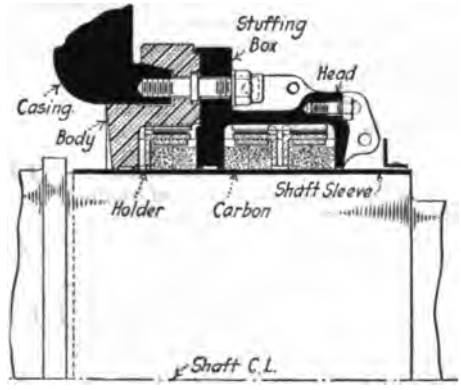


FIG. 32.—Carbon Ring Shaft Glands.

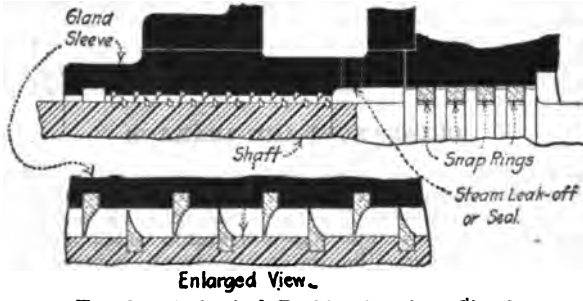


FIG. 33.—Labyrinth Packing for Shaft Gland.

thereabouts—is less than 0.58 of the absolute pressure inside the cylinder casing. In this case the following formulæ may be used:

For superheated steam, $W_s = 0.3155A\sqrt{\frac{P_1}{v_{s1}}}$ and for dry or slightly wet steam, $W_w = 0.0173A(P_1)^{8/12}$

where W_s = lbs. of superheated steam per sec.
W_w = lbs. of dry or slightly wet steam per sec.
P_1 = initial press. in lbs. per sq. in. abs. on turbine side of gland.
A = annular clearance area in sq. in.
v_{s1} = sp. vol. of superheated steam in cu. ft. at press. P_1

The gland at the **low-pressure end** of the turbine either is supplied with the steam which has leaked past the gland at the high-pressure end or with steam from one of the earlier stages of the turbine. In any case this steam can be assumed to reach the low-pressure gland in a saturated condition and the leakage can be calculated from the second equation given above.

The **clearance between the rings** of these glands and the shafts usually ranges from 0.005 to 0.008 inch. One of these glands is shown in Fig. 32. Some manufacturers of Parsons turbines use a form of gland packing where their spindles leave the cylinder such as is shown in Fig. 33. The rings shown in the former figure are replaced by labyrinth packing rings of brass. The peripheral speed of the shaft is low so that these can be caked in the shaft. The leakage can be calculated by the preceding equations or by the later one given for dummy leakage. Some turbines are fitted with glands having carbon rings instead of bronze. The leakage can be calculated by the preceding equations.

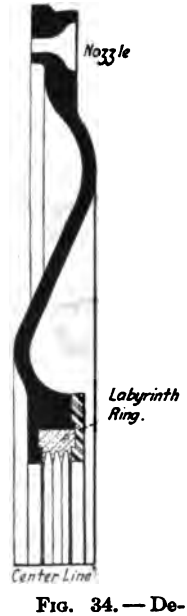


FIG. 34.—Detail Construction of Diaphragms.

The Diaphragm Gland between stages of an impulse turbine is usually of the form shown in Fig. 34. If the turbine has few stages as in the astern section, the pressure difference will exceed the critical ratio and the preceding formulæ may be used, for the serrated teeth have a retarding effect on the flow of steam and do not allow it to blow straight through. In multi-stage turbines the pressure difference between stages is usually less than the critical ratio and the loss of steam through leakage must be determined in a different manner.

Goudie in "Steam Turbines" suggests the following simple method. First determine the heat drop per stage as in the original design of the turbine. Assume that the steam expands through the clearance as through an orifice but with efficiencies as low as 0.80 to 0.85. Then the resulting velocity through the clearance space will be

$$V_1 = 223.7\sqrt{0.85H_s}$$

The **loss of steam** through the labyrinth in pounds per second will be $\frac{A_1V_1}{144v_1}$, where A_1 = the annular area of the clearance space in sq. in. V_1 =

the velocity found above and v_1 = the sp. vol. of the steam at the lower pressure.

The leakage loss in pounds of steam naturally decreases toward the low-pressure end of the turbine for the ratio of the area through the glands to the area through the nozzles decreases and in the last stages becomes a negligible factor. The usual method to allow for such leakage in turbine design is to assume a mean value of this leakage for the whole turbine.

The materials used for labyrinth diaphragms are babbitt, aluminum or bronze. The teeth are usually turned with very sharp edges. The shaft wears these down to its own clearance by being revolved slowly when first assembled. In general this clearance varies from 0.01 to 0.04 in.

Water-sealed Glands consist of an impeller with either curved or straight vanes on each side of the central web, fastened firmly to the shaft and revolv-

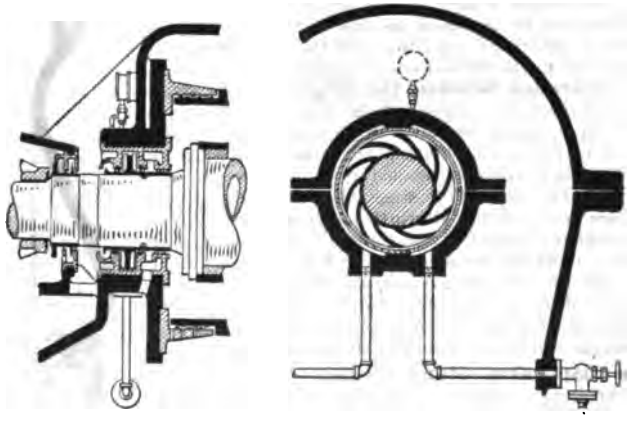


FIG. 35.—Allis-Chalmers Water Gland.

ing in a small chamber in the gland casing. This impeller is supplied with water under a head of 5 to 10 lb. per sq. in. A solid ring of water is formed at the outer ends of the impeller blades by the centrifugal pump action of the impeller and leakage of air into vacuum is thereby prevented. The water gland is more suitable for packing against vacuum than against pressure, for heat tends to evaporate the water. If this is a necessary condition of operation, provision should be made to circulate water through the gland, as in Fig. 35. When the gland is used against pressure much above atmospheric some labyrinth packings are usually placed between the impeller and the pressure chamber as shown in the figure. These wire-draw the steam and the amount which does actually pass them is either led into a lower stage or escapes to the exhaust, as shown in Fig. 35. In this figure the water gland seals the vacuum only against atmospheric pressure.

Power Loss in a Water-sealed Gland. The impeller is virtually a small centrifugal pump maintaining the ring of water under a pressure. Hence it consumes a certain amount of the power of the turbine. The following

method of calculating the power required by the gland has been suggested by Goudie. If the pressures on each side of the chamber were the same, the water surface in Fig. 36 would be the same in compartments A and B under the influence of the centrifugal pumping action. But the vacuum on side A causes the water in that compartment to be drawn to the centre of the shaft until the centrifugal force on the column h balances the difference in pressure between vacuum and atmosphere. Any water leakage inward goes to the condenser. Pure water must be used for such glands or the impeller will scale up or fill with dirt. The water gland is absolutely tight regardless of changes in vacuum.

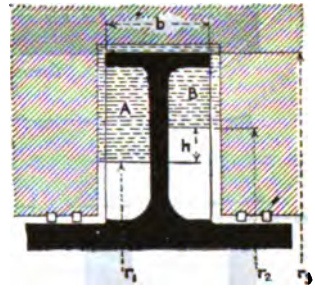


FIG. 36.—Water Gland Design.

Let N = R.P.M.
r_1 = radius of water on low-pressure side in ft.
r_2 = radius of water on high-pressure side in ft.
r_3 = outer radius of gland impeller in ft.
b = width of impeller in ft.

r_1 must be assumed from the construction of the gland and the shaft diameter. r_2 can be found from the equation $r_2 = \sqrt{13{,}510\frac{P}{N^2} + r_1^2}$, where P is the pressure difference across the gland in lb. per sq. in.

The horsepower lost in driving a water gland is

$$\text{H.P.} = 13.1\, f \left(\frac{N}{100}\right)^3 \{\tfrac{1}{5}(2r_3^5 - r_1^5 - r_2^5) + br_3^4\}$$

where f is a constant ranging from 0.004 to 0.008. The side clearance of the impeller depends on the distance from the thrust block and varies from 0.030 to 0.07 inch.

Dummy Piston Packings. Dummy pistons are placed in Parsons turbines to balance the end thrust on annular surfaces exposed to unbalanced axial pressure. There must be clearance between the revolving spindle and the stationary cylinder for mechanical reasons. Hence various forms of labyrinth packings are used to reduce the leakage through this clearance to a minimum. Fig. 37 shows one form of this packing.

Brass segments about 4 in. long are inserted in grooves in the casing and calked into place. They are then ground true by pulling the revolving rotor against them at slow speed. Clearance must be allowed between the ends of the segments to care for the difference of expansion of brass and cast iron. The steam is forced to travel toward the centre of the shaft in passing through the fine axial clearance between stationary and moving dummies, and this passage is opposed by centrifugal force.

Sometimes the spindle has considerable axial expansion as compared to the cylinder, as, for instance, when the balance piston is placed in the low-pressure end of a turbine. Then a labyrinth packing with radial clearance only, as shown in Fig. 38, is used.

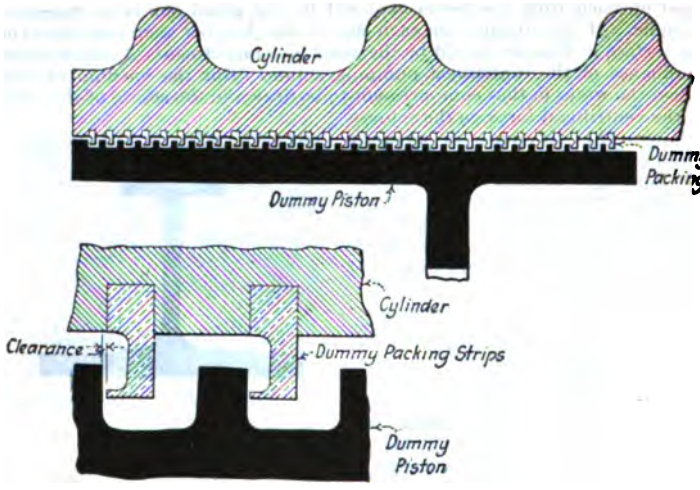


FIG. 37.—Parsons Balance Pistons with Axial Clearance.

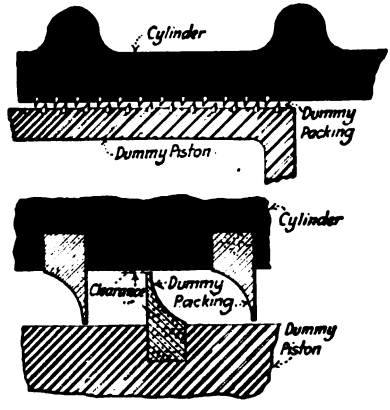


FIG. 38.—Labyrinth Packing with Radial Clearance Only.

The **loss** through both styles of dummies can be calculated by the following equation given by Martin:

$$w_d = 0.4722\, A \sqrt{\frac{P_1^2 - P_2^2}{P_1 v_1 (N + \log_e x)}}$$

where w_d = lbs. of steam per sec. passing through dummies
A = leakage area through dummies in sq. in.
P_1 = initial press. in lbs. per sq. in. on dummies.
P_2 = final press. in lbs. per sq. in. beyond dummies.
v_1 = specific volume of steam in cu. ft. at pressure P_1.
N = number of dummies.
x = ratio $\frac{P_1}{P_2}$.

Several graphic methods for calculating this leakage have been suggested and one of these is shown in Goudie's "Steam Turbines." From such a chart the number of throttlings required to limit the loss to certain limits can be found.

The designer is usually interested in determining the **number of dummy rings required at each piston** to limit the loss to a fixed amount. Morrow has suggested the following equation for this purpose:

$$n = \left(40P_1 - 2600\frac{w_d}{A}\right) + \left(540\frac{w_d}{A} - P_1\right)$$

where n = number of dummies on the piston.
P_1 = the abs. press. of the steam in lb. per sq. in. before piston.
w_d = permissible leakage of steam in lb. per sec.
A = annular area of clearance through dummy in sq. in.

When the clearance is axial as in Fig. 37, it varies from 0.006 to 0.020 in., depending on the size of the rotor. If the clearance is radial, as in Fig. 38, the clearance varies from 0.01 to 0.025 inch.

ROTATIONAL LOSSES

Disc, Blade and Drum Losses. Disc friction losses occur when a disc wheel carrying one or more rows of blades is revolved in a fluid like steam. These losses are due to two factors, (1) the actual friction loss of the fluid on the surface of the disc and (2) the pumping action of the disc which functions in a similar manner to the impeller of a centrifugal pump. The fluid is whirled by this centrifugal action from the centre of the disc to the periphery. Apparently this pumping loss is of considerable magnitude and steps should be taken to minimize it as much as possible. This can best be done by keeping the clearance on both sides of the disc as small as possible.

The whirling loss of a disc in free air can be reduced to ½ or ¼ by putting casings close to each side of the disc. This is the usual construction in steam turbines. Considerable study is now being given to designs for reducing this side clearance to a minimum and thus reducing the pumping losses.

Extensive experimental data on these losses are not available, for that obtained from discs in open air are not applicable to turbine conditions. Wm. Kerr's experiments at Glasgow indicate that the disc losses in B.t.u. per hour are as follows for dry steam:

$$N_d = 154.6D^2 \frac{1}{9}\left(\frac{U}{100}\right)^3$$

where N_d = B.t.u. loss per hr.
D = the mean blade diam. in ft.
v = the sp. vol. of the steam in cu. ft.
U = the peripheral vel. at the mean diam. of the blades in ft. per sec.

Blade Losses. A portion of the blade ring in the earlier stages of impulse turbines frequently does not run full of steam. The churning action of these empty rotating blades causes another friction loss. Kerr found that this loss in B.t.u. per hour could be expressed by the following formula when discs revolved in dry steam.

$$N_b = 1166.3BSD^{1.5}\frac{1}{v}\left(\frac{U}{100}\right)^3$$

where N_b = B.t.u. loss per hr.
S = fraction of mean circumference not receiving working steam,
D, v and U as stated in the preceding equation.
l = blade length in in.
B = a correction factor based on the number of rows of blades on the disc, as follows:

No. of rows.....	1	2	3	4
B = ...	1	1.23	1.8	2.9

This blade loss occurs when the idle blades run free in an atmosphere of steam. European practice now encloses all blades between closely fitting side rings to reduce this churning action. Solid rings are also provided for the idle section of the intermediate blade section of Curtis velocity stages.

Effect of Superheat and Moisture on Rotational Losses. The above formulæ give losses for discs whirling in practically dry steam. It has been noted on tests that the disc friction decreases with superheat considerably more than would be indicated by the density alone. Sufficient data is not at hand to indicate this decrease. In general this can be assumed to be 6 per cent decrease for 50 deg. superheat and 12 per cent decrease for 100 deg. superheat.

Moisture in steam increases the friction loss at a greater rate than the density increases. This correction is not definitely known. Some designers assume it as twice the percentage of moisture present.

Drum Friction Loss of Parsons reaction turbines is very small and may be neglected in designs. The blades all run with full peripheral admission and hence there is no correction for idle blades.

Astern turbines usually consist of impulse discs placed in the exhaust chamber of the main unit. These turn backward in vacuum when the main unit is running forward. Due to the low steam density in a 28 in. vacuum the friction losses become very small, some British tests indicating these to amount to only 0.3 per cent.

Condition of Surfaces. All of the above frictional losses are dependent to a large degree on the condition of the surfaces of the drums, casings and blades. Highly polished surfaces in all these parts tend to greatly reduce friction losses. Hence many builders now finish-machine all interior parts of casings as well as spindles.

Corrosion tends to roughen surfaces and to increase friction losses and hence every means tending to produce corrosion such as leaky throttle valves, impure feed, etc., should be avoided.

Bearing Friction

High-speed turbine bearings always have a perfect oil film between the journal and the bearing. This oil film thus carries the load and as it is rubbed both by the shaft and the bearing, its temperature is raised by the resulting friction. The heat generated is removed by supplying fresh cool oil to the bearing and cooling the hot oil in an oil cooler.

Various formulæ have been derived to calculate this friction loss in bearings. Martin in "Engineering," Aug. 7, 1918, gives a formula from which the following formula was derived:

B.t.u. friction loss in bearings per hr. = $0.00853d^2lS$, where d = diameter of bearing in in., l = length of bearing in in., S = rev. per min.

The friction loss in bearings of moderate sized turbines is generally not more than 1 per cent of the turbine output. But there is also the work of driving governors and oil pumps through gearing if these are furnished with the turbine. These may increase the total friction loss of the unit from 1.5 to 2 per cent on large machines and from 2 to 4 per cent on small machines.

Thrust-block Losses

Friction losses occur at the thrust blocks. In geared turbines there is little end thrust to be taken on the turbine thrust. The thrust of the screw must be taken on the thrust block of the main shaft. When turbines are direct connected to the propeller shafts the thrust blocks may have to take up considerable end thrust.

The following formula gives this loss in B.t.u. per second,

$$h_t = \frac{TrS}{30{,}680}$$

where T = total thrust in lb., r = mean radius of thrust collar in in., S = rev. per min. of shaft.

PROVISIONAL PROPORTIONS OF A STEAM TURBINE

Wheel Efficiencies

Turbine design can be greatly simplified by the use of curves giving the wheel efficiencies of various forms of construction for various ratios of wheel speed U to steam velocity V_s corresponding to the adiabatic heat drop. This relation is given in Fig. 27 from a paper on "Recent Developments in Steam Turbine Practice" by K. Baumann, Institution of Electrical Engineers, 1912.

The efficiencies include losses due to steam friction in the blades, windage losses in the blades and losses in the main glands and diaphragm glands and bearings. It has been pointed out that average values for usual turbine construction show efficiencies 2 to 3 per cent less than given in the curves. These lower values should be used in conservative design.

The Limiting Factors of Turbine Capacity

The last row of blades limits the capacity of a turbine. The mechanical stresses and the fatiguing effect of vibration are greatest in this row and hence a practical limitation is placed on the peripheral speed of this row of blades. These speeds have been referred to elsewhere. The B.t.u. drop is greatest in these last rows. Hence care must be taken in designing the blade angles to avoid excessive leaving losses in the exhaust.

The length of blades is limited, as previously pointed out, to about $\frac{1}{5}$ of the mean diam. Otherwise the blades, being radial, become so wide apart at their tips that the steam jet is not given direction

If the blade area is restricted or, in the case of the last nozzles of a Rateau turbine, if the pressure drop is great, the leaving loss from the last wheel becomes excessive.

The designer must therefore either place limits on the capacity of the turbine for given operating conditions, or must resort to unusual features in the last row, such as wing blades or pure impulse blades in the last rows of Parsons turbines, or use special alloy steel blades in the last stage of impulse turbines which permits an increase in wheel speed.

Provisional Proportions of an Impulse Turbine

The output at a stated speed, the initial pressure superheat and vacuum are usually given. The turbine may consist of a large number of stages of small diameter and low blade speeds or may be built with few stages of large diameter and high blade speeds.

Simple impulse and simple Curtis units may be designed directly from theoretical considerations as in the next section. Many other design methods are also in daily use.

Multi-stage turbines may also be designed from theoretical considerations. The usual method, however, is to arrive at some provisional sizes and then check over the design in detail after the general proportions have been calculated.

In starting this provisional design, a suitable blade speed based on the material used for blades, method of fastening, etc., must be assumed. This choice is governed largely by the manufacturer's blading practice. Then either the ratio of blade speed to wheel speed may be assumed and the stage efficiency found from Fig. 27, or the stage efficiency based on the performance of earlier turbines of the same type may be chosen and the ratio of blade speed to steam speed found from Fig. 27.

Blade speeds are constantly being increased. This permits the use of larger blade wheels and consequently longer blades, so that more steam can pass through the turbine. Hence greater output at given R.P.M. The steam speed must also be increased to maintain the desired ratio of blade speed to steam speed. This increase in size, however, decreases efficiency slightly for the friction losses are greater due to increased steam velocity and the disk friction is increased with increased diam.

When the blade speed has been selected the **mean diam.** can be found thus

$$d = \frac{60 \times 12 \times U}{\pi N} = 229\frac{U}{N}.$$

where d = mean diam. in in.
U = blade speed in ft. per sec.
N = R.P.M.

If R = the ratio of blade speed to steam speed, then the **allowable steam** speed can be found by the equation $V_s = \frac{U}{R}$.

The **adiabatic heat drop per stage** H_s can be determined from the formula

$$H_s = \left(\frac{V_o}{223.7}\right)^2.$$

If this heat drop does not exceed the critical value (*i.e.*, P_2 = or > $0.58P_1$) converging orifices can be used in the diaphragms. But if it exceeds this critical value then expanding nozzles must be used. The heat drop to produce

this critical velocity at the higher pressures averages about 45 B.t.u. so that the apparent limiting velocity

$$V_o = 223.7\sqrt{45} = 1500 \text{ ft. per sec.}$$

However, some under expansion is permissible in converging orifices with a small loss, as previously pointed out, so that somewhat larger heat drops and higher velocities than the critical value are permissible.

Where blade speeds in Rateau turbines from 750 to 850 ft. per sec. have been used in recent American designs for large output units, the orifices in the diaphragms have been made expanding nozzles by lengthening the parallel portion as in *A*, Fig. 39, and increasing the radial length of this parallel portion toward the mouth, as shown in section in *B*, Fig. 39.

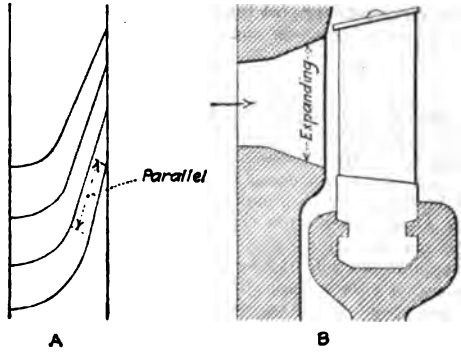


FIG. 39.—Recent Orifice Design.

When H_s has been determined the **number of stages** n can be found from

$$n = \frac{F(H_1 - H_s)}{H_s}.$$

where F is the reheat factor found from the assumed stage efficiency and $(H_1 - H_2)$ the total heat drop.

The **steam consumption** w in lbs. per S.H.P.-hr. can now be found

$$w = \frac{2546.5}{F \times E_s \times (H_1 - H_2)(1 - f_d)}.$$

where E_s = stage efficiency from curves,
and f_d = outside losses as bearing and thrust friction, etc.

The condition curve, cumulative heat curve, and specific volume curves can now be drawn and the pressures at each stage determined.

The next step is to check the limiting height of the last nozzle outlet and hence the height of the blades. The length of these blades in in. can be found from the equation

$$l = \frac{12 W v_o}{s\pi D V_1 \sin \alpha}$$

where W = total steam in lbs. per second passing through the turbines; v_o is the specific volume; s the ratio of free opening to full peripheral area of mean blade diam. (*i.e.*, allowance made for width of diaphragms), s varies from 83 to 90 per cent; D the mean diam. in ft.; V_1 the steam speed leaving the orifice and α the discharge angle of the orifice.

If $\frac{l}{12}$ exceeds $\frac{D}{5}$ then either a larger wheel and hence higher blade speed must be assumed or the discharge angle of the nozzles must be increased.

Usually the latter is adopted. The effect of this increase on the leaving loss of the last row of blades can be studied by means of velocity diagrams.

The length of blades at other stages can be calculated by the same equation until the minimum length of blade is reached. This length is usually about ¾ in. Then instead of admitting steam around the full periphery, partial admission in two groups of nozzles on opposite sides of the wheel is used. This tends to keep the wheel in better balance and reduces vibration. The necessary **circumferential length of nozzle mouth** K in in. for each group can be calculated from the equation

$$K = \frac{Wv_o \times 144}{2slV_1 \sin \alpha}$$

where l is the minimum blade length.

The whirling losses of discs increase rapidly as noted on p. 562, when portions of the blades are not filled with working steam. Some designers do not aim to get equal work from each stage but maintain a constant ratio of blade speed to steam speed throughout their machine. They decrease the blade speed from the exhaust end toward the steam end of the turbine, thereby securing wheels of smaller diam. toward the steam end. This decreases the whirling losses of the discs and permits full peripheral admission in earlier stages than where all wheels are of the large diam., because lower steam speeds must be used with the lower blade speeds.

In the first stage of a Rateau turbine with partial peripheral admission the rotational losses are often excessive. Another plan to reduce these is to allow a smaller ratio of blade speed to steam speed in this stage and thus secure a larger heat drop, lower pressure and somewhat lower stage efficiency for this first stage.

The general plan, however, to avoid these excessive rotational losses and to reduce the temperature and pressure inside the turbine casing is to substitute a Curtis wheel for several of the first stage Rateau wheels. Theoretically this should materially reduce the efficiency of the turbine. Commercially it does not seriously do so and besides it makes the machine much shorter. By using a Curtis wheel of moderate diam. the efficiency may be kept near maximum.

When the blade speed and ratio of blade speed to steam speed are chosen from Fig. 27, the heat assignment to this stage and to the remaining stages can be calculated as in a pure Rateau.

Provisional Proportions of a Parsons Turbine

The provisional design of a Parsons Turbine is not so simple as a Rateau. Generally each designer has some rules of his own for arriving at the general proportions and most of these rules and methods are kept secret. The methods outlined below, while providing rational designs, may differ widely from methods in use in particular offices. Manufacturers building turbines under Parsons license are furnished considerable confidential data which they are not permitted to make public.

Parsons turbines are usually made with three drums of blades of different diam., known as the H.P. intermediate and L.P. cylinders. The ratio of diameters common in British practice is $1:\sqrt{2}:2$ and the work done in each section is ¼ : ¼ and ½. These ratios are varied considerably in American practice. Shorter turbines are built by making the L.P. do the most work and a ratio of ⅓:⅓:⅓ has been used. Furthermore, to get the maximum amount of steam through a given turbine, blade speeds in the L.P. section

have been increased and drums larger than twice the H.P. diam. have been used for the last rows.

When two-cylinder turbines are built, the H.P. usually has three drums while the L.P is double flow with one drum diam.

The cumulative heat-volume diagram is first drawn for the conditions of operation with the pressures plotted on a logarithmic base. The efficiency ratio can be taken as 0.75 for the average Parsons machine, though higher values should be assumed on the larger sizes. The reheat factor can be found and the condition curve, cumulative heat curve and velocity curve can be plotted.

The cumulative heat curve is divided in proportion to the work done in the cylinders, for example, 1 : 1 : 2 and the pressures at these points found.

Allowance must be made for dummy leakage losses, outside friction, etc. usually about 8 to 12 per cent) and the lbs. of steam per sec. passing through the turbine to carry the given load can then be found.

Instead of calculating the length of blades in each individual stage of a Parsons turbine, as in the case of a Rateau, the blades are arranged in groups of equal length known as "expansions." The number of these expansions is optional at the will of the designer. American practice tends toward three expansions in the H.P., three or four in the intermediate and five to seven in the L.P., though the last of these may employ semi-wing or wing blades When these expansions have been decided upon the cumulative heat may be. divided among them, usually in equal amounts to each expansion.

If the blades are the same length in each expansion, theoretically the discharge angles should vary from the first to the last row. This is not always done in practice. Blades are sometimes arranged in groups of one to four rows of constant angle in each expansion, except on the low pressure, and the ratio of blade speed to steam speed decreases throughout each group, the average ratio for the group being that desired for the cylinder.

The specific volume of the steam used in calculating mean diam. when the ratio of blade speed to steam speed is constant, for the expansion is the middle ordinate on the specific volume curve for that expansion.

The **mean diam. in ft. of the first expansion** can be found by the formula

$$D_1 = \sqrt[3]{\frac{60RWv_1}{\pi^2 zNM \sin \alpha}}$$

where R = ratio blade speed to steam speed $\left(\frac{U}{V_1}\right)$ and

M = thickness coefficient.

W = lbs. of steam per sec. flowing through blades,

v_1 = specific volume at middle ordinate of first expansion,

z = ratio $\frac{\text{blade length}}{\text{mean diameter}} = \frac{l}{12D_1}$

l = blade length in in.

N = R.P.M.

α = discharge angle.

R for high speed turbines varies from 0.5 to 0.8 and for slow speed turbines 0.35 to 0.6. z should not be less than 0.03. In general it does not pay to use blades less than ¾ in. long in the first expansion, otherwise clearance losses are excessive.

When D_1 is found the **blade speed** U can be found. This may vary

from 100 to 130 ft. per sec. on small machines to get sufficient blade height to 150 and 225 ft. per sec. on large machines.

The drum diam. of the H.P. cylinder $D_2 = D_1 - l$.

The L.P. drum diam. can now be determined if the cylinder ratio has been settled. Formerly this blade speed was limited to 400 to 450 ft. per sec. but with improved blade materials and forms of blade fastenings this has been increased to 550 and 650 ft. on units where the last rows of blades are carried on discs of uniform strength instead of on drums as shown in Fig. 197.

The limiting factor as in the impulse turbine is the amount of steam that can pass through the last row without excessive leaving loss. The last blades should not exceed in length $\frac{1}{5}$ of the mean diam. and should preferably be $\frac{1}{6}$. But the steam speeds are lower than in impulse nozzles. Hence the question of sufficient discharge area in the last row of blades is a very important factor in Parsons design.

If this area is too small the steam leaving the last row of blades may reach the critical velocity without being expanded down to the full condenser pressure. Free expansion accompanied by considerable noise and loss then takes place and the equivalent work is lost

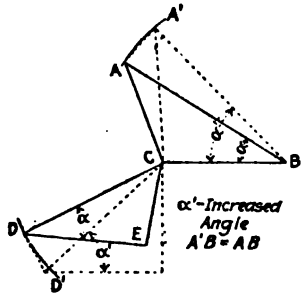


FIG. 40.

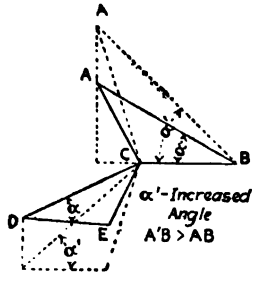


FIG. 41.

The speed of the L.P. blading is limited by the allowable stress in the blading. Hence the limit is soon reached in increasing blade speeds.

A second plan is to increase the discharge angle of the blades, *i.e.*, to use semi-wing and wing blades. This slightly increases the leaving loss. On the other hand, the velocity of the steam passing through the larger area is decreased and the ratio of blade speed to steam speed is increased and this offsets the leaving loss. When this is not sufficient the L.P. end is made double-flow.

It will be apparent that there is a limitation on the amount of vacuum it is profitable to carry at full load. Thus assume a turbine using 10 lbs. of steam per sec. has 32 in. mean diam. on the last row with 6 in. blading, 50 per cent gaging.

The area through the blades $= \pi d \times 6 \times 0.50 = 300$ sq. in.

At L.P. end the critical velocity may be taken as 1250 ft. per sec.

Then the specific volume of steam leaving last row of blades =

$$\frac{1250 \times 300}{144 \times 10} = 250 \text{ cu. ft.}$$

The specific volume at 28-in. vacuum is usually assumed as 300 cu. ft. Hence the vacuum with 250 cu. ft. is about 27¾ in., below which there is no gain from further expansion at full load.

There is a further consideration in regard to the last rows. If the blade angles are increased so that the ratio of blade velocity to steam velocity remains constant, the velocity diagrams change as indicated in Fig. 40, where it is evident that the work done per row and efficiency per row is materially decreased. On the other hand, if it is desired to keep the work per row constant, the steam speed must be allowed to increase and the diagram becomes as shown in Fig. 41.

The heat drop is increased and hence the heat assignment will need to be recalculated, or it may be possible to use less rows. The leaving loss is also increased. The advantage of this construction is that more steam can be passed through the last row of blades with the higher allowable velocity. This allows greater output from the turbine.

The **length of blades** at any expansion can be found from the formula

$$l = \frac{10{,}500\ W v R}{d^2 N M \sin \alpha}$$

where W = lbs. of steam per sec. passing through blades,
v = specific volume of steam at middle ordinate of expansion,
R = mean ratio of blade speed to steam speed for the expansion,
d = mean diam. in in.
N = R.P.M.
M = thickness coefficient.
α = blade discharge angle.

For each cylinder this becomes

$$l = \frac{bv}{d^2} \text{ when } b = \frac{10{,}500\ WR}{NM \sin \alpha}$$

This cannot be solved directly for in Parsons turbines it is general practice to keep the drum diam. constant and increase the length of the blade, thus increasing the mean diam. of the blades. Hence d, the mean diam. in the above, is dependent on l the length of blade. The equation is solved by trial and error by first assuming a length of blade, finding d and solving for l. Then correct d accordingly until the desired result is obtained.

Some designers find only the lengths of the blades of the first and last expansions and assume a geometric progression for the intervening rows.

Thus if l_1 is the blade length of the first expansion and l_m the blade length of the m^{th} expansion

Then $r^{m-1} = \frac{l_m}{l_1}$ gives the geometric ratio.

The intermediate blade lengths are then l_1, rl_1, r^2l_1, etc. When the last two sections of the L.P. are made the same length but with different angles, the ratio r_2 for the L.P. is calculated for the second to last row. Allis-Chalmers place these last two rows of some of their turbines on a larger diam. of drum than the remainder of the L.P. cylinder. This permits higher blade speeds and either shorter blades or smaller discharge angles on the last rows.

The trial and error method is rather tedious for finding the length of the blades in the last expansion of each cylinder. This can be approximated by multiplying the length of the blades in the first expansion by the volume ratio thus

$$l_m = \frac{v_m}{v_1} l_1$$

where l_m = approximate length of blade in the m^{th} expansion, l_1 = blade length first expansion, v_m = specific volume for mid-row of m^{th} expansion and v_1 for mid-row of first expansion.

These approximate lengths should be checked by the formula given above.

When the drum size changes, the length of blades l_a of the first row on the larger drum can be approximated by the equation

$$l_a = \frac{rl_m}{r_d^2}$$

where l_m = length of last row on preceding drum,
r = blade ratio of preceding drum,
r_d = ratio of drum diam.

This must also be checked by the equation for blade length

$$l = \frac{bv}{d^2}$$

Hence l is easily calculated. The blades are usually made in even $\frac{1}{8}$ in. changes of length and sin α is adjusted accordingly.

The blades on the last two expansions of the L.P. cylinder usually have larger angles than the other blades. In order that the same work may be done in each row the necessary blade angle can be found by the equation

$$\tan \alpha_{m+1} = \tan \alpha_m \frac{v_{m+1}}{v_m}$$

where α_{m+1} = desired discharge angle, α_m = normal angle of the blades of the same length when the volume at mid-ordinate is v_m, v_{m+1} = the volume at mid-ordinate of the expansion with increased angle. If the two last expansions have increased angles, the angle of the second of these can be found as follows:

$$\tan \alpha_{m+2} = \tan \alpha_m \frac{v_{m+2}}{v_m}$$

If the work done per stage remains the same, the heat drop per stage must increase and the ratio of blade speed to steam speed decreases. The new ratio is

$$R' = R \frac{\cos \alpha'}{\cos \alpha}$$

where R and α are normal ratios and blade angles and α' is the increased angle. The new steam speed (V_1[1]) must be checked as above noted to see that it does not exceed the critical speed.

Gaging

If it is desired to maintain the same ratio of blade speed to steam speed throughout an expansion, it is necessary to gage the discharge angle from beginning to end. With the blade length fixed and R constant

$$M \sin \alpha = Kv \text{ where } K = \frac{10{,}500\ W}{d^2Nl} R$$

M = thickness coefficient usually 0.75.

v can be found by assigning the same amount of cumulative heat to each row, as in Fig. 42, and finding the corresponding specific volume of discharge from that row.

In practice blades are not gaged by angles but by means of a thickness gage made to fit the required opening between two blades. The size of this thickness gage can be found from the blade designs and depends on the

pitch of blades and the blade thickness. If data is not at hand giving this information, it can be found graphically by drawing series of blades at various angles to large scale. This so-called "gaging" is then expressed on the drawings as a per cent of the full annular opening of the blade ring and this includes the correction for angularity.

Several companies use empirical rules for calculating this gaging based on an increase in the sine of the angle in a geometric ratio from the first to the last group.

Some designers assume that the thickness coefficient at the exit of Parsons blading may be taken as unity in calculating the steam flow on account of the curvature of the back of the blade. This is only approximately true if the

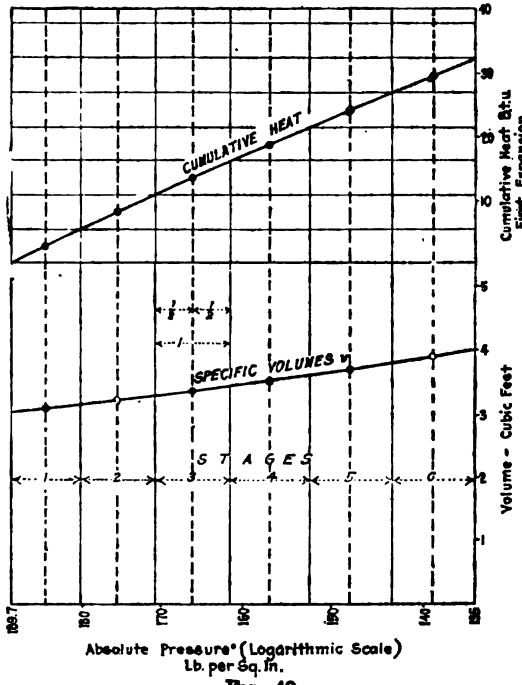


FIG. 42.

angle α for the back of the blades is used. In practice in Parsons design α usually refers to the angle of the blade point which is somewhere between the front and back blade angles. Besides as the gaging tool itself goes between the blades, the thickness of these blades must be considered in making the gage. Hence it is usual to multiply sin α by a constant M which is a thickness coefficient varying from 0.75 to 0.85. In calculating gaging M is usually assumed as 0.75.

The number of rows of blades can be readily calculated. If the blade speed is known and the ratio of blade speed to steam speed fixed, then the steam velocity V_1 can be found.

Since there is the same heat drop in guide blades as in stationary blades, the total heat drop per stage (consisting of one stationary row and one moving row), given on p. 550, will be

$$2\gamma\left(\frac{V_1}{223.7}\right)^2\left(2R\cos - R^2\right) + E_s$$

or the simpler formula given on p. 551 may be used.

The number of stages can be found by dividing the cumulative heat drop for the expansion by this stage heat drop.

It should be noted that as the blade diam. increases on the intermediate and L.P. cylinders, the allowable steam velocity will increase in the same proportion. But the heat drop per row (varying as the square of the velocity) will increase as the square of the ratio of diam. Hence if the cylinder ratios are $1 : \sqrt{2} : 2$ the rows of blades for the same heat drop will be $4 : 2 : 1$. Stepping the turbine considerably shortens the machine.

When double flow construction is used, half the steam volume is assumed to flow in each direction and the blades are designed accordingly.

The drum diam. at all times must be large to maintain the necessary rigidity in the spindle so as to avoid trouble from critical speeds and to keep the turbine length within reasonable bounds.

Provisional Proportions of Direct-coupled Parsons Marine Turbines

The dimensions of these turbines are less readily found than in the preceding cases, as they are modified to suit drum size, weight, etc. The ratio of steam speed to blade speed is much lower than in high speed machines and the efficiency ratio of the units average about 60 per cent.

The same general method above outlined may be applied after construction factors are determined.

Provisional Proportions of Astern Turbines

These are kept as small as possible and efficiency is sacrificed to secure this end. The first wheel is a Curtis two or three row velocity stage. The second wheel may also be a Curtis wheel or in some cases is a simple impulse wheel.

When the marine set consists of a high and low pressure cylinder, there must be a labyrinth gland between the astern section and the forward section in the H.P. cylinder. The astern section is then designed with high and low pressure sections.

The ratio of blade speed to steam speed is usually from 0.5 to 0.55 of that in the forward section. This decreases the efficiency so that with the same amount of steam only from 30 to 60 per cent of the forward power is available for astern, this fraction depending on the particular design.

Provisional Proportions of Curtis-Rateau Turbines

Many builders of Parsons turbines substitute a single Curtis wheel for the H.P. cylinder.

When the pressure drop and heat drop are known or are taken from the power assignment to cylinders, the ratio R of steam speed to blade speed can be assumed and the wheel speed calculated thus

$$V_s = 223.7\sqrt{(H_1 - H_2)} \text{ and } U = RV_s$$

The efficiency of this wheel is less than for the Parsons blading but the leakage over blade tips is avoided and furthermore all the steam entering passes through this wheel instead of 5 per cent or more leaking through the H.P. balance piston.

For provisional proportioning, the efficiencies given by the curves Fig. 27 can be assumed and the heat condition on the stage pressure curve can be found. This is the initial point for the condition curve of the Parsons section, and as it usually falls at or near the saturation line, the condition curve can be drawn straight to the final point. The Parsons section can be designed as above indicated.

TURBINE DESIGN

In considering turbine design the efficiency of the unit plays an important part. Turbines for high efficiency are longer and more expensive machines than those for low efficiency. Small auxiliary units are built as compact and cheaply as possible and therefore have relatively low efficiency.

The **ratio of wheel speed to steam speed R** is an important factor in determining the efficiency of the unit. The selection of this ratio is determined by the permissible size, weight and cost balanced against the economy of operation, and is further influenced by experience and commercial demands. The values of efficiency given in preceding sections may be used as a guide in this connection.

There are probably as many methods of designing turbines as there are designers. Each has empirical rules, plotted curves, diagrams and short cuts in design that he has accumulated from analysis of theoretical considerations and test data. These are generally considered confidential and part of the trade secrets of each concern. The methods to be hereafter described may therefore not coincide with the practice of any concern for merely rational solutions are presented.

Design of a Simple Impulse Turbine

Example. Design a simple impulse geared turbine of 55 H.P., 7000 R.P.M. supplied with dry steam at 125 lbs. gage press., and exhausting at 5 lbs. gage back press.

Total heat at initial conditions,
(139.7 lbs. abs. dry steam)... H_1.....1192
Total heat after adiabatic expansion
to 19.7 lbs. abs..... H_2.....1048
Heat drop $(H_1 - H_2)$..... 144

Assume y for nozzle = 0.06 y = per cent energy loss in the nozzle.

Then **velocity leaving nozzle** $V = 223.7\sqrt{(H_1 - H_2)(1-y)} = 223.7\sqrt{144 \times 0.94} = 2603$ ft. per sec.

Assume nozzle angle 18° and assume a wheel speed of 850 ft. per sec., construct velocity diagram, Fig. 43.

Heat converted into work crossing blade

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_4}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right)^2 + \left(\frac{V_3}{223.7}\right)^2$$

$$= 135.4 - 11.6 - 65.9 + 40.7 = 98.6 \text{ B.t.u.}$$

Hence total heat in the steam leaving blades is $1192 - 98.6 = 1093.4$ B.t.u.
The sp. vol. at that heat and 19.7 lbs. abs. is 18.8 cu. ft.

Wheel speed $U = \frac{\pi DN}{60}$

$\therefore$ mean diameter $= \frac{60U}{\pi N} = \frac{60 \times 850}{3.14 \times 7000} = 2.32$ ft. $= 27.84$ in.

Allowing 12 per cent for friction of bearings, gears, glands and governor drive, the shaft horse power $= \frac{55}{0.88} = 62.5$

To this must be added the friction horsepower due to disc friction and to blade windage.

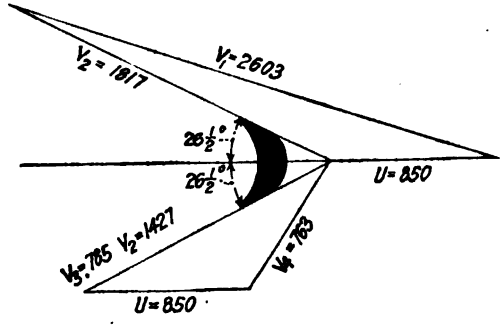


FIG. 43.

Disc friction loss in horsepower

$$\text{H.P.}_d = 0.0607\ D_1^2 \frac{1}{v}\left(\frac{U}{100}\right)^3 = 0.0607 \times (2.32)^2 \times \frac{1}{18.8} \times \left(\frac{850}{100}\right)^3 = 10.65.$$

Assume for first approximation that 0.90 of the circumference of the blade ring is not receiving steam, and $\frac{3}{4}$ in. blades.

H.P. lost from blade friction

$$\text{H.P.}_b = 0.458\ B \times S \times D \times l^{1.5} \times \frac{1}{v}\left(\frac{U}{100}\right)^3$$

$$= 0.458 \times 1 \times 0.9 \times 2.32 \times (0.75)^{1.5} \times \frac{1}{18.8} \times \left(\frac{850}{100}\right)^3 = 20.3$$

Hence total H.P. at blades $= 62.5 + 10.65 + 20.3 = 93.45$

and lbs. of steam per hour $= \frac{93.45 \times 2546.5}{98.6} = 2412$ lbs. per hr.

Lb. of steam per sec. $= \frac{2412}{3600} = 0.67$ lb.

Total heat at nozzle mouth $= 1048 + 0.06 \times 144 = 1056.6$ B.t.u.
and sp. vol. $= 18$ cu. ft. per lb.

$\therefore$ necessary mouth area in sq. in. $= \frac{144 \times 18 \times 0.67}{2603} = 0.667$ sq. in.

In two nozzles this provides 0.333 sq. in. per nozzle.
This corresponds to a diameter of 0.65 in.
Pressure at throat $= 0.58 P_1 = 0.58 \times 139.7 = 81$ lbs.
Specific volume $= 5.2$ cu. ft.
B.t.u. at 81 lbs. after adiabatic expansion $= 1148$
$\therefore H_1 - H_2 = 44$ B.t.u.
and throat velocity $V_2 = 223.7\sqrt{44} = 1485$ ft. per sec.

Necessary throat area $= \frac{144 \times 5.2 \times 0.67}{1485} = 0.338$ sq. in.

or 0.169 sq. in. per nozzle corresponding to 0.464 in. diameter. Hence nozzle is as per Fig. 44.

$$\text{Length } L = \frac{0.65 - 0.464}{2 \tan 6°} = \frac{0.186}{2 \times 0.105} = 0.875 \text{ in. or } \tfrac{7}{8} \text{ in.}$$

$$\text{Lbs. of steam per B.H.P.} = \frac{2412}{55} = 43.85$$

$$\therefore \text{ overall efficiency ratio} = \frac{2546.5}{43.85 \times 144} = 40.3 \text{ per cent}$$

The blade can be laid out as shown on p. 615 and the pitch found. If this pitch does not give a whole number of blades it can be altered to do so.

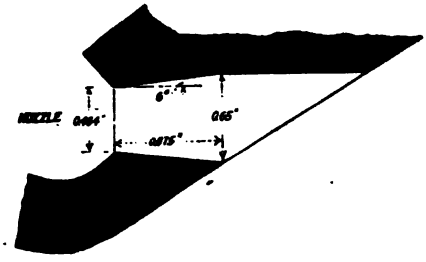


FIG. 44.

Design of Simple Curtis Turbine

Example. Design a simple Curtis turbine of 50 B.H.P., 3600 R.P.M. to operate in place of the preceding simple impulse turbine under same steam conditions.

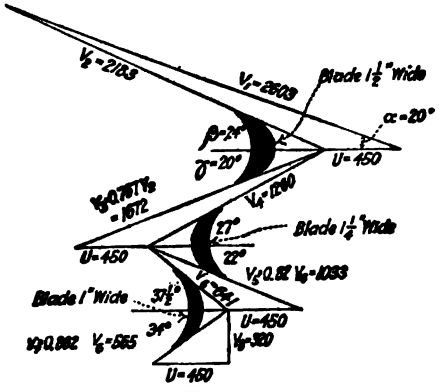


FIG. 45.

Let wheel speed be 450 ft. per sec. and nozzle angle 20 deg. Actual nozzle velocity as before = 2603 ft. per sec.

The velocity diagram is shown in Fig. 45.

Unequal blade angles are used throughout. The outlet blade angle is closed in a number of degrees in every case. Heat converted into work in crossing blade

$$= \left(\frac{V_1}{223.7}\right)^2 - \left(\frac{V_2}{223.7}\right)^2 - \left\{\left(\frac{V_2}{223.7}\right)^2 - \left(\frac{V_3}{223.7}\right)^2\right\} - \left\{\left(\frac{V_4}{223.7}\right)^2 - \left(\frac{V_5}{223.7}\right)^2\right\} - \left\{\left(\frac{V_6}{223.7}\right)^2 - \left(\frac{V_7}{223.7}\right)^2\right\}$$

$$= 135.4 - 2.0 - (95.2 - 56.1) - (31.7 - 21.3) - (8.2 - 6.4) = 82.1 \text{ B.t.u.}$$

$$\text{Wheel diameter} = \frac{60U}{\pi N} = \frac{60 \times 450}{3.14 \times 3600} = 2.387 \text{ ft.} = 28.65 \text{ in.}$$

Total heat leaving blades = 1192 − 82.1 = 1009.9 B.t.u.
and sp. vol. at 19.7 lbs. and 1109.9 B.t.u. = 19.2 cu. ft.
Allowing 4 per cent for friction of bearings, governors, etc., the S.H.P. = $\frac{50}{0.96} = 52$.

Horsepower friction loss of disc

$$\text{H.P.}_d = 0.0607\, D_1^2 \frac{1}{v}\left(\frac{U}{100}\right)^3 = 0.0607 \times (2.387)^2 \times \frac{1}{19.2} \times \left(\frac{450}{100}\right)^3 = 1.67 \text{ H.P.}$$

Assume mean height of blades of 1¼ in. in two rows and that 90 per cent of circumference does not receive steam,

$$\text{H.P. lost in blade friction} = 0.458\, BSDl^{1.5} \times \frac{1}{v}\left(\frac{U}{100}\right)$$

$$= 0.458 \times 1.23 \times 0.9 \times 2.387 \times (1.25)^{1.5} \times \frac{1}{19.2} \times \left(\frac{450}{100}\right)^3 = 8.0$$

$\therefore$ total H.P. at blades = 52 + 8 + 1.67 = 61.67

$$\text{and lbs. of steam per hour} = \frac{61.67 \times 2546.5}{82.1} = 1913 \text{ lbs.}$$

$$\text{Lb. of steam per sec.} = \frac{1913}{3600} = 0.531 \text{ lb.}$$

Sp. vol. at nozzle mouth = 18 cu. ft. per lb.

$$\therefore \text{ mouth area} = \frac{144 \times 18 \times 0.531}{2603} = 0.529 \text{ sq in.}$$

In two nozzles mouth area = 0.2645 sq. in. and diam. = 0.580 in.
With 5.24 cu. ft. as sp. vol. at throat

$$\text{necessary throat area} = \frac{144 \times 5.24 \times 0.531}{1485} = 0.268 \text{ sq. in.}$$

Throat area per nozzle = 0.134 sq. in. and diam. = 0.414 in.

The volume of the steam during its passage across the blades can be assumed to remain constant, but the velocity decreases; hence the outlet area of the blades must increase.

The mean diameter remains constant so that the blade height must be increased at outlet. This outlet height can be found from the formula

$$\text{Outlet height} = \frac{V_1 \sin \alpha}{V_2 \sin \gamma} \times \text{nozzle height.}$$

For first blades, outlet height $= \frac{2603 \sin 20°}{1676 \sin 20°} \times 0.58 = 0.9$ in.

For intermediate guide blades, outlet height $= \frac{2603 \sin 20}{1033 \sin 22} \times 0.58.$

$$= \frac{2603 \times 0.342}{1033 \times 0.374} \times 0.58 = 1.335 \text{ in.}$$

For 2d moving row outlet height $= \frac{2603 \sin 20°}{565 \sin 34} \times 0.58$ in.

$$= \frac{2503 \times 0.342}{565 \times 0.559} \times 0.58 = 1.635 \text{ in.}$$

The inlet area varies depending on whether a sloping or flat shroud is used. In any case the inlet should be $\frac{1}{16}$ to $\frac{3}{16}$ in. longer than the preceding discharge length. Assuming this increase to be about $\frac{1}{8}$ in., the inlet lengths of the blades are 0.7 in., 1.02 in. and 1.46 in. The blades can now be laid out and in elevation will appear as in Fig. 46.

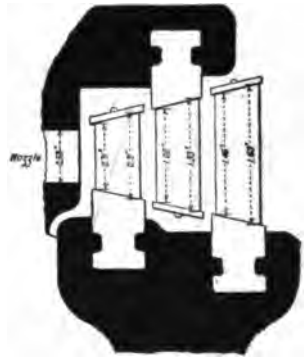


FIG. 46.—Curtis Stage.

Lbs. of steam per B.H.P. hr. $= \frac{1913}{50} = 38.26$

$\therefore$ overall efficiency ratio $= \frac{2546.5}{38.26 \times 144} =$ 44.4 per cent

Design of a Rateau Turbine

Example. Required the provisional dimensions of a 3000 B.H.P. Rateau turbine at 3600 R.P.M. Initial conditions 200 lbs. gage, 100 deg. F. superheat and 28 in. vacuum.

Assume a ratio R of blade speed to steam speed of 0.42 and let blade speed be 575 ft. per sec.

$$\text{Disc diam. } d = 229 \frac{U}{N} = 229 \times \frac{575}{3600} = 36.5 \text{ in.}$$

$$V_s = \frac{U}{R} = \frac{575}{0.42} = 1369 \text{ ft. per sec.}$$

$$H_s = \left(\frac{V_s}{223.7}\right)^2 = \left(\frac{1369}{223.7}\right)^2 = 37.2 \text{ B.t.u.}$$

Total heat at 214.7 lbs. abs., 100 deg. F. superheat	= 1259
Total heat after adiabatic expansion to 28 in. vacuum	= 896
$(H_1 - H_2)$	= 363

Fig. 27 gives stage blade efficiency of 79 per cent with $R = 0.42$. These curves refer to the L.P. section of a Rateau turbine. Allow 5 per cent for the increased friction, ventilation and leakage losses in the H.P. section. It will probably be found that the first few stages only have partial admission; hence the windage losses will be high. Then the average stage blade efficiency = 74 per cent.

The nozzle friction will amount to 6 per cent giving a net stage efficiency of 68 per cent. The steam for external glands, bearing friction, thrust fric-

tion and leaving losses will amount to another 6.4 per cent. The reheat factor F from Fig. 12 for 68 per cent efficiency is 1.05. Hence the net efficiency ratio = (68 × 1.05 − 6.4) = 65 per cent.

Then the number of stages

$$n = \frac{F\ (H_1 - H_2)}{37.2} = \frac{1.05 \times 363}{37.2} = 10.25.$$

Use 10 stages and allow 39 B.t.u. heat drop in the first stage and 38 B.t.u. in the succeeding stages. In actual designs of Rateau turbines better results

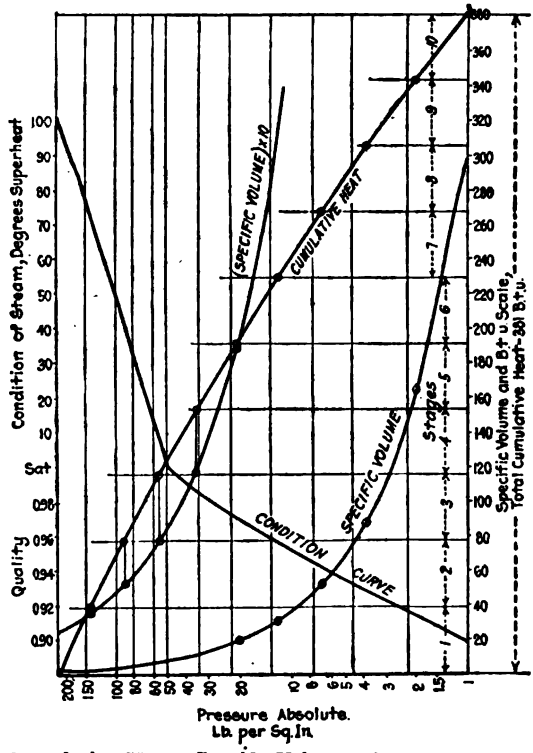


FIG. 47.—Cumulative Heat—Specific Volume Diagram for Rateau Turbine.

will be obtained with reduced diameters of blade wheels in the first stages on account of diminished friction losses in the denser steam. Lbs. of steam per shaft horsepower hour

$$w = \frac{2546.5}{E_n(H_1 - H_2)} = \frac{2546.5}{0.65 \times 363} = 10.8 \text{ lbs.}$$

and lbs. of steam per sec., $W = \frac{10.8 \times 3000}{3600} = 9$ lbs.

Table 4. Condition Curve, Cumulative Heat and Volumes

Pressure, abs.	Superheat or quality on condition curve	Specific volume v	Total heat on condition curve	Heat drop $(H_1 - H_2)$	Cumulative heat = $\frac{\text{heat drop}}{0.68}$
214.70	100 deg.	2.52	1259	...	....
150.00	73 deg.	3.38	1235	24	35.3
100.00	47 deg.	4.79	1212	47	69.1
80.00	32 deg.	5.77	1199	60	88.1
51.00	saturation	8.35	1174	85	125.0
34.00	0.986	12.20	1154	105	154.4
22.00	0.973	18.00	1132	127	186.8
13.86	0.959	28.80	1108	149	219.0
7.50	0.945	47.80	1084	175	257.3
4.50	0.933	76.00	1061	198	295.0
2.20	0.917	146.00	1032	227	339.0
0.96	0.899	305.00	1000	259	381.0

Table 5. Stage Pressure and Specific Volumes

Stage	Absolute pressure	Specific volume	Stage	Absolute pressure	Specific volume
1	142	3.5	6	12.00	31.5
2	92	5.1	7	6.80	52.0
3	57	7.7	8	3.75	90.0
4	35	11.8	9	2.00	163.0
5	21	18.8	10	0.96	305.0

The condition curve was plotted assuming 68 per cent internal efficiency (stage efficiency-nozzle loss) or assuming that (0.68 × 381) or 259 B.t.u. were turned into work. Data for plotting this curve and also the cumulative heat and volume curves are given in Table 4, and are plotted in Fig. 47.

The heat assignment per stage was set off and the pressure at the end of each stage was determined by the intersection of the cumulative heat curve and a horisontal line drawn through the heat assignment point. These stage pressures and the corresponding volumes are given in Table 5.

Assuming an orifice friction loss of 6 per cent average for the whole turbine, the velocity from the first stage nozzle will be

$$V_1 = 223.7\sqrt{39 \times 0.94} = 1355 \text{ ft. per sec.}$$

Considering the residual velocity from the preceding stage, little error is made in assuming that this nozzle velocity remains 1355 ft. per sec. from all orifices. The final calculations should take careful account of the residual velocities. The orifice plates must be as carefully designed as the stationary blades in a Curtis stage.

Assume for trial that the orifice angle of the last stage is 20 deg., the radial length of this orifice, assuming s, the opening ratio, = 0.85, will be

$$l' = \frac{Wv_e}{\pi s D V_1 \sin \alpha} = \frac{9 \times 305}{3.14 \times 0.85 \times \frac{36.5}{12} \times 1355 \times 0.342} = 0.729 \text{ ft.} = 8.75 \text{ in.}$$

The limiting length of the blade is ⅛ of the mean diam., or say 7¼ in. The nozzle radial length will therefore be 1/16 in. less or 7.1875 in. As it is not desirable to increase the wheel diam. the nozzle angle is increased. The

new angle α' can be found from the equation, $\sin \alpha' = \frac{l^1}{l} \sin \alpha = \frac{8.75}{7.1875} \times 0.342 = 0.416$.

Hence the new angle $\alpha' = 24° - 30'$ about. If a velocity diagram were drawn it would be found that the efficiency of this stage was slightly lessened, and a correction would have to be made for the same. This would not materially affect the design.

If the discharge angle of the other orifices is retained at 20 deg., then the radial length of these orifices can be found by multiplying the calculated length of the last stage by the ratio of their specific volumes.

For the ninth stage $l = 8.75 \times \frac{163}{305} = 4.67$ in.

For the eighth stage $l = 8.75 \times \frac{90}{305} = 2.58$ in.

For the seventh stage $l = 8.75 \times \frac{52}{305} = 1.49$ in.

For the sixth stage $l = 8.75 \times \frac{31.5}{305} = 0.904$ in.

If $\frac{3}{4}$ in. blades are the shortest that will be used, then the nozzle outlet should not be less than $1\frac{1}{16}$ in. or 0.6875 in. Hence only partial peripheral admission can be used on the first six stages. The circumferential length of each of the two groups of orifices can now be found.

$$K = \frac{Wv_o}{2sl\,V_1 \sin \alpha} = av_o.$$

For the sixth stage, $a = \frac{9 \times 144}{2 \times 0.85 \times 0.687 \times 1355 \times 0.342} = 2.39$.

$\therefore K = 2.39 \times 22 = 52.6$ in. for sixth stage.

For fifth stage $K = 2.39 \times 18.8 = 44.9$ in.

For fourth stage $K = 2.39 \times 11.8 = 28.2$ in.

For third stage $K = 2.39 \times 7.7 = 18.4$ in.

For second stage $K = 2.39 \times 5.1 = 12.2$ in.

For first stage $K = 2.39 \times 3.5 = 8.37$ in.

The orifices can be designed as shown on p. 542.

This completes the provisional design. The stages can now be gone over one by one and recalculated by the use of velocity diagrams, etc., as in the preceding section and such adjustments may be made as are deemed necessary. The heat drop and volume of each stage should now be calculated step by step. This affords a complete check on the provisional design.

Design of a Curtis-Rateau Turbine

The turbine is generally designed as a pure Rateau, for the limiting features are the length of L.P. blades and the blade speeds of these wheels. Then a Curtis wheel is substituted for a certain number of Rateau wheels and adjustments made in the heat assignments if necessary to suit the new conditions. Another method is to assume that

the Curtis wheel utilizes three to four times the heat drop of the same diameter Rateau wheel. The heat distribution is made on that basis and the design continued as before.

The blade efficiency of the Curtis wheel is somewhat lower than that of the Rateau. On the other hand the whirling and leakage losses will be less, for with the Curtis wheel and the large pressure drop in the nozzles the density of the steam in the first stage is also much less. Frequently the Curtis stage is designed for a smaller diameter than the remaining Rateau stages.

Data from previous designs will have to be studied to note the effect on efficiency of adding the Curtis wheel. It is largely dependent on the relative friction losses of the two types of wheel viz., Curtis and Rateau. For provisional calculations the efficiency ratio may be assumed the same as for a simple Rateau.

Example. Suppose the first three stages on the Rateau turbine just calculated are replaced by a Curtis wheel. The heat drop of this stage will be $39 + 2 \times 38 = 115$ B.t.u. and the theoretical steam velocity $V_0 = 223.7\sqrt{115} = 2400$ ft. per sec.

For best efficiency, the ratio R of blade speed to steam speed should be 0.225. As this is a H.P. stage, a ratio of 0.21 will be assumed to allow for friction.

hence $U = 2400 \times 0.21 = 504$ ft. per sec.

The wheel diameter, $d = 229\dfrac{U}{N} = 229 \times \dfrac{504}{3600} = 32$ in.

The expanding nozzles and blade lengths can be designed by the methods shown in the preceding sections. The turbine now consists of one Curtis stage and seven Rateau stages.

Some designs use higher wheel speeds and lower values of R. Hence a much shorter and lighter machine can be designed at a slight decrease in efficiency. Weight, size and cost can be materially reduced by increasing wheel speeds and decreasing velocity ratio-R.

Design of a Parsons Turbine

Example. Find the provisional dimensions of a 2500 B.H.P. Parsons turbine to run at 3000 R.P.M. Initial conditions 175 lbs. gage press., 150 deg. F. superheat and 28 in. vacuum.

Assume drum diam. ratios $1 : \sqrt{2} : 2$ and an internal efficiency ratio of 75 per cent. Allow 9 per cent for dummy leakage, gland losses, bearing friction and other losses.

Hence net efficiency ratio = 66 per cent.

B.t.u. available from adiabatic expansion from 189.7 lbs. abs. and 150 deg. superheat to 28 in. vacuum $(H_1 - H_2) = 1281 - 916 = 365$ B.t.u.

Heat turned to work $= 0.75 \times 365 = 274$ B.t.u.

$\therefore$ Final heat at 2 in. abs. $= 1281 - 274 = 1008$ B.t.u.

The reheat factor for an interval efficiency of 75 per cent and 150 deg. F. superheat is 1.045.

$\therefore$ Total cumulative heat $= 365 \times 1.045 = 381$ B.t.u. and the stage efficiency $= \dfrac{0.75}{1.045} = 71.8$ per cent.

The lbs. of steam per B.H.P., $w = \dfrac{2546.5}{E(H_1 - H_2)} = \dfrac{2546.5}{0.66 \times 365} = 10.56$ lbs.

Lbs. of steam per sec. $= \dfrac{2500 \times 10.56}{3600} = 7.33$ lbs.

To plot the cumulative heat curve and velocity curve, the increase in entropy per 100 B.t.u. heat drop, $\phi_s = 0.055 + 0.8/^2 - 0.00006\ (p_1 + t_s) = 0.055 + 0.8(1 - 0.718)^2 - 0.00006(189.7 + 150) = 0.0525$.

This is laid off on the heat-entropy diagram and the condition curve located, as explained on p. 537. The cumulative heat-volume diagram is laid out in Fig. 48, from data in Table 6 with the pressures plotted on a logarithmic base.

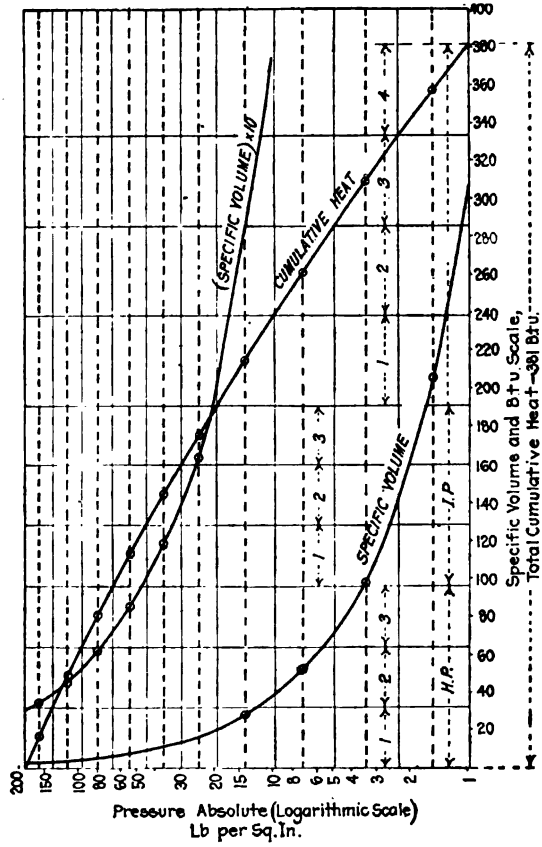


FIG. 48.—Cumulative Heat-specific Volume Diagram for Parsons Turbine.

Table 6

Pressure	Superheat or quality	Spec. vol. v	Total heat	Heat drop $(H_1 - H_2)$	Cumulative heat $\frac{(H_1 - H_2)}{0.718}$
189.70	150°	3.02	1281	...	.....
150.00	129	3.64	1264	17	23.6
110.00	101	4.70	1243	38	53.0
80.00	79	6.18	1223	58	80.8
50.00	41	9.10	1194	87	121.0
36.00	18	12.00	1176	105	146.0
28.00	Sat.	14.80	1163	118	164.0
18.00	0.986	21.80	1141	140	195.0
10 00	0.967	37.00	1111	170	237.0
6.00	0.953	59.20	1086	195	271.0
3.60	0.939	94.00	1063	218	304.0
2.20	0.927	148.00	1042	239	333.0
0.98	0.907	307.00	1008	273	381.0

From this diagram the press. at the beginning of the second and third cylinders are 66 and 19.5 lbs. abs. respectively.

Assume three expansions on the H.P. drum, three on the intermediate, and four on the low. The cumulative heat assignments and also the sp. vol. at mid-expansion can be found from the diagram.

Assume R for blades = 0.6 and z = 0.05, also that the H.P. balance pistons allow 5 per cent leakage. Then steam flowing through blade section of H.P. cylinder will be,

$$W_1 = 0.95 \times 7.33 = 6.965 \text{ lbs.}$$

The sp. vol. at mid-ordinates of the various expansions is given below in Table 7. The mid-ordinate is at the middle point of the length subtended by the expansion on the pressure scale of the cumulative heat diagram since pressures are plotted on a logarithmic base.

Table 7

	H. P.			I. P.			L.P.			
Expansion	1	2	3	1	2	3	1	2	3	4
Specific volume v	3.46	4.5	6.1	8.5	11.75	16.5	28	52	100	206.5

Let the outlet angle α of the blades on the mean diameter be 20 deg. at the mid-ordinate of each expansion and let M, the thickness coefficient for the blades be 0.75 for normal blading.

The mean diameter of the H.P. cylinder is

$$D_1 = \sqrt[3]{\frac{60RW_1v_1}{\pi^2 z N M \sin\alpha}} = \sqrt[3]{\frac{60 \times 0.6 \times 6.955 \times 3.45}{3.14 \times 3.14 \times 0.05 \times 3000 \times 0.75 \times 0.342}}$$

$$= 1.315 \text{ ft.} = 15.78 \text{ in.}$$

and blade length = $15.78 \times 0.05 = 0.789$ in. Assume 0.78 in. blades.

Hence drum diameter of the H.P. cylinder = $15.78 - 0.78 = 15$ in.

The drum diameter of the intermediate cylinder is therefore

$$15 \times \sqrt{2} = 21.21 \text{ in.}$$

and of the L.P. $15 \times 2 = 30$ in.

The approximate blade length of the third expansion on the first cylinder is

$$l_3 = \frac{v_3}{v_1} l_1 = \frac{6.1}{3.46} \times 0.78 = 1.375 \text{ in.}$$

Try blades 1.31 in. long.

Mean diameter = $15 + 1.31 = 16.31$ in.

Checking by formula $l_3 = \frac{b_1 v_3}{d_3^2}$ $\quad b_1 = \frac{10{,}500 W_1 R}{NM \sin\alpha} = \frac{10{,}500 \times 6.955 \times 0.6}{3000 \times 0.75 \times 0.342} = 56.94$

$$l_3 = \frac{56.94 \times 6.1}{16.31 \times 16.31} = 1.305 \text{ in.}$$

Hence blades 1.31 in. long would be used.

Let r_1 be geometric ratio of blade lengths.

$$r_1^2 = \frac{l_3}{l_1} = \frac{1.31}{0.78} = 1.678. \qquad r_1 = \sqrt{1.678} = 1.296.$$

$\therefore l_2 = 0.78 \times 1.296 = 1.01$ in. blades in the second barrel.

The blade lengths l_T on the first cylinder calculated for leakless flow are thus: 0.78 in., 1.01 in. and 1.31 in.

These heights are correct only if there is no leakage over the tips of the blades. For the shorter blades particularly, a leakage correction should be made to the heights.

The clearance can be found from the clearance formula.

$$C = 0.015 + 0.003\,D + 0.005 l_T$$

where C = the radial clearance in in.

D = mean diam. in ft.

l_T = calculated blade length for leakless flow.

The clearances for the three barrels as thus calculated are: 0.023 in.; 0.024 in. and 0.026 in.

The active blade length l_N can be found from the following formula:

$$l_N = g l_T$$

where g = leakage coefficient.

$$g = \frac{l_T M \sin \alpha - C}{l_T M \sin \alpha}$$

where M = thickness coefficient (in this case 0.75).

Hence for the first row

$$g = \frac{0.78 \times 0.75 \times 0.342 - 0.023}{0.78 \times 0.75 \times 0.342} = 0.872$$

$$l_N = 0.872 \times 0.78 = 0.690 \text{ in.}$$

The leakage coefficients for the second and third expansions are 0.908 and 0.922.

The active blade lengths for the second and third barrels calculated in the same manner are found to be

$$l_{N2} = 0.917 \text{ in. and } l_{N3} = 1.208 \text{ in}$$

Intermediate Cylinder

Assume a dummy leakage of 4 per cent on the intermediate dummy piston. Then $W_2 = 0.96 \times 7.33 = 7.035$ lbs. per sec. and $b_2 = \frac{10{,}500\, W_2 R}{N M \sin \alpha} = \frac{10{,}500 \times 7.035 \times 0.6}{3000 \times 0.75 \times 0.342}$

$= 57.6$

The length of the first row in this stage can be approximated by multiplying the last row on the preceding stage by the blade length ratio for that stage and dividing by the square of the ratio of drum diameters

$$l_{a_1} = \frac{r_1 l_3}{\left(\frac{D_2}{D_1}\right)^2} = \frac{1.296 \times 1.31}{2} = 0.85 \text{ in.}$$

Mean diam. = 21.21 + 0.85 = 22.06 in.

$$l_{a_1} = \frac{57.6 \times 8.5}{22.06 \times 22.06} = 0.996$$

A recalculation gives necessary blade length $l_{a_1} = 0.99$ in.

Approximate length of last row

$$l_{a_2} = l_{a_1} \times \frac{v_2}{v_1} = 0.99 \times \frac{16.5}{8.5} = 1.921 \text{ in.}$$

Mean diam. = 21.21 + 1.92 = 23.13 in.

$$l_{a_2} = \frac{57.6 \times 16.5}{23.13 \times 23.13} = 1.775 \text{ in.}$$

A recalculation gives necessary blade length $l_{a_2} = 1.795$ in.

$$r_2^2 = \frac{l_{a_2}}{l_{a_1}} = \frac{1.795}{0.99} = 1.813 \qquad r_2 = 1.346$$

Hence length of 2d row

$$l_{a_2} = 0.99 \times 1.346 = 1.332 \text{ in.}$$

The blade heights l_T on the second cylinder calculated for leakless flow are thus: 0.99 in., 1.332 in. and 1.795 in.

The clearance for the first barrel will be

$$C = 0.015 + 0.003D + 0.005 l_T = 0.015 + 0.003 \times \frac{22.2}{12} + 0.005 \times 0.99 = 0.026 \text{ in.}$$

For the other two barrels the clearances are 0.028 in. and 0.030 in. The active blade

length l_N allowing for loss from leakage through clearance on the first expansion would then be

$l_N = \rho l_T$ where l_T = length for leakless flow and ρ = leakage coefficient.

$$\rho = \frac{l_T M \sin\alpha - C}{l_T M \sin\alpha} = \frac{0.99 \times 0.75 \times 0.342 - 0.026}{0.99 \times 0.75 \times 0.342} = 0.898$$

$$l_N = 0.991 \times 0.898 = 0.889 \text{ in.}$$

The leakage coefficients for the other two expansions are 0.911 and 0.935.

The actual blade lengths for the other two barrels are 1.213 in. and 1.678 in.

Low Pressure Cylinder

Assume a dummy leakage of 3 per cent in the L.P. balance piston. Then $W_3 =$ $0.97 \times 7.33 = 7.11$ lbs. per sec. and $b_3 = \frac{10{,}500\, W_3 R}{N M \sin\alpha} = \frac{10{,}500 \times 7.33 \times 0.6}{3000 \times 0.75 \times 342} = 58.2$

The heat assignment per expansion for the third cylinder is nearly twice that for each expansion on the second cylinder, and is therefore nearly equivalent to two expansions. Hence, to find the approximate blade length under these conditions use r_2^2 in the approximate formula for length of the first row.

The approximate blade length of the first barrel

$$l_{e_1} = \frac{r_2^2 l_{a3}}{\left(\frac{D_3}{D_2}\right)^2} = \frac{1.346 \times 1.346 \times 1.795}{2}$$

$$= 1.626$$

Mean drum diameter = 30 + 1.626 = 31.626 in.

$$l_{e_1} = \frac{58.2 \times 28}{21.626 \times 31.626} = 1.628 \text{ in.}$$

A recalculation gives $l_{e_1} = 1.627$ in.

Approximate length of last expansion: Try 8.25 in. blade, mean diam. = 38.25 in.

$l_{a4} = \frac{58.2 \times 206.5}{38.25 \times 38.25} = 8.22.$

A recalculation gives this length as 8.23 in., but the blade length should be kept to within ⅛ of the drum diam. and therefore to 6 in. in length. The blade angles must be adjusted accordingly.

To get the blade ratio of this cylinder

$$r_2^2 = \frac{l_{e_4}}{l_{e_1}} = \frac{8.23}{1.627} = 5.055 \qquad r_2 = 1.716$$

Length of 2d expansion

$$l_{e_2} = 1.627 \times 1.716 = 2.792 \text{ in.}$$

Mean diameter = 30 + 2.792 = 32.792 in.

$$\text{and } l_{e_2} = \frac{58.2 \times 52}{32.792 \times 32.792} = 2.812 \text{ in}$$

A recalculation gives l_{e_2} as 2.81 in.

Length of 3d expansion $l_{e_3} = 2.81 \times 1.716 = 4.82$

Mean diam. = 30 + 4.82 = 34.82 in.

$$l_{e_3} = \frac{58.2 \times 100}{34.82 \times 34.82} = 4.8 \text{ in.}$$

Since the last row must not exceed 6 in. in height, the discharge angle of the mid-ordinate of this expansion must be increased. This angle can be found by calculation. M for wing blades may be taken as 0.85

$$\sin\alpha_4 = \frac{10{,}500\, W R v_4}{d^2 N M l_{e_4}} = \frac{10{,}500 \times 7.11 \times 0.6 \times 206.5}{36 \times 36 \times 3000 \times 0.85 \times 6} = 0.4666$$

$\alpha_4 = 28°$ approx.

The blade lengths l_T of the third cylinder calculated for leakless flow are: 1.627 in., 2.81 in., 4.8 in. and 6 in.

The clearance in the first expansion will be

$$C = 0.015 + 0.003 \times \frac{31.627}{12} + 0.005 \times 1.627 = 0.031 \text{ in.}$$

For the other barrels the clearances are 0.37 in., 0.048 and 0.054 in.

The active blade length l_N can now be calculated, allowing for leakage through clearance.

$$g = \frac{1.627 \times 0.75 \times 0.342 - 0.031}{1.627 \times 0.75 \times 0.342} = 0.927$$

$$l_N = 0.927 \times 1.627 = 1.507$$

The leakage coefficients for the 2d and 3d expansions are 0.949 and 0.962.

The active blade lengths for the 2d and 3d expansions are 2.665 in. and 4.615 in.

If the blade length of the final expansion were corrected in the same way the blade length would reduce to 5.89 in. However, it is more probable that the blade length would be kept constant at 6 in. and the blade angle decreased. If the blade angle were reduced to 26° 45′ with 6 in. blades, the leakage loss would be allowed for. The active blade lengths l_N for the third cylinder are therefore 1.507 in., 2.665 in., 4.615 in. and 6 in.

The blade lengths for the whole turbine may now be tabulated. These lengths as calculated may be used or the blades may be taken to the nearest ⅛ in. and the gaging corrected accordingly. This is usually done as in Table 8

Table 8

	High press.			Intermediate press.			Low press.			
Expansion	1	2	3	1	2	3	1	2	3	4
Active length (calculated)	0.690	0.917	1.208	0.889	1.213	1.678	1.507	2.665	4.615	6
Active length to nearest ⅛ in.	⅞	1	1⅛	⅞	1⅛	1⅝	1½	2⅝	4⅝	6

To calculate the number of rows.

The blade speed for the first expansion

$$U = \frac{dN}{229} = \frac{3000}{229} \times d = 13.1d = 206.3 \text{ ft. per sec.}$$

where d = mean diam. in in.

$$\therefore V_1 = \frac{U}{R} = \frac{206.3}{0.6} = 343.8 \text{ ft. per sec.}$$

The heat assignment per pair of rows assuming a nozzle efficiency of 90 per cent is

$$\frac{2\left(\frac{V_1}{223.7}\right)^2}{\text{nozzle efficiency}} = \frac{2 \times \left(\frac{343.8}{223.7}\right)}{0.9} = 5.24 \text{ B.t.u.}$$

The number of stages (consisting of a stationary and a moving row) on the first expansion

$$= \frac{32}{5.24} = 6.11, \text{ say 6 stages.}$$

The remaining expansions in the turbine are calculated in the same way and are given in Table 9.

This table gives 18 rows in the H.P. cylinder, 9 rows in the I.P. and 8 rows in the L.P.

Table 9

	High pressure			Intermediate pressure			Low pressure			
Expansion	1	2	3	1	2	3	1	2	3	4
Length of blade, in.	¾	1	1¼	⅞	1¼	1⅝	1½	2⅝	4⅝	6
Drum. diam., in.	15	15	15	21¼	21¼	21¼	30	30	30	30
Mean blade ring diam., in. (d)	15¾	16	16¼	22⅛	22½	22⅞	31½	32⅝	34⅝	36
Blade speed $U = 13.1d$	206.30	209.60	212.80	289.70	294.70	299.70	409.50	427.40	453.60	471.60
Ratio $\frac{\text{Blade speed}}{\text{steam speed}} = R$	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60
Steam speed $V = \frac{U}{R}$	343.80	349.40	354.60	483.00	491.30	499.50	688.00	712.20	756.00	786.20
Nozzle efficiency y of blades	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.94
Heat required per stage B.t.u $= \frac{2\times\left(\frac{V}{223.7}\right)}{y^2}$	5.24	5.42	5.58	10.35	10.71	11.08	21.00	22.50	25.40	26.30
Heat assignment per expansion	32.00	32.00	32.00	31.60	31.60	31.80	47.50	47.50	47.50	47.50
Approx. no. of stages	6.11	5.90	5.74	3.05	2.95	2.87	2.26	2.11	1.87	1.80
Actual no. of stages	6	6	6	3	3	3	2	2	2	2

With the number of rows fixed, the gaging can be found. Many manufacturers now gage each row separately. The gaging of the first expansion only will be calculated. Fig. 42 represents the heat-volume diagram. The angle is calculated for the specific volume at the mid-ordinate of the stage, as shown in Fig. 42, p. 571.

$$M \sin \alpha = Kv \text{ or } \sin \alpha = \frac{Kv}{M}$$

where M = thickness coefficient = 0.75 for first expansion,
v = sp. vol. at mid-ordinate.

$$K = \frac{10{,}500\ WR}{d^2Nl}$$

where W = lbs. of steam per sec. flowing through blades.

R = ratio wheel speed to steam speed, d = diam. in in., l = blade length in in., N = R.P.M., K for first expansion $= \dfrac{10{,}500 \times 6.965 \times 0.6}{15.75 \times 15.75 \times 0.75 \times 3000} = 0.0786$

Gaging is usually expressed as "per cent gaging." If the value of "per cent gaging" is known, then the opening between the blades for inserting the gaging tool can be found by dividing the annular area between spindle and cylinder by the number of blades per row and multiplying by the "per cent gaging."

Per cent gaging $= Kvg \times 100$.

For preliminary calculations the leakage coefficient g may be assumed constant. Later on the gaging can be corrected for the slight change in this factor if deemed necessary. Table 10 gives the gaging calculations for the first barrel.

Table 10. Gaging First Expansion

Stages	Specific volume mid-ordinate v	Leakage coefficient g	Per cent gaging $Kvg \times 100$	$\sin \alpha = \frac{Kv}{M}$	α
1	3.08	0.872	21.10	0.3227	18°-50′
2	3.21	0.872	22.00	0.3363	19°-40′
3	3.35	0.872	22.97	0.3510	20°-30′
4	3.52	0.872	24.13	0.3688	21°-40′
5	3.70	0.872	25.37	0.3877	22°-50′
6	3.90	0.872	26.75	0.4087	24°-10′

Similar calculations can be made on the rows of the other expansions.

Design of Curtis-Parsons Turbine

Example. Suppose a Curtis wheel replaces the H.P. cylinder of the preceding turbine. Steam expands to 66 lb. abs. and the heat assignment = 96 B.t.u.

Then steam velocity $V_0 = 223.7\sqrt{96} = 2192$ ft. per sec.

Assume $R = 0.22$ and stage efficiency = 0.675.

Then $U = RV_0 = 0.22 \times 2192 = 482.3$ ft. per sec.

and mean diam. $d = \dfrac{720 \times 482.3}{3.1416 \times 3000} = 36.82$ in., say $36\frac{13}{16}$ in.

The final condition at 66 lbs. = 1281 − (0.675 × 96) = 1216 B.t.u., which is somewhat dryer than the corresponding point on the straight Parsons design.

The stage efficiency of the remainding Parsons sections will vary from 75 to 78 per cent as the leakage loss is reduced. Hence the final point can be found, the reheat factor calculated, the cumulative heat and volumes curves drawn and the blade lengths and angles calculated as before.

In checking this design the windage losses of the Curtis wheel should be calculated and a check made on the assumed conditions at the beginning of the Parsons section.

MISCELLANEOUS THERMODYNAMIC CONSIDERATIONS

Radiation. The radiation losses from the larger marine steam turbines are approximately 0.5 per cent of the entering heat. This loss may amount to 1 per cent on small auxiliary turbines. Little information is available regarding radiation losses and they are generally neglected in turbine calculations.

It is general practice to reduce radiation loss to a minimum partly for the sake of economy but largely to reduce the temperature of the engine room. The high pressure section of the cylinder is covered with asbestos blocks held in place by wire netting fastened to the cylinder by screws. The outer surface is smoothed over with hard cement and covered with canvas unless sheet lagging is furnished. Sometimes the flanges even are covered by special moulded frames carrying the heat insulating material. All small non-condensing auxiliary turbines should be carefully covered.

Exhaust Loss. A certain amount of heat, corresponding to the kinetic energy of the absolute velocity leaving the last row of blades, is lost in the exhaust. If this velocity V_4 is known, then the **heat loss** is $H_E = \left(\frac{V_4}{223.7}\right)^2$ and the **percentage loss** $= \frac{H_E}{H_1 - H_2}$ where $H_1 - H_2$ is the total adiabatic heat drop.

The losses are given in the following table:

Absolute vel. in exhaust ft. per sec. V_4 =	400	500	600	700	800	900	1000
Heat lost in exhaust, B.t.u. H_E =	3.2	5.0	7.2	9.8	12.8	16.2	20.0

The usual heat available in a marine turbine ($H_1 - H_2$) ranges from 350 to 400 B.t.u. The absolute velocity V_4 leaving large multi-stage **impulse turbines** varies from 400 to 500 ft. per sec., so that the loss amounts to from 1 to 1.5 per cent. With simple impulse wheels and high pressures V_4 may be 1000 ft. or more and the loss then varies from 4 to 6 per cent.

In **reaction turbines** the leaving velocity is about 500 ft. per sec. and the loss varies from 1 to 1.5 per cent. In large sizes or where semi-wing and wing blades are used with large angles, the loss may run as high as 3 or 4 per cent. The design should aim to keep this loss within reasonable limits.

Probable Thermodynamic Development. Designing engineers are giving much attention to the possibilities of decreasing steam consumption and fuel consumption by the use of **higher steam pressures** and **higher superheat.** The possibilities of steam at 600 lbs. press. and 600 deg. temperature are being carefully considered.

Undoubtedly there is a considerable thermodynamic gain available by such pressures which would theoretically improve present performance by about 15 per cent. However, there are increased losses due to the use of denser steam in turbines and higher friction losses. This possible gain may not be fully utilized. Nevertheless, the advantages of higher pressures are such that steam pressures are steadily being increased until 350 lbs. gage is in use on land and higher pressures are contemplated.

Some interesting discussions on this subject may be found in the following papers, "Higher Steam Pressures," by Robert Cramer, Vol. 37, Trans. A.S.M.E.; "High Steam Pressure and Superheat," by Eskil Berg, Power, Dec. 10, 1918, and "Progress in Turbine Ship Propulsion," by Francis Hodgkinson, Trans. Soc. N. A. & M. E., Nov. 1918.

A small gain in efficiency may be secured by using **higher vacuum**, though this is limited by the temperature of cooling water available. Higher vacuum may be expected as engineers are appreciating the fact that air leakage largely controls both vacuum and heat transfer. Much study and experiment is being devoted to the laws of heat transfer in surface condensers and increased vacuum may be expected as new methods and apparatus are developed.

Another development that will increase turbine economy is the use of **higher blade speeds.** General Electric Co. have used speeds as high as 860 ft. per sec. on low pressure blade wheels. This has been exceeded by DeLaval on small machines where 1200 ft. per sec. has been used. Such wheel speeds result in fewer stages and less disc friction and leakage losses, and, as long blades can be used on the low pressure stage, there may be little or no increased exit loss. On the other hand, these higher speeds subject the blades and discs to severe stresses and only the highest grade of material will stand them. But as weight is a determining factor in marine service, these higher speeds may be adopted in order to reduce turbine weight.

Martin's "New Theory of the Steam Turbine"

A very interesting series of articles on "A New Theory of the Steam Turbine" was published by *Mr. H. M. Martin* in *Engineering*, July 5th and 19th, Aug. 16th and 23d and Sept. 6th, 1918, from which the following notes are abstracted. The articles are long and contain many diagrams and formulæ, which cannot be reproduced here for lack of space.

Some notable discrepancies between theory and practice have been noted from time to time in investigating the performance of steam turbines. Turbine design has now advanced to such a point that the probable gains in efficiency in any one direction will be of a small order. Hence it is necessary to study these discrepancies and find out wherein formerly accepted theory has been at fault.

Among the most disconcerting anomalies was the well established experimental result that under certain conditions with wet steam, the weight of steam discharged from a nozzle was actually greater than was physically possible, according to previously accepted theory. This had formerly been attributed to the presence of moisture in the steam for no such apparent discrepancy was noted with superheated steam. The author shows that this theory is untenable for it would require from 10 to 20 per cent moisture in the steam to account for the observed discrepancy and this degree of wetness in the entering steam was hardly probable.

Former theory is based on the assumption that the steam expands in thermal equilibrium during its expansion. This appears to be the fundamental error in formerly accepted theory according to the author. The experiments of *Wilson, Callender and Nicholson* and others are quoted to show that expansion under certain conditions may take place under supersaturated conditions, and this would account for the apparently increased discharge.

Evidence is produced to show that the steam during its passage through a turbine, particularly in the saturated field of the heat-entropy diagram, is never in thermal equilibrium but is really in a supersaturated condition. Under such conditions, the relation between volume and pressure of the wet steam is hence never the same as if thermal equilibrium were established. The difficulties that have prevented a detection of this condition are very fully discussed in the articles.

Wilson's experiments and those of *Dr. Stodola*, covering the commencement of condensation in a flowing jet, indicate that the speed of the steam in certain stages of the turbine may be sufficiently high to prevent the formation of particles of moisture with the consequent liberation of latent heat and the establishment of thermal equilibrium. This is further evidence that expansion takes place in a supersaturated state.

The author analyzes the results of these experiments and by applying his interpretation of the same is able to harmonize superheat and vacuum corrections. *Callender's Steam Tables* and constants were used in his analysis. A Heat-entropy chart is drawn from these tables with two sets of lines in the saturated state, one representing wet steam in thermal equilibrium and the other representing steam in a supersaturated state. It has been shown from experiment that save for a thin film of moisture on adjacent surfaces, no condensation occurs when the saturation line is crossed during the expansion of steam. The steam merely becomes supersaturated and continues in this condition until the Wilson line is reached, when condensation commences. The Wilson line is drawn on the diagram. Rational solutions of turbine problems are obtained by assuming supersaturated expansion on this chart.

Reheat factors are calculated on the basis of formulæ derived by the author. The method of calculating the Wilson line is shown. It is possible with this data to compute cumulative heat with reheat factors and supersaturated expansion. Calculations are given to show the application of the new methods to turbine design.

Martin recapitulates the principal points in his new theory as follows: "Starting from the now accepted view that wet steam is not discharged from a nozzle in a condition of thermal equilibrium, reasons have been given for believing that in actual turbine practice, the expansion of wet steam never does occur in thermal equilibrium. Evidence of this is afforded by the condition of the steam in the exhaust branch of the turbine where temperature measurements do not correspond with the pressure of the steam as simultaneously observed. True, the apparent defect of temperature is small, but it has been pointed out that the kinetic theory of gases requires that the readings of a thermometer immersed in undercooled steam should correspond to the pressure rather than to the temperature of undercooled steam. That the apparent defect of temperature may amount to many degrees, may be inferred from Wilson's experiments on the expansion of supersaturated vapor where, when the expansion was continued beyond the point at which moisture first separated out, the additional condensation came on new nuclei and these nuclei can be effective only when the defect in temperature amounts to some tenths of degrees. It was shown that on this hypothesis a rational explanation of the anomalies in superheat and vacuum corrections became possible, thus bringing theory in accord with actual experience."

This theory introduces some decidedly new views on the expansion of steam. Since condensation does not take place in the supersaturated vapor, its apparent volume at the lower pressures would be increased over that assumed when thermal equilibrium is established. This would involve some modification of the passages at the exhaust end of the turbine.

However, this theory has been so recently advanced that designing engineers have not had time to apply it to actual designs nor to study the performance of turbines designed in accordance with the new ideas. It will probably be found that this theory will largely rationalize certain empirical design constants now in use and in such case will not make any very marked change in the performance of turbines.

MECHANICAL DESIGN

Rotors for impulse turbines consist of a shaft carrying a series of **discs** as shown in Fig. 49. Those of reaction turbines are usually of the **drum type**, as illustrated in Fig. 197, p. 738, though several builders are now employing disc construction on the sections of Parsons turbines subject to the greatest stress. Combination types usually have both disc and drum construction.

Rotors with Discs. Turbines of the pressure-compounded impulse type, such as Rateau, Kerr and others, consist of a series of discs running at peripheral speeds of 400 to 850 ft. per sec. The rim dimensions are relatively small. Hence the discs are made with flat tapering sections and light hubs for the low speeds and with hyperbolic profile for the high speeds. A large number of wheels are required with low rotational speeds and this necessitates a large span between bearings. The shaft must then be increased in diameter to maintain its stiffness. This increase in diameter increases the leakage through the diaphragm glands between stages. Such loss may be of some magnitude at the H.P. end decreasing to an almost negligible amount at the last diaphragm. However, improved forms of glands reduce this loss to a small amount. The designer therefore has to choose whether his turbine will have high peripheral velocities, few stages and small short shaft, or low peripheral velocities, many stages and a long, large-diameter shaft.

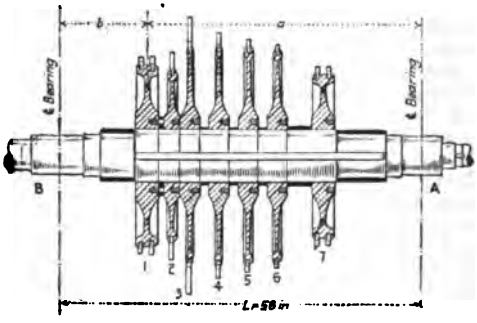


FIG. 49.—Rotor of Compound Impulse Marine Turbine.

Some designers use a shaft tapered to the centre from each end in a series of steps each carrying a disc. Each step is $\frac{1}{8}$ in. to $\frac{1}{32}$ in. less in diameter than the preceding step. This supplies the necessary stiffness and also the necessary series of steps required to assemble the discs.

A clearance of $\frac{1}{32}$ in. is sometimes provided between the hubs of all discs to prevent the bending of the shaft from sudden expansion of these discs when the load increases suddenly. With such clearance the steps in the shaft are about $\frac{1}{4}$ in. smaller in diameter than the preceding step so that each disc can be pressed up against a small shoulder. All discs are held at the end of the shaft by a locknut. Cases have occurred when the discs crept on the shaft due to sudden starting up from cold condition. Creeping was due to sudden heating and centrifugal force, making the bore so large that it was free on the shaft. This emphasizes the need of **heavy press fits.**

Discs are usually pitched as closely as possible on the shaft. The minimum distance between them is generally fixed by the space required by the diaphragm and its gland. As pointed out previously, disc friction losses are decreased by reducing the clearances around these discs. Fig. 49 shows how small diameter spacers are placed between the discs and form the gland between stages. This permits a large shaft diameter with small leakage losses.

Curtis velocity impulse wheels require a heavy rim to carry the two rows of blades, which usually run at peripheral speeds of 400 to 600 ft. per sec. Hence such discs are made with rapidly tapering hyperbolic profiles and heavy hubs. As the number of stages is small, a short small diameter shaft is generally used. However, pure Curtis turbines are now seldom used except for auxiliary drives on shipboard.

In Europe practically all discs are made of forgings. But these are not so readily secured in America and steel castings have been generally used. These have to be carefully inspected and tested for flaws.

A certain amount of steam leaks by the diaphragms. Four to six holes, from 2 to 4 in. in diam., are left in each disc to equalize any possible axial thrust that may be set up by this leakage steam. These holes are a source of weakness and some discs have failed by breaking off a quadrant of the rim from one of these holes. Some manufacturers allow an extra thickness of metal on the disc at the section where these holes are drilled.

Formerly these holes were placed quite near the outer edge. They are being moved nearer the centre of the disc on many new machines.

It is certain that the presence of these holes increases the disc friction losses. Methods have been suggested for strengthening the discs and reducing friction losses by eliminating these holes, but so far no wholly satisfactory plan has been offered.

The discs are forced on the shaft with an allowance for pressing of 0.0015 in. per in. of diam. This requires from 1¼ to 2 tons pressure per in. of diam. of shaft. Such fit is sufficient to insure that the disc will not creep.

Discs are held to the shaft by a single keyway. The boss of the disc must always be made sufficiently thick to be of ample strength above this keyway.

Disc Materials. The permissible diameter at any speed is fixed by the safe working stress of the material. Mild steel is largely used in America for discs, though nickel steel is in general use in Europe. A usual specification for such mild steel is—ultimate tensile strength 80,000 lbs. per sq. in. when annealed, true elastic limit 30,000 lbs. per sq. in., elongation not less than 15 per cent in 2 inches, with a reduction in area of 40 per cent. These discs are rough turned in the shop, annealed, partly finish machined, and assembled on the shaft where the machining is completed. The maximum working stress at 25 per cent overspeed should not exceed the true elastic limit which should be about ⅓ of the ultimate strength. At normal speed this allows a working stress of 16,000 lbs. per sq. in.

Nickel steel has an ultimate strength of 90,000 lbs. per sq. in. and an elastic limit of about 65,000 lbs. per sq. in. Hence, considerably higher speeds can be used with this material. As a rule the overspeed governors are set to operate at 10 per cent overspeed. But the material should stand considerably more than this overspeed.

Design of Turbine Discs

Disc wheels are usually of the form shown in Fig. 49. The required blade diameter is selected together with the form of blade base and the method of securing these to the disc. This gives the form of the rim. The **thickness of the disc** below the rim is chosen so that the disc will not buckle or bend during the mechanical operations of handling and erection. In practice it has been found with simple disc wheels carrying one row of blades that this thickness should not be less than ⅜ in. and is frequently ½ in. Curtis wheels with two rows of blades have this thickness from ¾ in. to 1 in.

Such wheels seldom exceed 1½ in. in thickness just below the rim. This thickness is afterward checked by the calculations of strength which follow and the disc is thickened up or reduced according to the strength found.

The disc thickness at any other radius is found from the equation

$$t = cr^a$$

where t = thickness in in., r = rad. in in., and c = a constant. The exponent "a" varies with the taper of the disc and for hyperbolic forms is negative. Slow speed discs with light rims according to Goudie have values of a ranging from -0.4 to -0.8 and an average of -0.6. American practice seems to favor the higher values. Discs with heavy rims, as in Curtis wheels, have values of $a = -1$.

The value of c is found from the assumed thickness under the rim

$$c = \frac{t_1}{r_1^a}$$

The dimensions of the hub must be selected by the designer so that a proper forced fit can be made and also that it will be strong enough to stand all strains after allowing for the keyway.

Stresses in Disc Wheels. After this preliminary design has been made, it is necessary to check over the stresses in the disc wheel and to modify the assumed dimensions if necessary, as indicated by these calculations.

The exact calculation of the stresses in disc wheels is very complex and the final result is subject to considerable doubt. Many excellent attempts have been made to simplify the calculations by the use of graphs and these are of great assistance to engineers. Hubs are added to enable the discs to be fixed to the shaft, and rims are added to carry blades. These modify further the theoretical solution of hyperbolic disc stresses.

For practical purposes the engineer needs to consider only radial and tangential stresses. Assuming that the disc can be divided into a series of hyperbolic sections, two sets of equations can be derived, one giving the radial and tangential stresses, the other the corresponding strains. The derivation of these formulæ is due to Dr. Stodola and will be found in his book "Die Dampfturbinen."

The following formulæ will aid in the calculation of disc-wheel stresses. The centrifugal force exerted by a given weight

$$F = 0.00001421\, N^2 wd$$

where N = R.P.M., w = weight in lbs. and d = diam. in in. The tangential stress in lbs. per sq. in. in a thin ring of any shape at radius r is

$$T = \frac{12U^2\mu}{g}$$

where μ = density of material in lbs. per cu. in., U = peripheral velocity at rad. r and $g = 32.16$. For steel $T = 0.1045U^2$.

The additional unit stress due to any outer load on the thin ring is $S = \frac{F}{2\pi m}$ where F = total centrifugal force of blades, etc. in lbs. m = effective circumferential net area of cross-section of ring in sq. in.

The maximum stress due to disc in a thin disc of uniform thickness with a small hole at the centre

$$T_{max} = \frac{12\mu U^2}{g}\left(\frac{3+v}{4}\right) = 0.08619U^2 \text{ (for steel)}$$

additional stress due to outer load is $T_0 = \dfrac{2Sr_2}{1-\left(\dfrac{r_1}{r_2}\right)^2}$

additional stress due to a hole in the centre $= \dfrac{12\mu U_1^2}{g}\left(\dfrac{1-v}{4}\right) = 0.0182U_1^2$

(for steel) where v = Poisson's ratio = 0.3 for steel, r_1 = inner rad. and r_2 = outer rad., U_1 = velocity in ft. per sec. at bore and Sr_2 = outer radial stress due to blades, etc. in lbs. per sq. in.

The average tangential stress in a disc of any shape is found by dividing the total bursting force by the net area of the disc cross-section on each side of the axis and for safety this should not exceed the elastic limit of the disc material.

The bursting force on each side of the axis due to the disc alone $M = 2E\mu I$, where $E = 0.00001421N^2$ and N = R.P.M., μ = density of the material as before in lbs. per cu. in., I = moment of inertia of section on one side of the axis calculated about the axis of rotation.

The additional bursting force on each side of the axis due to the outer load of blading shrouds, etc. is $S' = \dfrac{F}{2\pi} = \dfrac{F}{6.28318}$ where F (as before) = total centrifugal force of blades, shrouds, etc.

The average tangential stress $T_a = \dfrac{M+S'}{A_1}$ where A_1 = net area of disc cross-section on each side of the axis.

The thickness of a solid steel disc of uniform strength can be found at any point when the thickness at the centre is known by the formula $\log \dfrac{t}{t_0} = -\dfrac{0.0227U^2}{S}$ where t_0 is the thickness at the centre and t is the required thickness at radius r, where the peripheral speed is U ft. per sec. and S is the desired uniform stress in lbs. per sq. in., or $\log \dfrac{t_1}{t_2} = \dfrac{0.0227}{S}(U_2^2 - U_1^2)$ where t_1 is the thickness at a point where the peripheral speed is U_1 ft. per sec. and t_2 the thickness where the peripheral speed is U_2 ft. per sec.

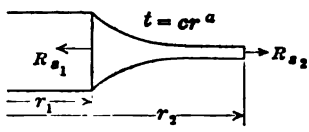


FIG. 50.

Stresses in Discs of Hyperbolic Form. The following general formulæ apply to discs whose outline can be closely approximated by one or more hyperbolas of the form $t = cr^a$, as stated above and as shown in Fig. 50.

The value of a for a given disk profile can be found from

$$a = \frac{\log \dfrac{t_2}{t_1}}{\log \dfrac{r_2}{r_1}} \quad \text{or} = -\frac{\log \dfrac{t_1}{t_2}}{\log \dfrac{r_2}{r_1}}$$

and $\log c = \log t_1 - a \log r_1$.

Stodola gives the radial stress R_s and the tangential stress T at any radius r in lbs. per sq. in. as follows:

$$R_s = \frac{E_1}{1-v^2}\left[(3+v)pr^2 + (m_1+v)b_{11}r^{m_1-1} + (m_2+v)b_{11}r^{m_2-1}\right]$$

and $$T = \frac{E_1}{1 - v^2}\left[(1 + 3v)pr^2 + (1 + m_1 v)b_1 r^{m_1 - 1} + (1 + m_2 v)b_{11} r^{m_2 - 1}\right]$$

where m_1, m_2 and p are algebraic quantities found as follows:

$$m_1 = -\frac{a}{2} - \sqrt{\frac{a^2}{4} - va + 1}$$

$$m_2 = -\frac{a}{2} + \sqrt{\frac{a^2}{4} - va + 1}$$

$$P = -\frac{(1 - v^2)\mu\,\omega^2}{E_1[8 + (3 + v)a]}$$

where a = shape constant of profile of disc section, μ = weight of 1 cu. in. of material, ω = angular velocity of rotation = $\frac{2\pi N}{60}$, E_1 = Young's modulus of elasticity = 29,000,000 for mild steel and v = Poisson's ratio = 0.3 for steel. b_1 and b_{11} are boundary condition constants to be derived from known values of R_e and T. For instance the radial stress at the bore is equal to zero, also the radial stress at the outer radius is known from the centrifugal force of the attached blades. These two known conditions are sufficient to enable one to determine b_1 and b_{11} and hence T and R_e at any radius.

The radial stretch of a narrow rim attached to a disc

$$y_1 = r\left(0.00002842N^2 u r^2 + \frac{C}{2\pi m} - \frac{r_e t_e R_e}{m}\right)$$

where r_e = radius at which ring and disc join, in.; t_e = thickness of disc where it joins rim, in.; R_e = radial stress on disc where it joins the rim in lbs. per sq. in.; N = R.P.M.; u = weight of material, lbs. per cu. in.; r = radius to centre of gravity of cross-section of ring, in.; m = area of cross-section of ring in sq. in. and C = total centrifugal force of blades (if any) in lbs.

S. H. Weaver in his paper on "Disc Wheel Stress Determinations," Vol. 39, Trans. A.S.M.E., gives the following working formulæ, for solving the stresses in discs:

The tangential stresses at any radius r_1 and r_2 may be expressed as follows:

$$T_1 = Ar_1^2 - BR_{e_1} + CR_{e_2} \qquad T_2 = Dr_2^2 - ER_{e_1} + FR_{e_2}$$

where A, B, D, E and F have the following values:

$$B = \frac{m_1 K^{m_2 - 1} - m_2 K^{m_1 - 1}}{K^{m_2 - 1} - K^{m_1 - 1}} \qquad E = \frac{m_1 - m_2}{K^{m_2 - 1} - K^{m_1 - 1}} \qquad C = \frac{E}{K^{a+2}}$$

$$F = B + a \qquad A = \frac{3.3(C - K^2 B) - 1.9K^2}{1 + 0.4125a} \qquad D = \frac{3.3(F - K^2 E) - 1.9}{1. + 0.4125a} \qquad K = \frac{r_1}{r_2}$$

and m_1, m_2 and a have values given above.

Mr. Weaver also shows simple alignment charts based on approximate equations with known negligible errors for the values of a, A, B, F, C, D and E. These are reproduced by permission of the General Electric Review and Mr. Weaver in Figs. 51 to 55 inc.

These charts enable rapid calculations to be made and the stresses can be readily found with sufficient accuracy for check purposes.

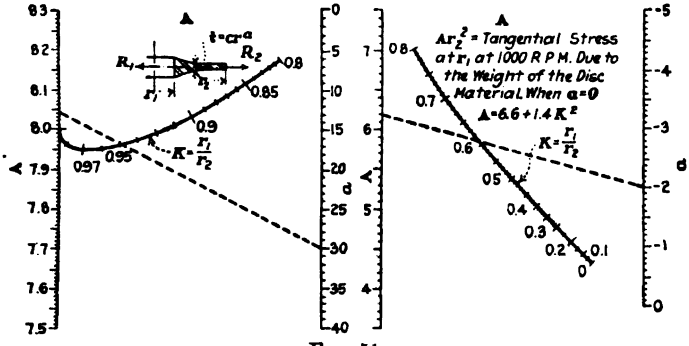


FIG. 51.

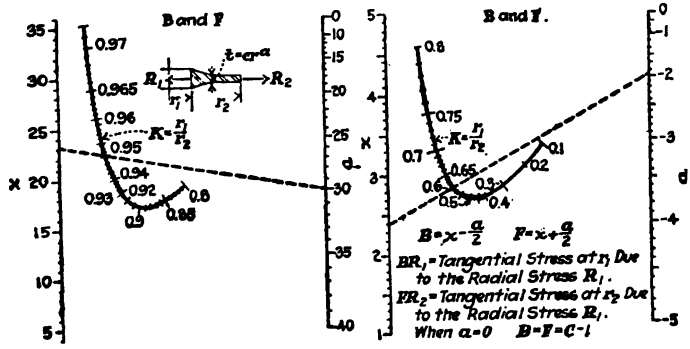


FIG. 52.

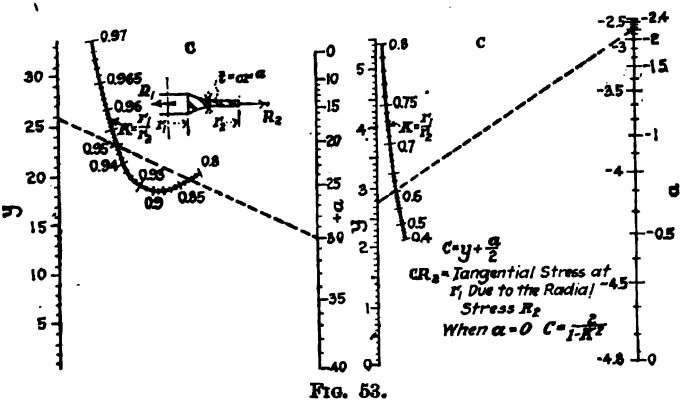


FIG. 53.

The Radial Strain at any raduis r can be found from the formula

$$y = \frac{(T - 0.3R_s)r}{E}$$

where T and R_s and the tangential and radial stresses and E is Young's Modulus.

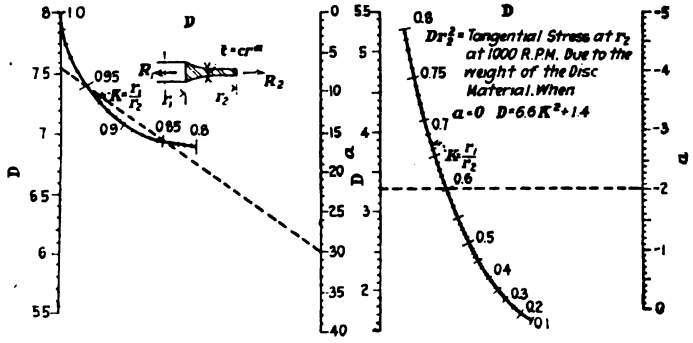


FIG. 54.

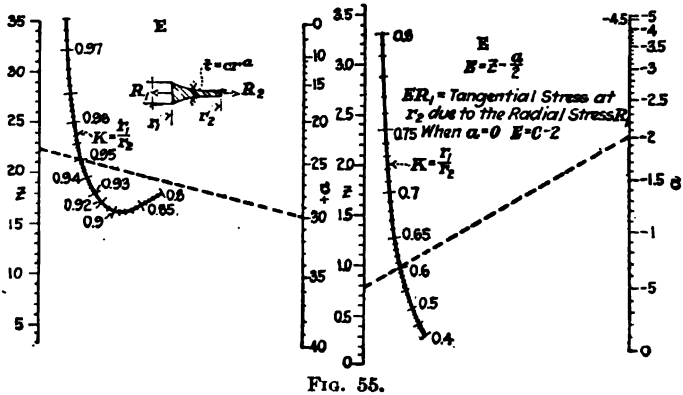


FIG. 55.

The following example shows the application of these charts. Divide the section of a wheel as in Fig. 56 into a number of rings 1, 2, 3, 4 and 5. The profile of the disc is assumed to be represented by a number of connecting hyperbolas whose a value can be found as above from the equation $t = cr^a$. Rings 1 and 5 have uniform thickness and hence a is zero.

Two radial stresses must be known. The expansion of the bore due to centrifugal force is sufficient at some moderate overspeed to equalize the shrink fit. The radial stress at the bore can thus be assumed to be zero. The radial stress at the circumference due to the blade and shroud load per unit area at 1000 R.P.M. equals 14.2 × wt. of blades, shrouds, etc., in lbs. ×

diameter in inches ÷ outer cylindrical area of the ring in sq. in. For the ring shown in Fig. 56 this radial stress at the circumference of the disc amounts to 230 lbs. per sq. in.

The section between rings as at m for rings 1 and 2 has the same thickness for both rings and the same radial stress must exist at this point for either ring. Also the two rings at section m have the same tangential stress and the same elongation.

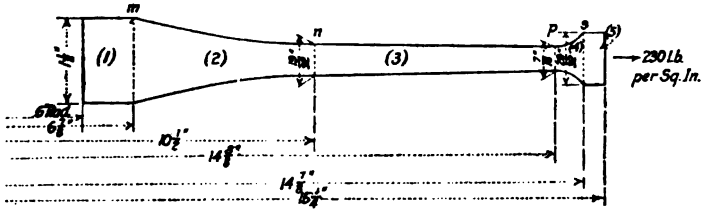


FIG. 56.

Let the radial stresses at the junctions of the different rings at m, n, p and s be taken as e, f, g and h. At section m the radial and tangential stress on the circumference of the imaginary ring 6⅞ in. diameter will have to be the same as the radial and tangential stress on the bore of the imaginary ring of 6⅞ in. diameter. Hence for ring 1 at section m the tangential stress from the preceding equation is

$$T_2 = D \times (6.875)^2 - E \times o + F \times e$$

and for ring 2 at section m the tangential stress is

$$T_1 = A \times (10.5)^2 - B \times e + C \times f$$

Since these two tangential stresses are the same, the following equation can be written

$$D \times (6.875)^2 - E \times o + F \times e = A \times (10.5)^2 - B \times e + C \times f$$

Similar equations can be written for the other sections. The radii, ratio of radii, thickness, values of a and other data are tabulated in the following table:

Ring No.	1	2	3	4	5
r_1	6.00	6.875	10.50	14.375	14.875
r_2	6.875	10.5	14.375	14.875	15.25
t_1	1.50	1.50	0.59	0.44	0.91
t_2	1.50	0.59	0.44	0.91	0.91
$K = \frac{r_1}{r_2}$	0.873	0.654	0.731	0.966	0.975
a	0	−2.20	−0.934	21.12	0
R_1	0	e	f	g	h
R_2	e	f	g	h	230
A	7.67	6.45	7.20	7.96	7.93
B	7.40	3.84	3.83	20.56	39.80
C	8.40	2.07	3.66	42.26	40.80
D	6.43	3.80	4.80	7.63	7.68
E	6.40	2.28	2.65	18.74	38.80
$F = B + a$	7.40	1.64	2.91	40.56	39.80

The values of A, B, C, D, E and F are found from the curves Figs. 51 to 55 inclusive, when the value of a is known. Substituting these values in the above equation for radius 6.875 in. and in similar equations for the other radii, the following equations are derived:

At radius 6.875 in.

$$6.43 \times (6.875)^2 - 6.4 \times 0 + 7.40 \times e = 6.45 \times (10.5)^2 - 3.84 \times e + 2.07 \times f$$

At radius 10.5 in.

$$3.80 \times (10.5)^2 - 2.28 \times e + 1.64 \times f = 7.20 \times (14.375)^2 - 3.83 \times f + 3.66 \times g$$

At radius 14.375 in.

$$4.80 \times (14.375)^2 - 2.65 \times f + 2.91 \times g = 7.96 \times (14.875)^2 - 20.56 \times g + 42.26 \times h$$

At radius 14.875 in.

$$7.63 \times (14.875)^2 - 18.74 \times g + 40.56 \times h = 7.93 \times (15.25)^2 - 39.80 \times h + 40.8 \times 230$$

These equations are solved simultaneously for the radial stresses e, f, g and h and give the following values at 1000 R.P.M. of the disc: $e = 151$, $f = 623$, $g = 546$ and $h = 246$. The tangential stresses at each of the ring sections can be found by substituting the radial stresses on one side of each of the above equations. The tangential stress at the bore and at the circumference of the disc can now be computed from the general formula. Thus the tangential stress at the bore $= Ar_1^2 - BR_{s1} + CR_{s2} = 7.67 \times 6^2 - 7.4 \times 0 + 8.4 \times 151 = 1630$ lbs. The stresses in the disc at 1000 R.P.M. are shown in the following table:

Stresses at 1000 R.P.M.

Radius r =	6	6.875	10.5	14.375	14.875	15.25
Radial stress R_s =	0	151	623	546	246	230
Tangential stress T =	1630	1421	1099	922	1437	1395

Stresses at intermediate points can be calculated in the same manner if so desired.

The wheel runs at 3500 R.P.M. and the equivalent radial and tangential stresses at this speed are obtained by multiplying the values in the above table by the square of the ratio of the speeds, *i.e.* $(3500 \div 1000)^2 = 12.25$. These values are as follows:

Radius r =	6	6.875	10.5	14.375	14.875	15.25
Radial stress R_s =	0	1,850	7,632	6,688	3,013	2,817
Tangential stress T =	19,967	17,407	13,463	11,294	17,603	17,089

These stresses at normal speed are well within safe limits. Stresses at overspeeds can be calculated in the same manner. As noted before, the radial elongation at any radius r is $y = (T - 0.3R_s)\, r \div E$. E for steel may be taken as 29,000,000. The radial elongation y at 3500 R.P.M. is shown in the following table:

At 3500 R.P.M.

Radius r =	6	6.87	10.5	14.37	14.87	15.25
Radial elongation y =	0.00413	0.0040	0.00405	0.00460	0.00856	0.00854

The following graphical method of calculating the strength of rotating discs was proposed by Mr. R. Haerle in "Engineering," Aug. 9, 1918.

Curves are drawn, as in Fig. 57, for certain values of S and D which were derived from Dr. Stodola's formulæ for the strength of rotating discs as given above by assuming that the disc is of constant thickness.

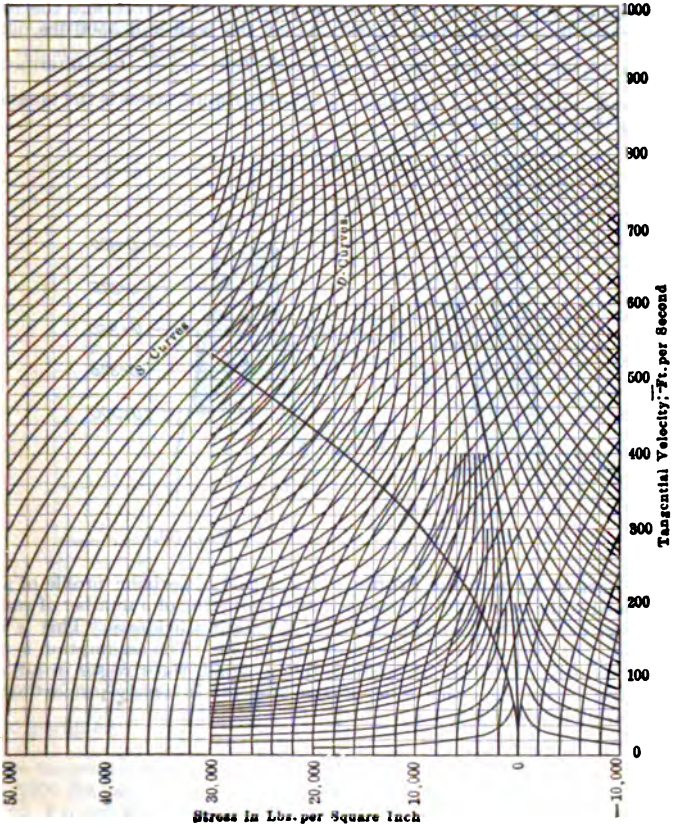


FIG. 57.—Haerle's Curves for Graphical Solution of Strength of Rotating Discs.

For a disc of constant thickness, the radial stress at any point $R_s = \frac{S - D}{2}$ and the tangential stress $T = \frac{S + D}{2}$.

The use of this diagram can be best illustrated by a series of examples:

(1) **Disc of uniform thickness** with hole in centre. Let the rim speed be 500 ft. per sec. and the bore speed be 100 ft. per sec. The peripheral or tangential velocity of the disc at the given speed at both rim and bore can be found as follows:

At both rim and bore $R_s = O$ and hence $S = T + R_s = T$ and $D = T - R_s = T$. Hence $D = S$ at both rim and bore.

Select an S and a D curve that intersect at both the 500 ft. and the 100 ft. lines. Such lines intersect at 22,000 lbs. at the bore and 5500 lbs. at the rim. Applying formulæ $R_s = \frac{S - D}{2}$ and $T = \frac{S + D}{2}$ for intermediate points and plotting values will show that the maximum stress is the tangential stress at the bore of 22,000 lbs.

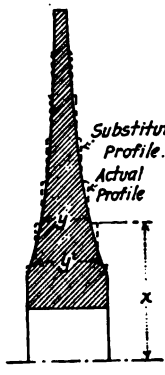


FIG. 58.

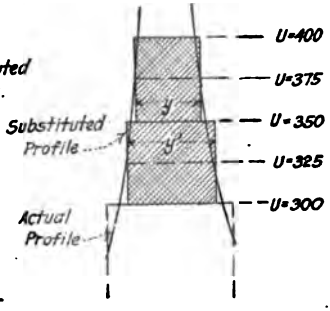


FIG. 59.

(2) **Discs of Hyperbolic Profile.** The actual hyperbolic profile of a disc is assumed to be replaced by a stepped disc consisting of a number of concentric rings, each section or ring being of constant thickness. This is illustrated in Fig. 58 and to a larger scale in Fig. 59. The assumption is made that, provided the steps be comparatively small, the stresses in adjacent concentric layers on either side of the step are inversely proportional to the axial dimensions y and y'.

Hence we may write

$$\frac{y}{y'} = \frac{R'_s}{R_s} = \frac{T'}{T}$$

Letting ΔR_s be the increment (positive or negative) of the radial stress at the step, then

$$\Delta R_s = R_s\left(1 - \frac{y}{y'}\right) \qquad \text{also } v\Delta R_s = \Delta T$$

where v = Poissau's ratio (0.3 for steel)

Then $\Delta S = 1.3\Delta R_s$ and $\Delta D = -0.7\,\Delta R_s$

The general case of a steam turbine disc is one where rows of blades are carried in grooves at the periphery of the wheel, as in Fig. 60. This particular

disc belongs to a 2000 K.W. turbine, operating at 3000 R.P.M. with a mean blade diameter of 48 in. The actual profile is replaced by a series of suitable steps, as shown, chosen at convenient intervals.

The weight of the two rows of blades and shrouds was assumed to be 70 lbs. and the centre of gravity of bucket and shroud is taken at a radius of 24 in. The diam. at the circumference of the disk itself is 46⅞ in. at a peripheral speed of 615 ft. per sec.

The centrifugal pull of the blades per in. length of disc circumference is

$$p = \frac{0.00001421N^2wD}{\pi D} = 2852 \text{ lbs. per in. length.}$$

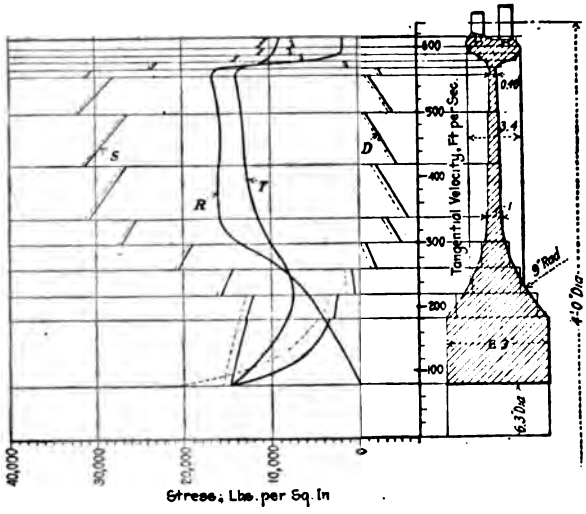


FIG. 60.—Application of Haerle's Method to a Turbine Disc.

The net width of the disc at the periphery after deducting the dovetail grooves $y' = 2.2$ in. Hence the radial stress due to the blade loading is

$$R_s \text{ (at periphery)} = \frac{2852}{2.2} = 1300 \text{ lbs. per sq. in.}$$

The tangential stress at the rim must be assumed for the first trial. Let $T = 9200$ lbs. per sq. in. Hence, at the rim, $S = 9200 + 1300 = 10{,}500$ lbs. per sq. in. and $D = 9200 - 1300 = 7900$ lbs. per sq. in.

These values constitute the points of origin for the S and D curves across the outermost steps.

At the bore of the disc $R_s = O$ and $S = D = T$.

Hence the S and D curves intersect at the bore. However, if it is desired to take into consideration the effect of a forced fit, R_s may have a positive or negative value to meet the case.

The proceedure is as follows:

Beginning at the periphery follow the S and D curves on the chart across the radial depth of the first step, *i.e.*, from $V = 615$ ft. per sec. to $V = 600$ ft.

per sec., at the end of the step ($V = 600$) read off $R_s = \frac{S - D}{2}$; find $\frac{y}{y'}$ from the substituted profile and calculate ΔR_s, ΔS and ΔD by means of the preceding equations. Step over on the chart for the values of ΔS and ΔD and continue the new S and D curve across the second step, etc. If T at the periphery had been chosen correctly the S and D curves at the innermost step must intersect at the bore of the disc.

Table 11 gives the results of the calculations with an assumed rim value of $T = 9200$ lbs. per sq. in. The S and D curves corresponding to this table are shown in Fig. 60 as dotted lines. It will be seen from the table that T was chosen too high for R_s at the bore figures out a negative value instead of zero.

A correction must now be made in the assumed value of T at the rim and the graphical solution repeated a number of times until R_s is found equal to zero at the bore. If T is assumed as 9000 and the values recalculated in Table 12 the desired result is obtained. The new values of S and D are plotted as full lines in Fig. 60.

Table 11

U	y	y'	$\frac{y}{y'}$	$1 - \frac{y}{y'}$	R_s*	ΔR_s	ΔS	ΔD	T
615	.....	2.20	.....	.....	1,300	0	0	0	9,200
600	2.20	3.26	0.675	0.325	2,100	680	890	− 480	
590	3.26	1.80	1.810	−0.810	1,950	−1,580	−2,050	1,100	
580	1.80	0.68	2.650	−1.650	4,080	−6,740	−8,760	4,720	
570	0.68	0.47	1.440	−0.440	11,400	−5,020	−6,520	3,520	
560	0.47	0.55	0.855	0.145	17,040	2,480	3,220	−1,740	
500	0.55	0.73	0.754	0.246	18,200	4,500	5,840	−3,150	
420	0.73	0.92	0.794	0.206	18,100	3,740	4,850	−2,620	
340	0.92	1.18	0.780	0.220	18,400	4,050	5,270	−2,840	
300	1.18	1.90	0.620	0.380	16,200	6,160	8,000	−4,310	
260	1.90	3.20	0.594	0.406	11,400	4,640	6,020	−3,250	
220	3.20	5.11	0.625	0.375	7,500	2,810	3,650	−1,960	
185	5.11	6.30	0.811	0.189	4,800	910	1,180	− 636	
82	6.30	6.30	1.000	0.000	−3,200	0	0	0	18,900

* Values at inner diameter of steps.

Table 12

U	y	y'	$1 - \frac{y}{y'}$	R_s*	ΔR_s	ΔS	ΔD	T actual	R_s actual
615	.....	2.20	.....	1,300	0	0	0	9,000	1,300
600	2.20	3.26	0.325	2,120	690	900	− 484	9,400	1,300
590	3.26	1.80	−0.810	1,960	−1,580	−2,060	1,150	9,800	2,500
580	1.80	0.68	−1.650	4,100	−6,760	−8,800	4,750	10,900	6,800
570	0.68	0.47	−0.440	11,450	−5,030	−6,580	3,550	13,600	16,300
560	0.47	0.55	0.145	17,050	2,480	3,220	−1,740	14,100	16,600
500	0.55	0.73	0.246	18,250	4,500	5,850	−3,150	13,500	16,200
420	0.73	0.92	0.206	18,170	3,740	4,860	−2,620	13,100	16,200
340	0.92	1.18	0.220	18,440	4,060	5,300	−2,850	11,700	16,000
300	1.18	1.90	0.380	16,280	6,200	8,060	−4,350	10,000	13,600
260	1.90	3.20	0.406	11,500	4,670	6,100	−3,270	8,300	8,800
220	3.20	5.11	0.375	7,800	2,920	3,800	−2,050	7,700	5,900
185	5.11	6.30	0.189	5,150	970	126	− 680	8,300	4,200
82	6.30	6.30	0.000	0	0	0	0	14,800	0

* Values at inner diameter of steps.

The axial thickness of the several steps of the substituted profile correspond to the thickness of the true profile at points midway between the steps. Hence the stresses as determined by the S and D curves at these points coincide with the actual stresses in the original profile. If the distance between the S and D curves at these middle points be bisected, points on the T curve are obtained directly. A smooth curve drawn through these T points represents graphically the distribution and magnitude of the tangential stresses in the disk. Values of R_e can now be found at these middle points and the R_e curve similarly drawn. The values of T and R_e in the last two columns of Table 2 were found by scaling off the diagram at the various radii and do not follow directly from the calculations.

This graphical method offers a ready solution of all disc stress problems after the S and D curves are drawn to a sufficiently large scale.

Several other graphical methods have been offered for the solution of disc stress. Goudie's "Steam Turbines" contains another method. A. M. Martin designs his discs in quite another way in his book "Steam Turbines."

Critical Speed of Rotors

In a horizontal rotor there is always a certain deflection due to its own weight, which causes the centre of mass to lie below the axis of rotation. With increase of speed, this eccentricity of mass results in an increasing centrifugal force which tends to bend the shaft. This is aggravated if the mass is out of balance. When a certain speed is reached this centrifugal force neutralizes the elastic force of the shaft that normally resists vibration and the deflection of the shaft becomes excessive. Severe vibrations are set up and parts frequently foul each other. Above this speed the shaft appears to revolve about its true centre of mass, the deflection decreases and the vibration ceases. This speed of extreme vibration is known as the first "critical speed." Other critical speeds may occur but these are usually at higher speeds than are normal in turbine practice. The critical speed is therefore equal numerically to the frequency of vibration of the shaft with the masses mounted on it, if deflected by an external force while the shaft is at a standstill.

The calculation of critical speeds is extremely difficult except in the simplest forms of shafts and wheels. However, no great refinement of calculation is usually required in such calculations for the shaft is usually designed to run well above or below the critical speed, usually 30 per cent above or below.

A complete discussion of critical speeds cannot be given in this book. For more extended discussion, reference may be made to Goudie's "Steam Turbines," Stodola's "Steam Turbines," Morley's "Strength of Materials" and to the following articles: "Calculation of Vibration and Whirling Speeds" by Prof. A. Morley, in "Engineering" July 30, 1909, and "The Whirling Speeds of Loaded Shafts" by Wm. Kerr, in "Engineering," Feb. 18, Mar. 3, 10 and 17, 1916.

Some designers use the following simple equation for the first approximation of the critical speed. Critical speed $= \frac{188}{\sqrt{y}}$ where y = maximum shaft deflection. Generally more complete data is required and the following methods may be used.

Three general cases of critical speed usually arise: (*a*) a small shaft with a single wheel whose weight is large compared with that of the shaft, as the DeLaval wheel, (*b*) an approximately uniform shaft loaded by its own weight and a series of wheels, as a Curtis-Rateau rotor, and (*c*) a shaft of varying

section made up of a stepped drum or disc and drum such as the usual Parsons spindle.

The shaft may be treated as a "supported beam" if held in comparatively short spherically seated bearings and as a "fixed beam" if held in long rigid bearings which tend to constrain the shaft.

Shafts, particularly Parsons spindles, generally are designed to run well below the critical speed. Some small turbines are in use which run above the critical speed and must be carried through this point rapidly when running the turbine up to speed. The following notes are based on methods described in Goudie's "Steam Turbines."

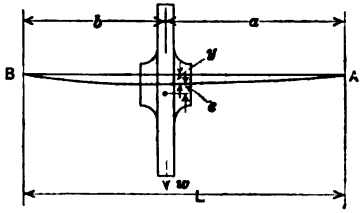


FIG. 61.—Case (a) Small Shaft with Single Heavy Wheel.

CASE (A). **Small shaft with single heavy wheel (Fig 61).**

Let W = weight of wheel in lbs.
e = eccentricity of the mass centre.
L = span of bearings A and B in in.
a = distance of wheel from bearing A in in.
b = distance of wheel from bearing B in in.
y = deflection of shaft from the central position of AB in in.
N_c = R.P.M. at critical speed.
g = 32.2 × 12 = 386.4.
F = stiffness or force required to produce unit deflection at the wheel.

Deflection $y = \dfrac{WL^3}{KIE}$ where I = moment of inertia of uniform shaft about its diametral axis $= \dfrac{\pi}{64} d^4$ where d = shaft diam. in in. E = Young's Modulus for steel = 29,000,000.

K = coefficient depending on values of a and b and the nature of the end supports.

$$\text{Simple supported shaft } K = \frac{3L^4}{a^2b^2}$$

$$\text{Shaft fixed at ends } K = \frac{3L^6}{a^3b^3}$$

$$\text{End } A \text{ fixed } B \text{ supported } K = \frac{12L^6}{a^3b^2(3a + 4b)}$$

With unit deflection F may be substituted for W and the stiffness of the shaft determined

$$F = \frac{KIE}{L^3}$$

The critical speed $N_c = \dfrac{60}{2\pi}\sqrt{\dfrac{IEgK}{L^3W}} = 223{,}950\dfrac{d^2}{L}\sqrt{\dfrac{K}{LW}}$ **for steel**

Example. The wheel of a single disc turbine weighing 125 lbs. is mounted centrally on a shaft of average diameter of 2½ in. Bearings are spherically seated 22 in. apart. Neglecting the effect of the shaft in this case

$$K = \frac{3L^4}{a^2b^2}, \quad N_c = 223{,}950\,\frac{d^2}{L}\sqrt{\frac{K}{LW}} = 223{,}950 \times \frac{(2.5)^2}{22}\sqrt{\frac{3 \times (22)^4}{(11)^2 \times (11)^2 \times 22 \times 125}}$$

= 8410 R.P.M.

The running speed of this unit would probably be 3600 R.P.M. or about 43 per cent of the critical speed.

CASE (B). **Uniform shaft carrying a number of wheels.**

Dunkerly's semi-empirical method offers the best solution for this case. The critical speed of the shaft alone without wheels is first calculated. Then the critical speed of each wheel is figured on the assumption that it is mounted on a weightless shaft as in the previous case.

Let n_1, n_2, n_3, etc. be the critical speeds of the several wheels in revolutions per second and n_s the critical speed of the shaft alone, n_c = the critical speed of the whole rotor.

Dunkerly's equation states that

$$\frac{1}{n_c^2} = \frac{1}{n_s^2} + \frac{1}{n_1^2} + \frac{1}{n_2^2} + \frac{1}{n_3^2} \text{ etc.} = \frac{1}{n_s^2} + \Sigma\,\frac{1}{n^2}$$

For a uniform shaft $n_s = \frac{1}{2\pi}\sqrt{\frac{IEgK_s}{W_sL^3}}$ where W_s = weight of shaft in lbs. and

$K_s = (\pi)^4$ for shaft supported at each end,

$= \left(\frac{3}{2}\pi\right)^4$ for shaft fixed at each end,

$= \left(\frac{5}{4}\pi\right)^4$ for shaft fixed at one end and supported at the other.

Dunkerly's equation can now be written $\frac{1}{(n_c)^2} = \frac{(2\pi)^2L^3}{IEg}\left(\frac{W_s}{K_s} + \Sigma\,\frac{W}{K}\right)$

$= \frac{1}{13{,}940{,}000}\,\frac{L^3}{d^4}\left(\frac{W_s}{K_s} + \Sigma\,\frac{W}{K}\right)$ for a steel shaft.

Example. A Curtis Rateau rotor with astern section is shown in Fig. 49. The approximate values of disc weights and distances a and b are as follows:

Wheel	1	2	3	4	5	6	7
Weight (W) lbs.	410	290	360	360	360	360	460
a, in.	44	40	37	32½	27¾	23	16
b, in.	14	18	21	25½	30¼	35	42

Assume that the stepped shaft is replaced by a uniform shaft 8 in. in diam., 58 in. long and supported at each end. The weight W_s of the shaft at 0.29 lb. per cu. in. is 845 lbs.

K_s for shaft supported at each end $= (\pi)^4 = 97.4$.

$$\therefore \frac{W_s}{K_s} = \frac{845}{97.4} = 8.675.$$

For each wheel $K = \frac{3L^4}{a^2b^2} = \frac{33{,}949{,}388}{a^2b^2}$

$\Sigma\,\frac{W}{K}$ can be found from Table 13 following.

Table 13

Wheel	a, in.	b, in.	K	W, lbs.	$\frac{W}{K}$
1	44	14	89.45	410	4.585
2	40	18	65.50	290	4.425
3	37	21	56.25	360	6.400
4	32½	25½	49.44	360	7.280
5	27¾	30¼	48.15	360	7.570
6	23	35	52.40	360	6.870
7	16	42	75.15	460	6.120
					43.250

$$\Sigma \frac{W}{K} = 43.25$$

$$\frac{1}{n_c^2} = \frac{1}{13{,}940{,}000} \times \frac{L^3}{d^4}\left(\frac{W_s}{K_s} + \Sigma \frac{W}{K}\right) = \frac{1}{13{,}940{,}000} \times \frac{194{,}112}{4096}\,(8.675 + 43.93)$$

$$= \frac{195{,}112 \times 51.915}{13{,}940{,}000 \times 4096},$$

or $n_c = \sqrt{\frac{13{,}940{,}000 \times 4096}{195{,}112 \times 51.915}} = 75$ revolutions per sec. $= 4500$ R.P.M.

This critical speed may be raised by increasing the shaft diameter.

The normal speed of this turbine was 3250 R.P.M. or 28 per cent below the critical speed as calculated above.

Case (c). **Drum-type of rotor** with irregular cross-section and varying moment of inertia as well as varying rate of loading.

This is solved by assuming that the total resilience of the rotor is equal to the total kinetic energy of the rotor in the mid-position of vibration.

It is also assumed that the deflection curve from statical loading is the approximate actual deflection curve when whirling commences.

On this curve if there is a deflection y at any point due to a substituted load W at this point, the resilience is $\frac{1}{2}Wy$. The total resilience for the whole rotor with the assumed deflection curve is $\frac{1}{2}\Sigma Wy$.

If ω is the speed in radians, then the total kinetic energy of the mean or straight position of the shaft can be shown to be $\frac{1}{2}\Sigma \frac{W}{g}\omega^2 y^2$ and this is assumed equal to the total resilience.

Hence at critical speed $\Sigma Wy = \frac{\omega_c^2}{g}\Sigma Wy^2$

or the critical speed in R.P.M. is $N_c = \frac{60}{2\pi}\sqrt{\frac{g\Sigma Wy}{\Sigma Wy^2}} = 187.7\sqrt{\frac{\Sigma Wy}{\Sigma Wy^2}}$.

This method involves considerable graphical work in the determination of the curve of statical deflection and force and funicular polygons are used. An excellent specimen computation is given in Goudie's Steam Turbines on pages 252 to 254.

The critical speed causes intense vibrations in some machines when certain parts are slightly out of dynamic or static balance. These vibrations can be considerably reduced by very careful machine work and balancing. A well balanced shaft may pass through its critical speed with very slight vibration.

Design of Rotor Shafts

The diameter of a shaft is fixed more with regard to stiffness than to strength. The drum construction is very stiff and the usual maximum statical deflection ranges from 0.001 to 0.005 in. It is usually run on spherically seated bearings so that the ends are simply supported. The maximum bending moment generally occurs at the middle of the drum.

The disc type of rotor is less stiff and with spherically seated bearings its maximum statical deflection ranges from 0.005 to 0.03 in. These shafts are run in fixed bearings in many cases to keep the stiffness as great as possible. In that case the above deflections are decreased by 50 per cent. With fixed bearings the maximum bending moment occurs at the bearings. The journal sizes are influenced by the allowable journal speeds referred to later in the section on "Bearings," p. 636.

The stress at any section from tension or torsion may be found from the formulæ $f_t = \frac{M}{Z}$ or $f_s = \frac{T}{Z}$ where Z is the section modulus, T is the torque and M the bending moment. The maximum value of f should not exceed 5000 lbs. per sq. in.

A keyway must be cut in the shaft for the key that holds the disc in place. It has been found that the shaft stresses are not in equilibrium with a single keyway. Hence it is quite general practice to cut a second keyway on the opposite side of the shaft to equalise strains. This second keyway is filled with a feather flush with the surface of the shaft and held in place by countersunk screws.

Drum Rotors

The drum diameter of Parsons turbines must be large enough to maintain the necessary rigidity in the spindle so as to avoid trouble from critical speeds and also to keep the length of the turbine within reasonable bounds.

The diameter must be chosen with regard for the desirable working stresses of ordinary steels.

Where L.P. sections must run at high peripheral speeds in order to handle the steam economically, it is often desirable to carry these sections of blading on discs of hyperbolic cross-section designed for uniform stress. Each disc carries 4 to 7 rows of blades.

Shaft Couplings

All Parsons types of turbines use some modified form of flexible claw coupling. A coupling of this type, furnished by Allis-Chalmers Co., is shown in Fig. 62. The spindle ends of forged steel are pressed and keyed on the shaft and have suitable claws machined on the periphery which engage with similar claws on the coupling sleeve, which is free to adjust itself in these claws. The sleeve is split in the middle and held together by reamed bolts. All wearing surfaces are well lubricated.

A similar coupling, shown in Fig. 63, is used by the General Electric Co. The sleeves are supported on bronze bearing rings carried in dovetails in the male end on the shaft. A unique feature of this design is the use of renewable bearing face plates held in dovetails in the jaws of the male ends. The jaw clearances range from 0.003 to 0.007 in.

Ample lubrication is also provided by the oil catchers and holes as shown. The coupling is also provided with a small shock absorbing spring placed in the centre of the shaft to take up shocks due to axial movement of the high speed shaft pinion.

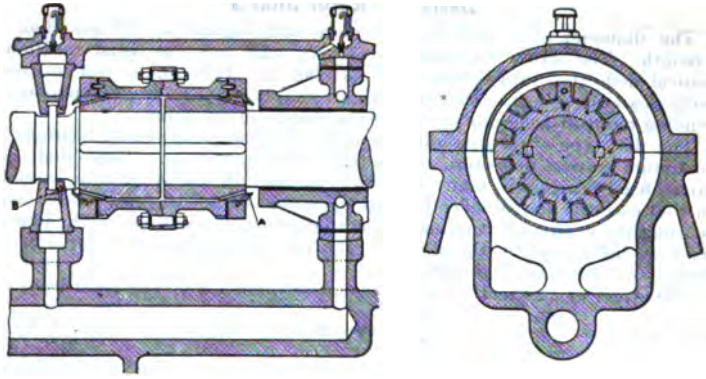


FIG. 62.—Allis-Chalmers Claw Coupling.

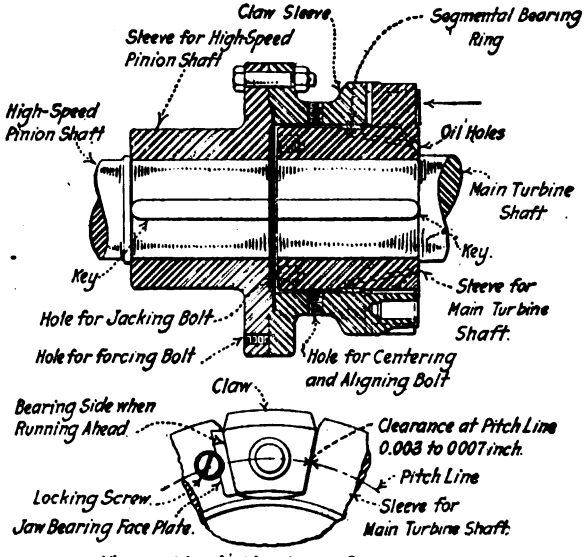


FIG. 63.—General Electric Coupling.

Westinghouse now uses a slightly different form of coupling, shown in Fig. 64.

The male ends *A* and *B* are pressed and keyed on the shafts of turbine and pinion. The coupling sleeve *C* is bolted to the pinion and extends over the turbine end as shown. This sleeve carries a number of hardened bolted pins *D* which engage corresponding bronze bushed holes in the coupling end of the turbine spindle. Clearance is provided on each end of these pins so that the pinion is free to move axially without stressing the turbine end. All pins and bearing surfaces are lubricated as shown.

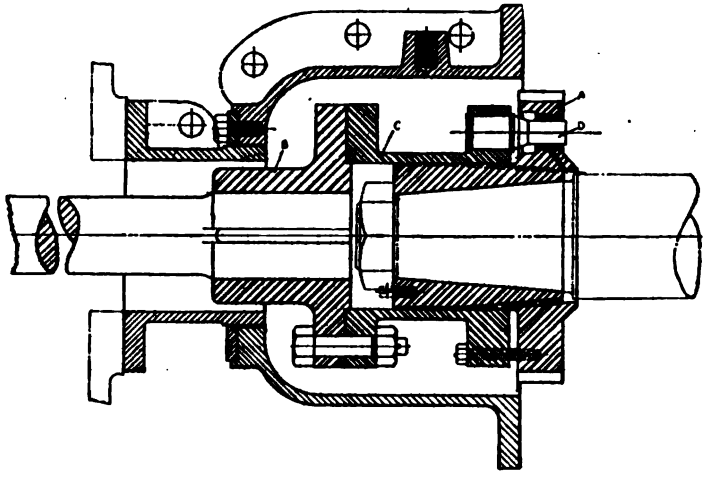


FIG. 64.—Westinghouse Coupling.

The oiling holes of all these forms of couplings must be inspected periodically to see that they are open and in good working order.

Several other forms of flexible coupling are in use on small turbines.

Small Kerr units use two ordinary flange couplings. One carries bolts which project into holes in the other half which are lined with bushings of rubber. These give the necessary elasticity and have worked well in practice.

Some new forms of coupling with springs are being tried on propeller shafts to keep the vibration of the propeller off the gears when used. No data are yet available regarding their performance.

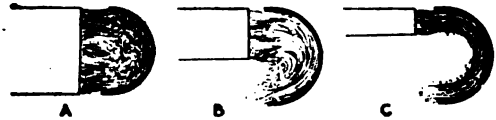


FIG. 65.—Effect of Jet Flowing Across Curved Blade.

Blading

Design of Impulse Blading. Take a bucket as shown in Fig. 65 with full return angle. If a stream of steam is directed into this bucket as at *A*, it will fill and the steam will eddy around producing very low efficiency. If

the steam enters only on half the width of the bucket, as in *B*, there will be flow toward the outlet on the other side, but still with much eddying and friction.

There will be a certain maximum discharge nozzle that will produce practically no eddies, as shown in *C*.

If the blade angle is less than 180 deg., as in an actual turbine blade, there will also be some maximum width of stream that will produce minimum losses in crossing a blade. This width of stream is known as the "**ratio**" of the blade and may be taken as the ratio of the clear width of passage between blades measured along the centre line of the blade divided by the radius of curvature of the front of the blade, as in Fig. 66 where **ratio** $= \frac{T}{R}$.

In fixing the width of blade sections it must be kept in mind that long easy curves on the face of blades produce less shock and eddies but have a greater wall area which causes higher friction losses than narrow blades with sharp curves. The friction losses are of a greater magnitude than the usual losses from shock. Hence the smallest sections that can be easily manufactured and that possess the necessary strength and ruggedness are generally used.

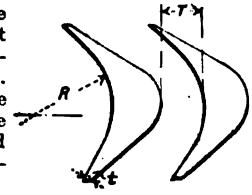


FIG. 66.—Impulse Blade Proportions.

Sections of impulse blading less than ¾ in. wide are seldom used. The usual widths are 1 in., 1¼ in., 1½ in. and 2 in. with larger blades in some low-pressure stages. In general the higher the steam pressure and the greater the velocity, the greater the width required for a given blade height. The value of the "ratio" given above varies from about 0.35 with blades having sharp angles, to about 0.60 in blades with wide angles as in the second revolving row of a Curtis section.

Blades must be proportioned to withstand centrifugal force, bending and vibration. These combined limit the length of blades for any given width. When the blade length is increased above about ¼ of the mean blade ring diameter, the width of passage between tips of blades becomes so large that the steam no longer flows in any fixed path and the energy of the steam is largely dissipated in turbulent flow. This can be corrected to some extent by the use of wider blades. But these blades require heavier sections, produce greater centrifugal forces, and necessitate more elaborate fastenings at their base. Furthermore, the centrifugal and bending stresses increase as the blade length increases and the liability to vibration also appears to increase.

The **centrifugal force on a blade** is given by the equation

$$F = 0.00002842N^2Wr$$

where N = R.P.M.; W = weight of blade and shroud ring, beyond the section of the base carrying the load, in lbs.; r = rad. in in. to centre of gravity of blade figured beyond the section under stress.

Blades are seldom of uniform cross-section and frequently have the distance piece integral with the base of the blade. Hence the weight of the blade and its centre of gravity must be found by calculation from the design. The following weights of blading materials will assist in this calculation: Nickel steel, 0.285 lb. per cu. in.; forged steel, 0.29 lb. per cu. in.; bronze, 0.295 lb. per cu. in., monel metal, 0.323 lb. per cu. in.

The centrifugal force is exerted at the base of the blade where it bears against the edges of the slot, as shown in various figures later. This section

is invariably less than the full section of the blade. The stress, either tension or crushing, should not exceed the safe working strength of the material when operated at 20 per cent over normal speed. In general the stress at the base should not exceed 15,000 lbs. per sq. in., although in some turbines this has been allowed to go as high as 20,000 lbs. on account of high wheel speeds and special material in the blading.

The degree of **vibration in turbine blades** and the methods of calculating such vibration are not very definitely known. This vibration is a combination of the natural period of vibration of the spindle as a whole due to its unbalanced forces and to its critical speed and the period of resonance set up in the blade due to the velocity of the steam in passing over it. Empirical formulæ have been devised by manufacturers to calculate these vibrations. Care must be taken that the section of the blade be so designed that the combined vibration curve has no extreme peaks.

The blade itself when steam is flowing across it, acts like a beam uniformly loaded and held either at one end, as in Parsons blading, or at both ends, where shrouds are used, but the degree of support afforded by the shroud is not definitely known. Very little work has been done on blade vibration and research in this line might clear up some questions of blade failures. A very complete discussion of blade strength appears in a paper by Gerald Stoney on "Stresses in Turbine Blading," Engineering, April 26, 1918.

The steam velocity also causes vibrations. Impulse blades are strengthened in three ways to overcome this difficulty. The blades are made wide where the velocity is high; thus in a Curtis stage the width of the first blade may be 1½ in., the stationary reversing blades 1¼ in. wide and the second moving row 1 in. The blades are also reinforced by thickening the back of the blade at the base, depending on the width of the blades, from $\frac{1}{16}$ to $\frac{3}{16}$ more than the standard cross-section and tapering off buttress-like to the normal cross-section at about ¼ of the total blade length. Impulse blades are also frequently provided with a shroud ring to which their outer ends are firmly fastened. The centrifugal force of the section of the shroud ring carried by each blade must be considered in calculating the centrifugal force on the base of the blade. This shroud ring exerts considerable influence in reducing vibration in blades. Blades are also stiffened against vibration by a wire lacing in the middle of the blade. This method has not been used in impulse blades although widely used in reaction construction. Impulse blades should not have ratios of blade length to width exceeding 8 or 9, to provide sufficient stiffness to avoid undue vibration.

The **thickness** t, Fig. 67, **of the inlet and exhaust edges** of impulse blades varies with different manufacturers. Too thin an edge involves manufacturing difficulties and the blade is easily injured. Too thick an edge causes steam shock. It is questionable whether the thickness of this edge has much effect on efficiency. Thin edges are so easily distorted that manufacturers are using thicker edges. This thickness generally varies from 0.0175 to 0.025 in. The practice of one leading manufacturer is as follows:

t = 0.0175 for 1-in. blades
= 0.020 for 1¼-in. and 1½-in. blades
= 0.025 for 1¾-in. and 2-in. blades.

The inlet angle of a blade is determined by the angle the steam makes relative to the blade at entrance, which is shown in the velocity diagrams on p. 574.

Theoretically the front edge of the blade is tangent to V_2, the relative entering velocity shown in Fig. 67. The rear face of the blade is also parallel to

V_2 and at a distance t from the front face at the edge of the blade. In practice this is subject to some modifications. Some designers increase the angle α about 5 deg. This is done to allow for inconsistencies in design and construction, low steam pressure, etc. It has been found in practice that serious losses occur if the relative velocity at entrance is such that the steam strikes

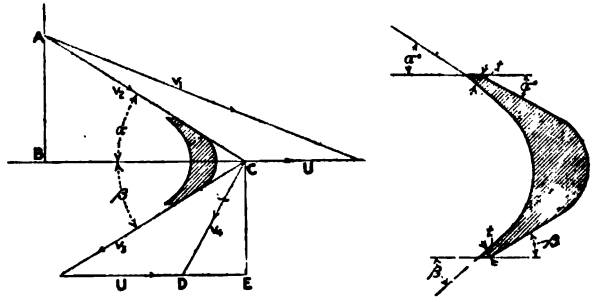


FIG. 67.

the rear of the blade. Or in other words, if the blade is running at too high a speed for the jet velocity, the losses are high.

On the other hand, the loss from shock is very slight if the steam strikes the face of the blade at a slight angle as has been shown by experiments in England where it was found that the inlet angle could be considerably too wide without serious results. Furthermore the wider inlet angle allows for occasional low steam pressures and other incidents of operation without causing serious decreases in efficiency. Hence, most manufacturers of marine turbines now make the blade inlet angle somewhat larger than the velocity diagram demands.

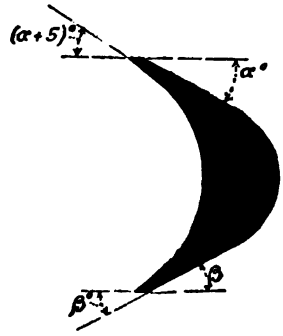


FIG. 68. — Impulse Blade with Increased Inlet Angle.

Some engineers have modified the design by having the front edge tangent to an angle of $(\alpha + 5)$ deg. while the rear face makes an angle of α deg. with the axis as shown in Fig. 68. This is not so good as increasing the angle on both front and back.

The amount of work done per stage increases as the outlet angle of the blade decreases. At the same time the end thrust on the machine is increased. But the velocity of the steam is decreased when crossing the blade and the leaving velocity is less than the entering. Hence a greater outlet area must be provided. If the outlet angle decreases the blade height at exit must therefore be increased. The amount that this angle can be economically decreased in design varies from 10 to 15 deg. In general the outlet angles in blades of impulse turbines vary from 16 deg. to 35 deg. and may be larger in Curtis wheels and in the last rows of a high vacuum turbine.

The back curve of a blade is settled by consideration of eddy losses. In Fig. 69, *A* shows thin sheet blades and it is apparent that eddies will form

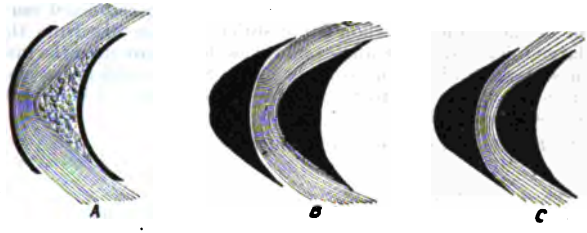


FIG. 69.—Influence of Back of Blade on Flow of Steam Jet.

behind the blade, due to compression of the steam and also because the passage is one of irregular width. *B* shows a blade with equal inlet and outlet angles and a constant width of passage. This is a normally designed blade. *C* shows a blade with steam at a very high velocity that causes the steam to compress slightly on the face of the next blade. The blade has extra thickness in the middle to prevent eddies forming due to this compression.

The section of the blade can be laid off from a velocity diagram after certain assumptions are made. It should be drawn twenty or fifty times full size so that all dimensions can be accurately scaled. Even then it is well to check all these calculations mathematically. The following method has been used to lay off impulse blades and to check the drawing by calculation.

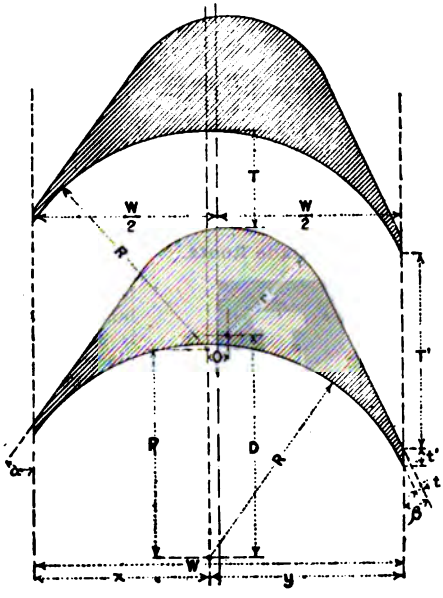


FIG. 70.—Design of Impulse Blade.

Assume that the width of blade *W*, Fig. 70, the inlet angle α (opened wider than the velocity diagram if desired) and outlet angle β and the thickness *t* of the edge of the blade are given, also *T* the ratio of width of passage to radius of front of blade measured on the centre line of blade is given. *T* varies with the curvature of front of the blade, with the width of the blade and with the steam velocity. Different designers use different ratios. In general *T* varies

from $0.35R$ to $0.7R$ in the last rows of Curtis stages and of Rateau turbines, where R is the radius of curvature of the front of the blade. T is modified by the pitch and this is figured to get the proper number of blades per row in the blade ring. In general the pitch approximates the radius of curvature of the front of the blade and should not differ very greatly from that figure. Where large inlet and outlet angles and wide blades are used the pitch cannot be taken so large and seldom exceeds 1 in. Some builders have a rule that blade pitch should be 0.5 to 0.6 of the blade width. This checks closely with the above method.

T = % R % assumed for made width.

$$R = \frac{W}{\cos \alpha + \cos \beta}$$

$$x = R \cos \beta$$

$$y = R \cos \beta$$

$$t' = \frac{t}{\sin \beta} \qquad m = \frac{\alpha - \beta}{2}$$

$$T' = \frac{T - t \dfrac{\sin m}{\sin \beta \cos (\beta + m)}(1 - \sin \beta)}{1 + \dfrac{\sin m}{\sin \beta \cos (\beta + m)}(1 - \sin \beta)}$$

$$P = \frac{T' + t}{\sin \beta}$$

$$r = R - T' - \frac{P \sin m}{\cos (\beta + m)}$$

O = offset of centre r from R

$$= \frac{P \sin m \cos \beta}{\cos (\beta + m)}$$

D = distance of center r from $R = P + \dfrac{P \sin m \sin \beta}{\cos (\beta + m)}$

Forms of Blade Roots. The usual form of blade root for impulse blades is shown in Fig. 71. This form is subject to many minor modifications by various builders. One important feature of design is to avoid sharp corners. Westinghouse places a double wedge on one side of the blades to make them tight sideways in the grooves.

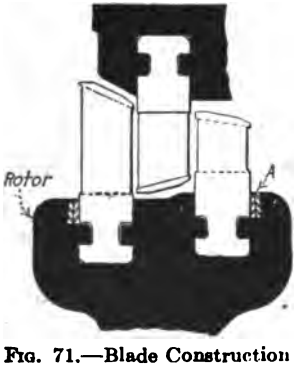


FIG. 71.—Blade Construction of Velocity Stage.

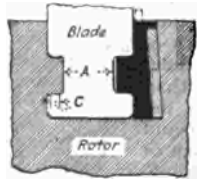


FIG. 72.

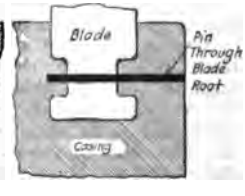


FIG. 73.

Various devices are used to close up the opening in the slot through which the blades are fed, one form of such closure being shown in Fig. 72 where

the edge of the wedge is peaned down in place after all parts are inserted. The first and last blades of the stationary row of a Curtis stage are usually pinned, as shown in Fig. 73.

Blades are often machined from drawn bar and the base is left of such size as to act as a distance piece, as shown in Figs. 74 and 75. Some other designs have distance pieces that fit in the slots between blades, and thus provide the necessary spacing. Where it is desired to keep the rotor disc as light as possible and particularly where high blade speeds are used, the General Electric Co. uses the form of blade root shown in Fig. 76.

The stresses the blade must withstand are tension and crushing. The tensile stress from centrifugal force must be calculated at the thinnest sections,

FIG. 74.

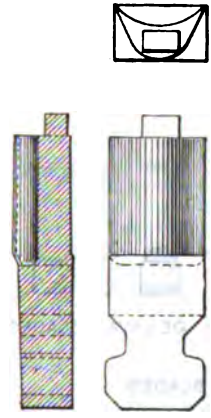


FIG. 75.

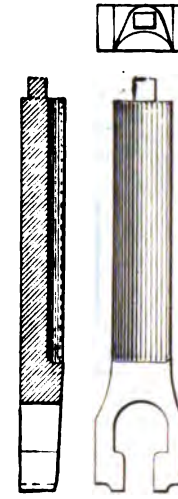


FIG. 76.

as at *A*, Fig. 72 and at *B*, *B*, Fig. 77. The tensile stress in the blade section itself may also need checking if the shroud section is heavy and the blade narrow and of small cross-section. The tensile stress should not exceed 15,000 lbs. for bronze or 20,000 lbs. for soft steel, nickel steel or monel metal.

The crushing stress must be figured at *C*, Fig. 72 and at *D*, Fig. 77. In calculating this stress, the radius of curvature in the corners should not be considered. Thus take Fig. 78 which shows the left-hand portion *D* of Fig. 77. In order that the blades may drive easily over the shank on the rotor disc, there must be some clearance all around and particularly at the sides. When subject to centrifugal force the blade is firm against the disc shank. The fillets however do not necessarily come into contact and hence for calculations of strength these should be neglected and the section *D* only assumed to carry the crushing load. However, if stretch does occur, the fillets may come in contact and thus increase the factor of safety. But it is not safe to count on this fillet support in the original calculations.

Blades of DeLaval turbines are usually drop forged and the bulb and shank afterward accurately machined as shown in Fig. 79. These blades

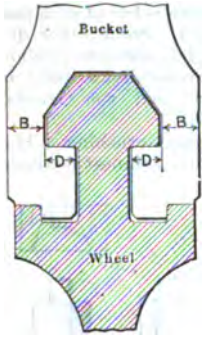


Fig. 77.

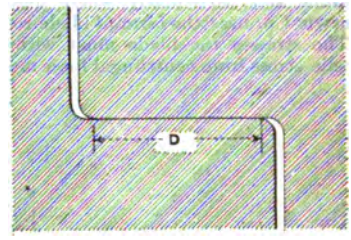


Fig. 78.

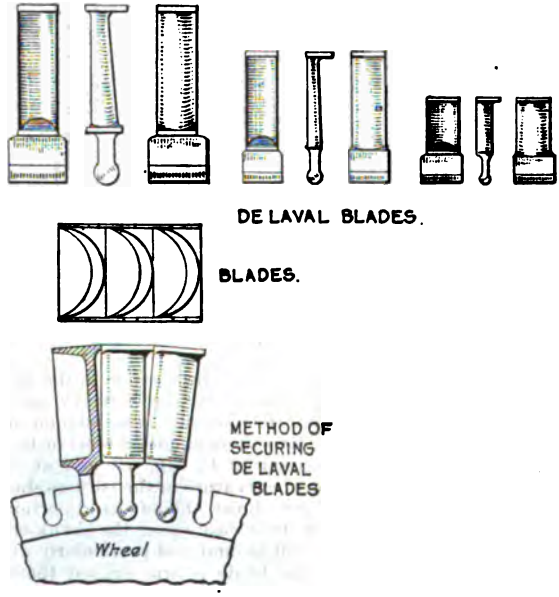


Fig. 79.—DeLaval Blades.

fit into slots in the wheel and are peaned in place. The base flange and shroud flanges are forged integral with the blade.

Shroud. The simplest form of shroud is to place a slotted flat shroud ring, as shown in Fig. 80, over a tit on the ends of the blades and rivet over this tit as shown. Some manufacturers used this form of shroud on blades whose outlet edge was longer than the inlet edge. This made the tit on the the blade subject to bending as well as centrifugal stresses and as a result a number of failures occurred. **General Electric Co.** now favor only flat shrouds parallel to the shaft axis and sacrifice some theoretical blade efficiency for rugged construction.

Spot welding is sometimes resorted to instead of riveting and a stronger shroud is procured. **Westinghouse** use a very ingenious shroud ring on their Curtis blades as shown in Fig. 81. The blades have shoulders milled at the end to provide spacing. These projections after assembly have a dovetailed groove turned in the end. A soft steel shroud strip *B* in short lengths is hammered down into this groove and spot welded in place. This forms an excellent binding and provides great rigidity.

In Curtis stages it is usual to increase the inlet edge of each row of blades about $\frac{1}{16}$ in. greater than the outlet edge of the preceding row to allow for spreading of the steam in crossing the clearance between blades. This is shown in Fig. 71.

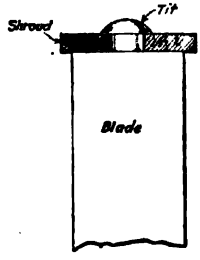


FIG. 80.

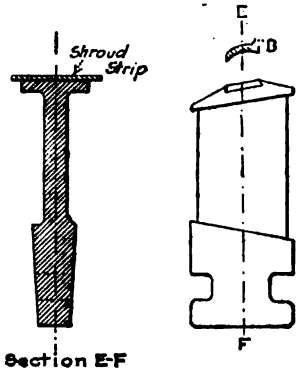


FIG. 81.—Westinghouse Impulse Blading.

The distance between the edge of a shroud and the base of the next blade in a Curtis stage is usually about $\frac{1}{16}$ in. and should not exceed $\frac{1}{8}$ in. Shrouds and bases usually project about $\frac{1}{32}$ in. over the edges of the blades for mechanical protection.

Minimum Blade Length. This should not be less than $\frac{1}{4}$ in. If longer blades are used the design of the nozzle should be modified to secure the greater length. Another plan of the General Electric Co. is to use smaller diameter wheels on the high pressure end of the turbine so that longer blades can be used.

Clearances Between Impulse Blades. The end clearances are not a matter of importance and may be fairly large. The side clearances should be kept as small as possible to maintain full stream lines from nozzles and blades and to reduce eddies and friction. The figures given above for shroud clearances largely determine the amount of this axial clearance.

Steam Leakage and Reverse Operation. Serious attention should be given in marine turbines to the effect of steam leakage on sections of blading

running reversed. For instance, suppose the main throttle valve was leaking and the turbine was running full speed astern when maneuvering out of harbor; it has been found by calculation that dangerously high temperatures may be generated in the forward blading under such conditions and at full speed the high centrifugal stresses may cause serious damage to the blading. In fact some accidents have been attributed to this cause. There is a certain amount of evidence that such heating actually occurs, for the steel blading of certain stages will often be found to have a blue tempered appearance on opening up.

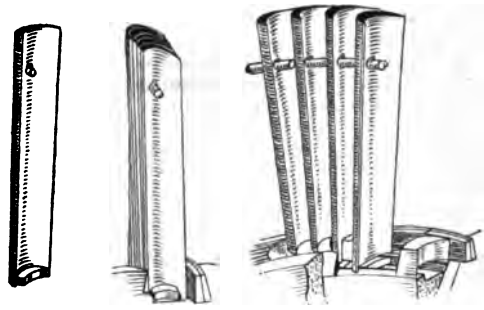


FIG. 82.—Westinghouse-Parsons Blading.

Reaction blading does not conform as closely to theoretical considerations as does impulse blading. The outlet angle of the blades is varied according to the gauging of the blades. The usual angle varies from 18 to 22 deg., averaging 20 deg. The tangent to the curve of the blade face at the inlet edge usually conforms to the relative angle of inlet on the velocity diagram and generally is about 85 deg. On the last rows of Parsons turbines so-called "**semi-wing**" blades with outlet angles of about 35 deg. are used and also "**wing**" blades with angles as large as 50 deg. have been used on the last rows of some Parsons turbines. These however entail a large leaving loss due to the high exit velocity, but allow high vacuums to be carried at heavy loads.

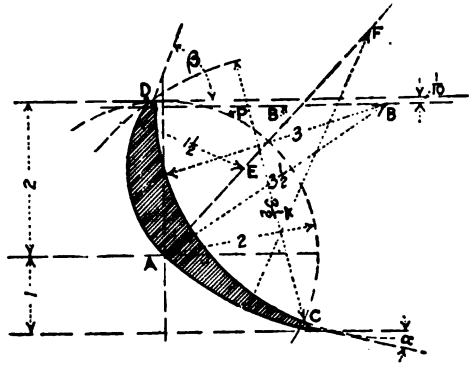


FIG. 83.

The form and thickness of the blades beyond these two angles are subject to much variation and little experimental work has been done to determine the most efficient form, each builder preferring his own particular design. Fig. 82 shows the blading of Westinghouse turbines.

Martin has suggested the following method of laying out Parsons blading:

Draw two parallel lines 3 units apart, as shown in Fig. 83. At point *A* distant 1 unit from the bottom line, describe the arc *DPC* with radius 2 units. Draw the line *BB'* distant $\frac{1}{10}$ unit from the top line and find point *B* at a radius of $3\frac{1}{2}$ units from *A*. With *B* as centre and radius 3 units, draw the

face of the blade cutting arc *DPC* at *C*. With *C* as centre draw an arc with 3½ units radius to cut arc *DPC* at *D*. Draw an arc *DA* from centre *E* with 1½ units radius to pass through *D* and *A*. Produce *AE* to *F* and with *F* as centre and 4 units as radius, draw arc through *A* forming the remainder of the back curve of the blade. The bisectors of the tangents to the two curves at inlet and exit gives the inlet and exit angles β and α.

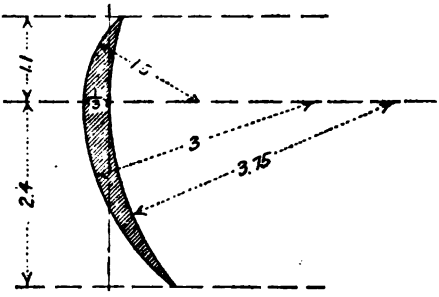


FIG. 84.

Martin shows the method of designing full wing blading as illustrated in Fig. 84. Blade pitch seldom is less than 0.2 in. and even with the longest blades generally does not exceed 1 in. The usual sizes of blades vary in pitch from 0.25 in. to 0.7 in. A rule followed by some builders of Parsons turbines is to make the pitch of the blades from 0.6 to 0.7 of the blade width.

Parsons uses the form of blades shown in Fig. 29. The blades are drawn to form in specially constructed dies. They are cut to length and indentations milled near the base, as shown. Distance pieces of very soft metal also cut from drawn bars form the spacers between blades. The grooves in which the blades and distance pieces are inserted are slightly dovetailed or have small grooves cut in them as shown. After the whole ring is in place, the distance pieces are caulked down into the dovetail or into the small grooves by a special caulking tool.

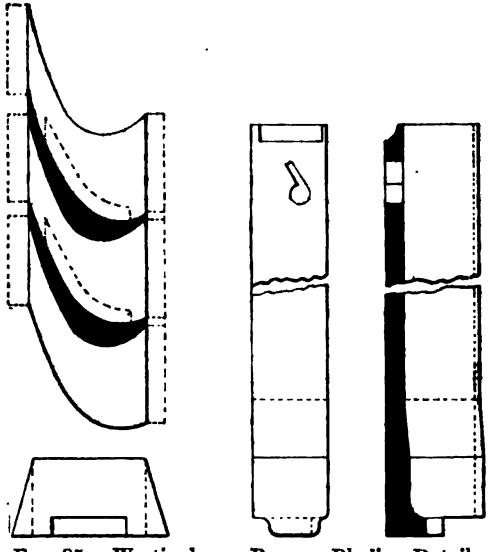


FIG. 85.—Westinghouse Parsons Blading Details.

In Parsons blading one of the greatest causes of accidents is the failure of the blade material from repeated stresses due to harmonic vibrations set up by the steam. Every means is taken in modern designs to prevent this. Further developments in Parsons blading will largely be concerned with means of reducing this to a minimum.

Vibration of the blades is prevented by a reinforcing wire either threaded through a hole in the blade or inserted in a slot in the side. This wire is tied to each blade by a fine copper wire and silver soldered in place. This lashing also maintains the pitch of the blades at their outer ends.

Many manufacturers also thread the roots of the blades and distance pieces onto a wire and this serves also to reinforce the blade ring. This is known as the **"rosary" construction.** In such a case the blades are made up in segments and afterward inserted in the turbine.

Practically all Parsons blades are thinned down at their tips as shown in Fig. 29 to avoid injurious effects to spindle and casing if rubbing should occur.

Westinghouse use a more highly developed form of blade, shown in Fig. 85. The blades are thickened near the base by an upsetting operation. This makes them less subject to vibration troubles. A small projection is also upset on the bottom of the blade. This foot fits under the distance piece which is dovetailed and is held in place by the wedges shown on the side. No caulking is necessary. It is quite improbable that the blades will pull out of place. The outer ends are reinforced by a comma-shaped wire with the tail bent over between blades by a special spacing tool after assembly is complete. Long blades have several of these reinforcing wires. The tips of the blades are thinned off as shown.

Allis-Chalmers place their blades in a specially designed mold and cast the metal forming the foundation ring around the base of these blades. The blades are made in sections with their bases firmly imbedded in this solid foundation ring. This ring is then machined and is held in a dovetailed groove by a soft metal packing strip which is calked down in place as in Fig. 28. The outer end of each blade is silver-soldered to an indented channel-shaped shroud ring. These shroud rings provide thin wearing edges in case of a rub and also maintain accurate spacing of all blades. Blades three inches and over are also provided with one or more reinforcing wires silver soldered to the blades to prevent vibration.

Where short compact light weight disk and drum turbines are desired, designers have resorted to high blade speeds and long blades in the low pressure stages. These long blades are then made with ⊥ shaped bases like Curtis blades and are either drop forgings or are machined from the solid metal. The base is of such size that it forms its own distance spacer. These blades are either of mild steel or of electric nickel-steel.

Materials. Impulse blading material may be soft steel, electric furnace nickel steel, monel, or brass alloy. The blades may be drop-forged as in the DeLaval turbine, may be cut from extruded bars as in low speed units, or may be machined from forgings or bars as in most machines where the wheel speeds are high.

Interesting data on the performance of Curtis-Rateau blading is included in a paper on "The Deterioration of Curtis-Rateau Turbine Blading" by A. Fenwick, Engineering, May 7, 1917. He concludes that 25 per cent nickel steel is absolutely unsuited for blading, as it rapidly becomes brittle. Aluminum bronze was found unsuitable for stages where the temperature was high, although suitable for low pressure stages. Mild steel proved quite suitable for high temperature sections. Monel metal and brass were also satisfactory.

In general, electric furnace carbon steels and 5 per cent nickel or chrome steels make satisfactory high temperature blading. Phosphor bronze is suitable for both high and low temperatures and brass or bronze works well in the low pressure end. However, with high blade speeds, steel is also used in the low pressure sections.

Table 14. Blading Data of 3000 S.H.P. Westinghouse Compound Turbine

		Stage No.	Type	Mean diam.	No. of rotating rows	Height in inches: Inlet	Height in inches: Outlet	Radial clearance	Blades per row: Cylinder	Blades per row: Spindle	Per cent gauging	Abs. press. at full power	Dummy mean dia.
Ahead Turbine	H.-p. turbine	1	Impulse	32	1st row rotating	.652	¾	..	.....	200	.....	185 to 69	15⅞
					Stationary	.863	1¼	..	107	.....	.....		
					2d row rotating	1.352	2⅛	..	.....	200	.....		
		2	Reaction	19	8	2		.025	194	180	17 to 19	69 to 35	
		3	Reaction	20	6	3		.030	196	206	19 to 30	35 to 19.5	
	L.-p. turbine	1	Reaction	27	3	3		.030	258	258	17 to 25½	19.5 to 10.5	24
		2	Reaction	28	2	4		.035	210	224	20 to 27	10.5 to 7	
		3	Reaction	29	1	5		.045	222	224	22 to 24	7 to 5.5	
		4	Reaction	30	3	6		.045	150	145	21 to 39	5.5 to 4	
		4	Reaction	30	1	6		.045	188	182	44 to 50	4 to 1	
Astern Turbine	H.-p. turbine	1	Impulse	32	1st row rotating	.652	¾	..	.....	200	.....	185 to 20	
					Stationary	.863	1¼	..	107	.....			
					2d row rotating	1.352	2⅛	..	.....	200			
	L.-p. turbine	1	Impulse	32	1st row rotating	2.488	3⅝	..	.....	118	.....	20 to 1	
					Stationary	4	4	..	142	.....			
					2d row rotating	4⅛	4⅛	..	.....	179			

Reaction blades are made of various brass and bronze mixtures, of delta metal and of nickel-copper alloys. The latter material, though expensive, appears to stand up well and to resist corrosion. The material is usually drawn to form and cut off into lengths as required. Distance pieces where used are usually of a soft brass alloy. Lashing wires are generally of brass. Shrouds are usually of steel, though brass has been sometimes used.

The following **specifications for blading material** may aid in the selection of a proper metal:

Nickel-steel, with nickel not exceeding 3 per cent, sulphur not over 0.05 per cent, phosphorus not over 0.04 per cent, ultimate strength 90,000 lbs., elastic limit 50,000 lbs., elongation not under 20 per cent and reduction in area not under 40 per cent.

Monel metal, with 72 per cent nickel, 26½ per cent copper and 1½ per cent iron; ultimate strength 90,000 to 100,000 lbs., yield point 44,000 to 50,000 lbs., elongation 30 to 40 per cent.

Extruded brass has a yield point about 29,000 lbs. and should not be stressed over 15,000 lbs. in a machine.

Table 14 gives the complete blading data of a 3000 H.P. compound Westinghouse marine turbine.

Cylinder Construction

Cylinders of marine turbines are split horizontally so that the upper half can be lifted for inspection and repairs. The construction of these two halves varies greatly with the make and type of turbine and also with the operating conditions. Some different constructions are shown under the section dealing with "Types of Turbines," p. 732.

In some cases each half is divided into two or more sections bolted together around the circumference, usually with tongue and groove joints. This construction ensures good sound castings. Cylinders are usually of cast iron. The high-pressure end is often cast steel, particularly when a Curtis wheel and nozzles using high-pressure and highly superheated steam are employed. Cast steel is also used in some high-pressure sections of Parsons turbines. **Dummy rings** on Parsons turbines are sometimes cast separately, and bolted to the casing after machining. **Nozzle blocks** of bronze or cast steel are almost invariably made up separately and bolted to the cylinders. **Diaphragms** of impulse turbines are usually cast separately and after machining are fastened to suitable grooves in the upper and lower halves of the casing.

The Kerr turbine, however, is made up in sections, each section consisting of a diaphragm ring with flanges at the outer end which when bolted together form the cylinder of the turbine. All are split horizontally so that the top half can be lifted for inspection. This construction is shown in Fig. 218.

Diaphragms, particularly when made in halves, call for very careful design to avoid deflections from difference of pressure. Many turbine troubles have been traced to dishing or breathing of diaphragms in operation. This is partly due to pressure changes but often due to sudden changes of temperature. The problem is aggravated by the fact that the outer edges of the diaphragms form the orifices for the next set of blades. These orifices are built by placing formed sheet-steel partitions in the cores and casting the metal around them. The steel partitions are less rigid than a solid cast iron wall and hence the diaphragm is more liable to strain under sudden changes of temperature and pressure. Diaphragms are generally made as heavy as possible and are dished as much as possible toward the high-pressure end of the turbine to provide additional strength.

Another source of trouble with cast iron diaphragms is the growth of the metal under the influence of high temperatures. This causes the diaphragm to be distorted and in certain cases has caused the failure of the metal partitions. Many builders make all their high pressure diaphragms of cast steel when superheat is used with carefully selected cast iron for the exhaust stages only

Many marine turbines are built with high- and low-pressure sections and with astern elements in both sections. With such construction a diaphragm must be placed in the high-pressure casing between the forward and astern elements. This diaphragm will be subject to great differences of temperature and pressure and must be very ruggedly designed. It must also be provided with more dummies than the ordinary diaphragm between stages.

Casings. In the design of casings rigidity is usually more important than strength. Rigidity together with a consideration of the needs of the foundry and the machine shop often determines the thickness. Casings are sometimes cast smooth on the outside merely with flanges to bolt up the two halves. Often heavy ribbing is used, both in circumferential and longitudinal directions. Great care must be taken that these ribs and the flanges are not made too deep, for they would then cause distortion from temperature changes. The outer edges of such projections would not respond quickly to such changes and the casing would deform.

Large abrupt changes in diameter must also be avoided for such changes would likely produce a "breathing" action such as takes place in diaphragms. The resulting distortions are of a very serious nature.

Turbine cylinders are now made simpler and with fewer changes of form and diameter than on the earlier turbines. In fact modern designs all approach the barrel form. Separate blade rings may be used to advantage in the cylinder when the extra weight is not prohibitive. This plan materially simplifies cylinder design.

The cylinder wall must be figured to safely withstand a bursting stress in its thinnest section of 50 per cent more than the maximum working pressure in the cylinder. For average designs this may be taken as 200–250 lbs. on high-pressure cylinders and 30–50 lbs. on low-pressure cylinders. There is also the question of deflection of the casing between the supporting pedestals. This deflection is affected in some cases by the upward, downward or side pull of the vacuum in the exhaust pipe when an expansion joint is placed between the turbine and the condenser. An exhaust outlet on the top of the casing may thus counterbalance deflection from weight alone.

Large exhaust nozzles and ends of turbines are stiffened and braced in some designs by heavy steel stay bolts and pipe ferrules inside the casing. Some large Parsons turbines with high- and low-pressure cylinders and which are furnished with highly superheated steam at high pressure, have specially constructed high-pressure cylinders. An inner cylinder of cast steel carrying the blading is fastened at certain points to an outer cast iron cylinder that forms a steam jacket around the steel cylinder. Distortion from temperature changes is thus avoided.

Cylinders should in every case be steamed at 20 to 25 lbs. pressure for about 24 hours before final boring.

Provision should be made for a suitable number of dowel pins in each section to ensure accurate alignment in reassembling.

Pressure equalising pipes outside the casing with provision for expansion are now used in place of passages cast integral with the cylinder itself.

The astern casings may be cast as part of the main turbine but are usually

cast separately and bolted in place as shown in various commercial designs.

Care must be taken in the design of the cylinder supports. In general the base of these pedestals should be as near the center line of the shaft as possible, or the supports must be made so that they are uninfluenced by heat changes in the main cylinder. Otherwise the alignment of the shaft will be affected by changes of load and vacuum. The pedestal at the low-pressure end is generally bolted down tight. Provision is made at the high-pressure end to allow for expansion or contraction either by holding down gibs allowing longitudinal movement or by oval holes and bolts screwed up to a shoulder and permitting movement in the base.

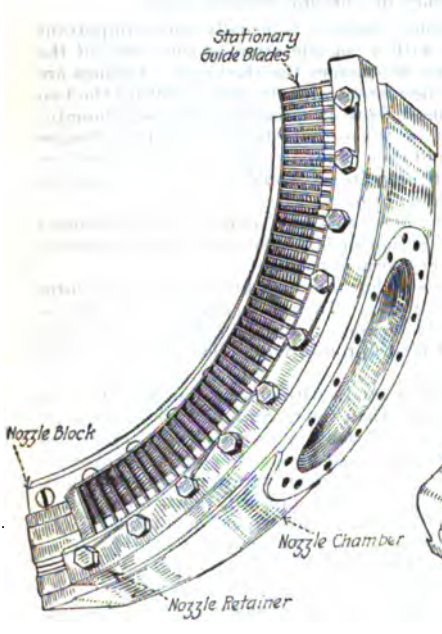


FIG. 86.

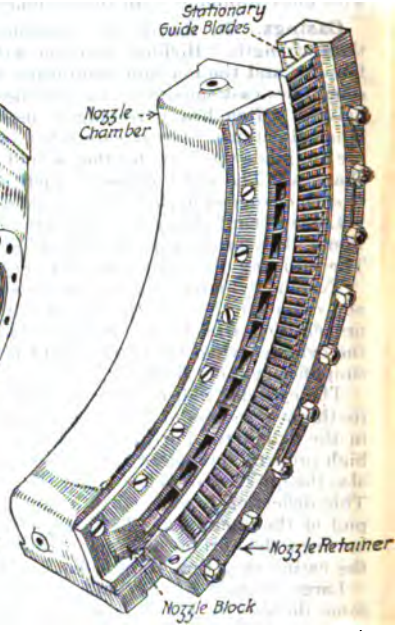


FIG. 87.

Nozzles. The first sets of nozzles on a turbine are invariably built up in the form of a nozzle block separately from the casing and firmly bolted to it. Figs. 86 and 87 show the first Westinghouse nozzle and stationary guide blades for the first impulse stage of their Curtis-Parsons type. This nozzle block is fastened to the casing by the steam inlet bolt holes shown in Fig. 86 and is thus free to expand and contract independently of the main casing. When the high-pressure end is a steel casting, the nozzles are often fastened directly to ports in the casting instead of to a nozzle block as shown. Some nozzles are formed by drilling and filing to shape. Many Westinghouse nozzles are built up and are accurately machined before assembly. One form of nozzle in general use is shown in Fig. 17. These passages are

accurately milled in the nozzle block and an outer shroud or flange is fastened on and held in place by the retainer shown in Fig. 87.

General Electric Co. drill and file all their first nozzles to shape and polish carefully. Their nozzle block is a single bronze casting which bolts against the steam inlet space.

Nozzle blocks are usually fitted in pairs on the first stage of Curtis-Rateau turbines, one block on the upper half and one on the lower half of the casing, so that there is no unbalancing of effort on the rotor. The diaphragms following are arranged to conform with this design, allowing for the carrying forward of the steam in the direction of rotation. This arrangement on the high-pressure end requires either a steam belt around the front end of the casing or two steam-supply connections. Some manufacturers do not favor such construction for high pressures and high superheat and hence use only one nozzle block.

Nozzles may be designed as outlined on p. 538. The section from throat to mouth is usually formed by straight lines instead of conforming to theoretical dimensions. The efficiency of nozzles varies from 85 to 95 per cent depending on the pressure drop and the relation of designed proportions to those in the actual nozzles.

Nozzles are now usually of rectangular cross-section as this gives fullest steam flow at the blades. The throat is made as nearly square as possible. The radial width of the mouth is chosen such that it will give sufficient length of blades in the first row, generally ¼ in. or ¾ in. If possible, the mouth is also made square at right angles to the flow, as well as the throat.

Nozzles of circular cross section are still used in De Laval marine turbines. These are also used very extensively in small turbines for ship auxiliaries. Some forms of marine turbine have the inlet mouth in bowl form which is utilized as the seat of a valve to open or close off the nozzle when hand regulated

Some foreign designs of marine turbines have the high-pressure nozzles inclined toward the axis of the shaft so that the jet discharge is not parallel to the axis. This is done in order to get a more effective tangential component of the velocity as the steam flows through the revolving blades.

Nozzles are made of bronze to avoid corrosion and are highly polished to lessen friction losses. The nozzles or orifices in the diaphragms usually consist of sheet material of nickel steel or other alloy bent to the proper form and cast in place in the diaphragm body. Some builders, however, make these up in nozzle blocks and bolt to the diaphragms. Such partitions are seldom less than ¼ in. thick and in the low-pressure sections must be sufficiently thick to avoid vibrations when their radial length becomes great. The inlet edges of these partitions are usually ground to a sharp edge to lessen steam shock at admission. The cast iron forming the edges of these orifices is chipped, filed and polished to reduce friction losses.

It is evident that orifices formed in such a manner are not free from eddies and losses. If it were therefore possible to insert partitions of correct crescent form in place of the sheet metal plates, the turbine efficiency would be improved thereby. Such construction is quite feasible where solid diaphragms are used, as in land turbines, but is not so easily done on split diaphragms.

DeLaval constructs many solid diaphragms by turning the outer edge and inserting bronze partitions of drawn cross-section in suitable grooves on its outer edge. These form the walls of the desired orifice and can be of any desired form. They are held in place by shrinking a ring over their tips. The orifice is thus theoretically correct and is said to give a much higher efficiency than the usual form with cast-in partitions.

The radial length of the passage through the orifice will be governed by the calculations given on p. 542 for "orifice area." A question arises in regard to the spacing of diaphragm partitions. If these are too close together friction losses will be excessive. If too far apart suitable direction will not be imparted to the jet and it will tend to spread. The correct distance is a matter of judgment. Common practice in designing orifices is given in Table 15. The length of orifice is the radial length. The width w is measured at right angles to the jet and the circumferential width $\left(w' = \frac{w}{\sin \alpha}\right)$ is that measured along the circumference of the diaphragm. The thickness of the diaphragm varies from $\frac{1}{16}$ in. for very short orifices up to $\frac{1}{4}$ in. or more for long orifices.

Table 15

Radial length of orifice in in.	Thickness t in in.	Width w in in.
½ to 1½	1/16	½
1¼ to 4	1/8	½
4 to 8	1/4	¾
8 up	3/16 up	1

Turbine Drains

The presence of condensation inside of a turbine is always a source of possible trouble and arrangements have to be made to keep the turbine casing and connected pipes thoroughly drained at all times.

High pressure drips from the main steam pipe above the throttle valve can usually be taken care of by a steam trap. The tilting bucket type appears to be most favored for marine service.

Cylinder Drains. The cylinders of marine turbines must be properly drained to remove condensation, particularly when the exhaust pipe leaves the top of the casing. This is one of the most important operating considerations in marine steam turbine practice. Lack of proper drainage has been known to have caused many of the accidents to turbines and in the average case this has been due to gross negligence. There is still room for the design of a simple automatic trap device that will positively remove such condensation under all possible operating conditions.

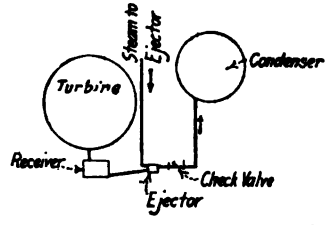


FIG. 88.—Cylinder Drain for Marine Turbine.

In the case of a **single cylinder impulse turbine**, the exhaust can be connected through a check valve to the suction of a wet air pump if used, provided this pump is at a lower elevation than the lowest part of the casing. When dry air pumps or steam ejector pumps are used the best plan is to place a small receiver below the casing and provide an ejector to discharge this water up into the condenser itself, as in Fig. 88. This must be discharged at regular intervals on each watch.

In the case of **high and low pressure turbines**, it is necessary to drain not only the exhaust end of the L.P. cylinder but also both forward and astern exhaust ends of the H.P. cylinder, as either of these ends will be under vacuum when the other is in operation. Cross-over piping unless placed above the turbine must also be drained.

Traps cannot be used and one must therefore resort to receivers and ejectors as in the case of the single turbine described above. These are shown in Fig. 89. It is best to have several receivers unless the ejector is kept in continuous operation. In that case all drains can connect to a common header leading to the ejector.

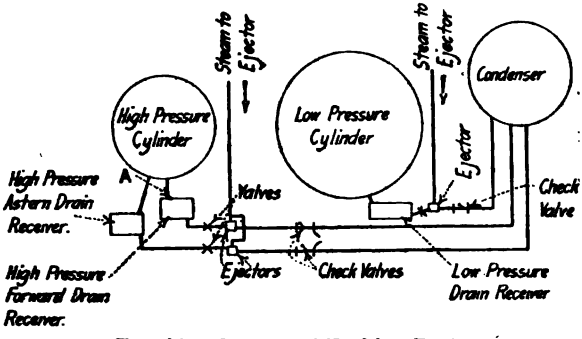


FIG. 89.—Compound Turbine Drains.

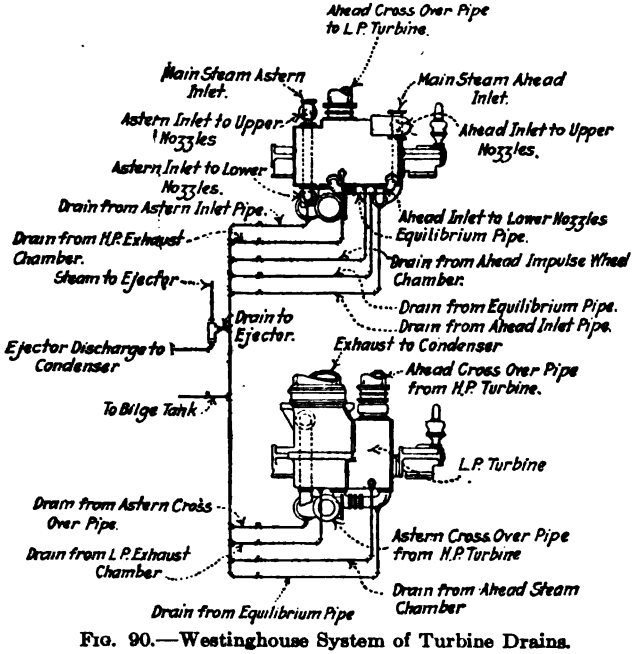

Ahead Cross Over Pipe to L.P. Turbine.
Main Steam Astern Inlet.
Main Steam Ahead Inlet.
Astern Inlet to Upper Nozzles
Ahead Inlet to Upper Nozzles.
Astern Inlet to Lower Nozzles.
Drain from Astern Inlet Pipe.
Ahead Inlet to Lower Nozzles
Drain from H.P. Exhaust Chamber.
Equilibrium Pipe.
Steam to Ejector
Drain from Ahead Impulse Wheel Chamber.
Drain from Equilibrium Pipe.
Drain to Ejector.
Drain from Ahead Inlet Pipe.
Ejector Discharge to Condenser
Exhaust to Condenser
Ahead Cross Over Pipe from H.P. Turbine.
To Bilge Tank
L.P. Turbine
Drain from Astern Cross Over Pipe.
Astern Cross Over Pipe from H.P. Turbine
Drain from L.P. Exhaust Chamber
Drain from Ahead Steam Chamber
Drain from Equilibrium Pipe

FIG. 90.—Westinghouse System of Turbine Drains.

When running full speed ahead there will probably be sufficient pressure in the H.P. forward drain receiver to force the water into the condenser as soon as the discharge valve is open. The same is true with the H.P. astern receiver when running astern. At all other times the ejectors must be used to free the cylinders from water.

With **Reaction turbines** the arrangement is slightly different from the systems shown above. Where a single turbine is used drains must be provided from the ahead impulse wheel chamber, from the exhaust chamber and from the equilibrium pipe joining the exhaust end to the forward side of the balance piston. Westinghouse practice is to carry all these to a common header connected to an ejector which delivers the drips into the condenser.

On the H.P. cylinder of **Westinghouse cross-compound units** drains must be taken from the ahead impulse wheel chamber, the ahead exhaust chamber and the equilibrium pipe which lies directly under the turbine. These are carried to a common drip pipe leading to the ejector. The drip pipe picks up the drains from the ahead inlet pipe and the astern inlet pipe.

As the turbines are operated in the ahead direction by far the greatest part of the time, it is best practice to locate the ahead cross-over pipe above the turbine so that there is no chance for water to collect in it. The astern cross-over pipe is then placed beneath the turbine and a drain from it is very essential.

Drains in the L.P. section are taken from the ahead steam chamber, from the exhaust chamber and from the equilibrium pipe. Fig. 90 shows the complete drain system for a cross-compound unit.

All drains are connected through individual valves into a common header. Either an ejector or lifting trap with a receiving tank may be used to lift the collected water into the condenser. About 50 lbs. of steam per hr. will be consumed by such an ejector when working continuously. There is also a common drain from all drips to the bilge which, with all the drain valves, should be open when the unit is idle. This prevents any accumulation of water that would start corrosion inside of the turbine. The valves on the drains and drain line should be arranged with extension stems which are easily reached from a point near the operating platform.

Before starting up such a turbine, all drains should be opened and the ejector put in operation. When it is reasonably sure that all parts are free of water, all the drains can be closed with the exception of the one on the astern cross-over pipe below the turbine, the drain from the L.P. equilibrium pipe and the one from the exhaust chamber of the L.P. turbine. These should be kept open all the time to prevent a gradual accumulation of water which would eventually give trouble. In order to reduce the amount of steam which would flow through the open astern drain cross-over pipe drain on reversing a small hole may be drilled through the drain valve. The valve may then be kept shut when the unit is in service. The necessity of keeping the drains open is notably true in the case of the L.P. exhaust chamber. If the reversing wheel does not extend to the very bottom of the case, considerable water may accumulate when running ahead. Then if the turbine is suddenly reversed a slug of water will be thrown back into the ahead blading. This is almost certain to cause trouble.

In addition to these drains on the main turbine unit, there are, of course, the usual **drains on the steam piping to the auxiliaries.** These may be taken care of by traps. Drains from the auxiliaries themselves can be handled in the usual way.

Dummy Pistons, Glands, Labyrinths

End Thrust. The discs of **impulse turbines** revolve in chambers with equal pressure on each side of the disc. Hence there is no end thrust due to unbalanced pressures provided that the shaft is substantially the same diameter at the steam and exhaust glands. Any axial force due to inequality of the axial components of the inlet and exit velocity of the blades caused either by friction or by closing the exit blade angle is very small and can be easily handled by the usual thrust bearing described in the next section.

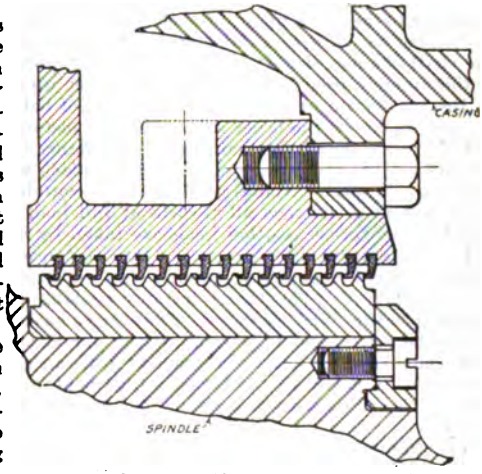


FIG. 91.—Balance Piston with Axial Clearance.

End thrust must be carefully considered in **Parsons turbines.** First, there is a cumulative pressure due to the drop in pressure crossing each row of blades and second, there is also an axial thrust due to the pressure on the end of each drum when the diameter changes. This total thrust can be balanced either by a single thrust block or in direct-connected turbines by the thrust of the screw or by equalizing the thrust by building a balance piston of such area that an equal opposing thrust is set up.

Generally the force due to the unbalanced axial component of the velocities entering and leaving the reaction blades may be neglected as it amounts to only 1 or 2 per cent of the total thrust.

Considerable thrust is now carried by the thrust blocks so that the diameter of the balance piston can be reduced. This results in a decrease in dummy leakage and an increase in turbine efficiency.

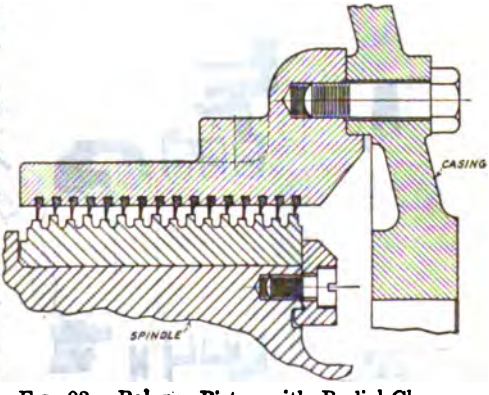


FIG. 92.—Balance Piston with Radial Clearance.

The **reaction turbine** consists of sections of blades of the same mean diameter and height.

The **area of the required balance piston** can be found by the following formula: $A_p = \frac{\Sigma A_m P_d + A_a P_a - T}{P_i - P_e}$, where A_p = area in sq. in. corresponding to the required diam. of the balance piston, A_m = the area in sq. in. corresponding to the mean diam. of each blade section, P_d = the press. difference in lbs. per sq. in. across this section, A_a = any annular area in sq. in. where the shaft changes size, P_a = the unbalanced press. in lbs. per sq. in. exerted in this annular area, T = thrust in lbs. taken up on thrust block, P_i = press. in lbs. per sq. in. applied on pressure side of balance piston, P_e = press. in lbs. per sq. in. on the other side of balance piston.

In early designs it was customary to balance each mean diameter of the spindle by its own balance piston, one of which was sometimes placed in the exhaust. Modern turbines generally only contain one balance piston so proportioned as to balance all thrusts at full load on the machine. Any inequalities of thrust at other loads are taken by the thrust block itself.

Figs. 91 and 92 show sections of these balance pistons and dummy rings. These dummy rings are frequently cast separate and bolted to the casing after machining. They carry dummy packing strips of brass or aluminum calked into the rings, as shown. The strips have a wearing shoulder against which the corresponding dummy on the spindle is ground on first assembling and which is afterward maintained at a distance of 0.010 in. to 0.015 in. in operation by means of the thrust adjustment. This clearance varies somewhat with the size of the machine and must be checked at intervals by the operator.

The dummy ring on the spindle body is bored separate and forced in place and locked usually by a shrink ring over a split ring. Clearance of 0.015 in. must be provided at the ends of this dummy ring to allow some freedom for endwise expansion. This will prevent bending of the spindle if the dummies rub.

The **number of rows of dummies** is a matter of choice with the designer; the larger the number the less the steam leakage, but the longer the shaft.

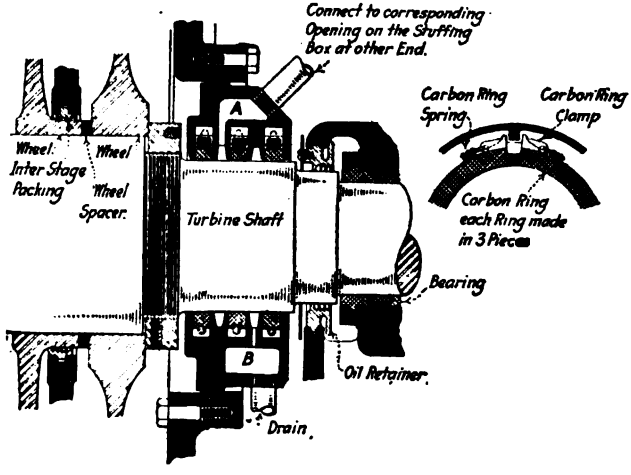


FIG. 93.—DeLaval Carbon Ring Glands.

Casing Glands

Casing glands are made in a variety of forms. Fig. 93 shows one form of gland. It consists of a number of arch-bound carbon rings made in three segments held firmly together by a garter spring. One of the segments is provided with a stop to prevent rotation with the shaft. Segments butt against each other and have a play of 0.006 in. between their inside surface and the shaft. Both glands are provided with a steam outlet between the inner ring and the remaining ones. The steam leakage at the high-pressure

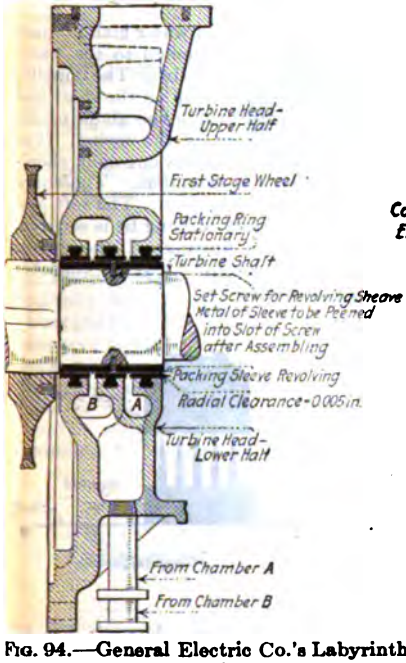


FIG. 94.—General Electric Co.'s Labyrinth Packing.

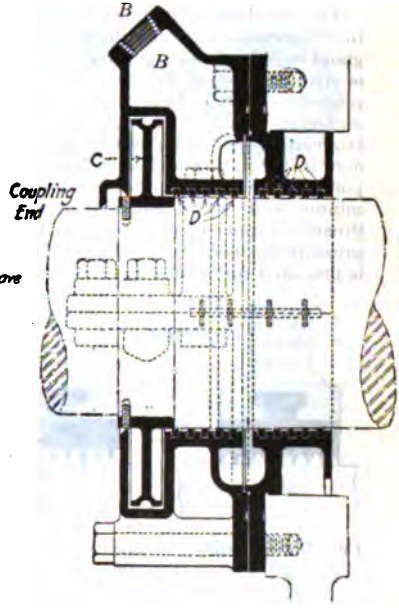


FIG. 95.—Westinghouse Combined Steam Seal and Water Gland.

end is led around and seals the gland on the low-pressure end from air leakage; the amount of steam passed is regulated by a valve. Live steam may also be supplied through a separate valve for astern operation.

Another form of gland is the labyrinth type, shown on Fig. 94. On the high-pressure end it consists of three bronze stationary rings dovetailed and firmly secured to the casing and with axial passages between them connecting to chambers *A* and *B*, which collect the leakage at normal load. The revolving sleeve with the labyrinth grooves is pressed on the shaft and held from turning by two screws securely peaned in place. The packing is supposed to have a clearance of 0.005 in. when assembled. The ends of the labyrinth teeth are knife edge and a running clearance is secured by turning over slowly when first operated. The revolving sleeve is set forward so that 60 per cent of the total axial clearance occurs between the after faces of the revolving teeth and the

forward faces of the stationary teeth. The low-pressure packing is similar to high pressure except that but two stationary rings are employed.

At normal speed the steam from chamber *B* of the high-pressure end is conducted to the single chamber of the low-pressure gland and seals this gland against air leakage. The steam from *A* is carried to one of the low-pressure stages of the turbine. A live steam connection to *B* must be provided for operation astern or at slow speed and in each case the valve from *B* must be closed. This gland requires close attention in operation.

Water glands have been largely used for casing seals on land turbines, but they do not permit starting under vacuum and furthermore become ineffective at less than half speed.

Fig. 95 shows a modified water gland. . It consists of a water gland *C* used in connection with labyrinth packings *D*. Water is supplied to the water gland impeller from a gravity tank under a head of about 8 lbs. The impeller is virtually a small double impeller centrifugal pump shrunk on the shaft and pinned in place. It maintains a solid ring of water inside the gland casing and thus prevents leakage of air into or of steam out of the casing. The labyrinth portion consists of aluminum rings with shoulders engaging shoulders in the casing and with flat springs behind these shoulders. The sharp points of the rings do not at first allow the shoulders to seat. The soft aluminum soon wears down in operation and a perfect running fit is secured. Steam is supplied to *B* to seal the glands as desired in starting. If the normal pressure in operation is greater than atmosphere, a second leak-off chamber is provided connecting to a lower stage of the turbine.

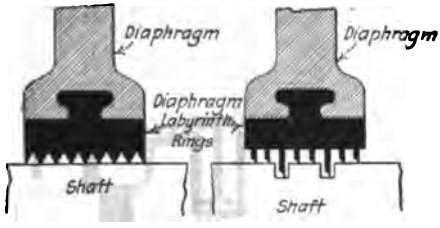


FIG. 96.—Forms of Diaphragm Labyrinth Packing.

FIG. 97.

Diaphragm seals are generally of the labyrinth forms shown in Fig. 96 with a single alloy ring.

The diaphragm ring is made in halves and securely fastened to the diaphragm by dovetails or other means. The knife-edge teeth of soft alloy are in contact with the shaft when first assembled and are allowed to wear to a running fit. The number of teeth is fixed by the designer.

Labyrinth sections are quite generally made of aluminum. It is light weight, does not cut the shaft and wears easily to a fit. Other metals used are soft yellow brasses and occasionally bronze.

All of the forms of labyrinth packing shown above have the teeth cut on the stationary portion and are arranged so that these teeth rub on the shaft. Should such a rub occur, the shaft will be heated locally at the point of contact and it may be seriously distorted. The casing however will be uninjured for the fine edges of the teeth will rapidly wear away and the large surface of the teeth will dissipate the heat without serious results. This possibility of

damage to the rotating shaft has led designers to change the arrangement of the rubbing parts. It is now proposed to make the diaphragm glands as shown in Fig. 97.

The fine teeth are cut very deep in collars placed between the disks on the spindle. The inner bore of the diaphragm is smooth. Should a rub occur with this construction the heating of the diaphragm can cause no material distortion as the heat dissipating surface of the diaphragm increases radially as the heat is conducted away from the bore. The thin teeth on the shaft section dissipate very easily any heat before it can reach the shaft itself. These teeth are generally of bronze or soft brass and wear away quickly if a rub occurs.

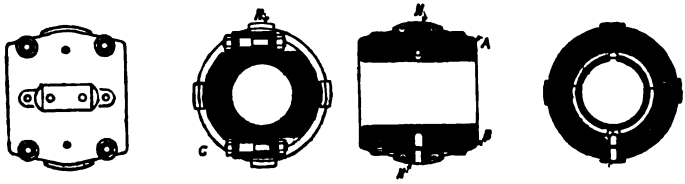


FIG. 98.—Westinghouse Bearings.

Bearings

Main turbine bearings usually consist of two cast iron shells lined with babbitt and provided with spherical self-aligning pads. **Westinghouse bearings** are shown in Fig. 98, where the alignment is secured by steel pads *H* screwed to the cast iron shells *A* and *B*. Sheet-steel liners of varying thickness are placed under these pads. These liners are of assorted sizes and are interchangeable between pads so that the spindle can be adjusted vertically or sideways in increments of five-thousandths of an inch and can thus be placed central with respect to the casing. This adjustment is often necessary with reaction turbines requiring fine clearances at the ends of the blades.

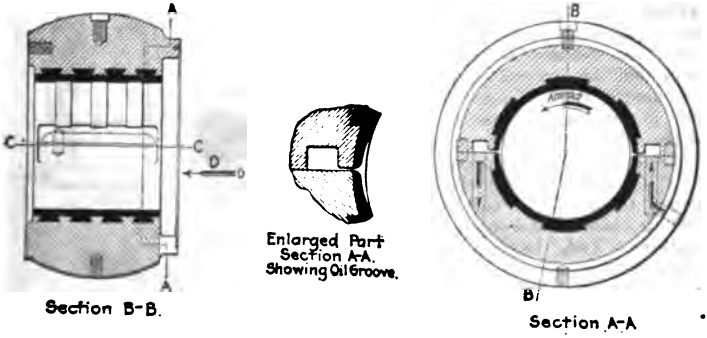


FIG. 99.—General Electric Co.'s Bearings.

The oil enters the bearing from the pedestal through a hole in the lower liners and pad and then passes through copper pipe cast in the babbitt around the bearing and enters a shallow groove at the top. The bearing shells must

be designed strong enough to have no deformation at the ends due to the cantilever action on the supporting pad.

The bearing used by the **General Electric Co.** on turbines is shown in **Fig. 99.** Close clearances at the ends of the blades are not required in the impulse type of turbines and hence while spherical seats are employed, adjusting pads with liners are not necessary. Oil is supplied to this bearing at one side, as shown, and the excess hot oil is wasted on the other side at the opposite joint. Circumferential oil grooves are provided in the top half only to supply the necessary excess of oil.

All bearings are bored and scraped to the shaft before use. They are usually bored about 0.001 in. large per in. diam. of journal. Bearings should be rebabbitted or replaced when the clearance exceeds 0.002 in. per in. diam. of journal.

Lower halves of bearings of all types are scraped to fit the journal for about an arc of 120 deg. at the bottom, the sides for about 30 deg. from the joint are eased off the whole length of the bearing except about ¾ in. at each end. This provides ample clearance for an abundant supply of oil even when the bearing becomes warm. The upper halves are generally eased off in the same way at the joints. Unless such easing-off is done, the babbitt tends to close at the joint as it warms up. This will cut off the oil supply to the bottom half and the bearing will soon overheat.

Gear and pinion bearings are not intended to be self-aligning and hence consist of cylindrical cast iron shells lined with babbitt. **Oil throwers** frequently are turned on the shaft itself just beyond a bearing. Oil guards or **deflectors** are also provided where the shaft passes through the casing of the bearing. These prevent oil working along the shaft and still allow any entrained air to readily escape. Usually such oil guards are made of sheet brass and fit the shaft very closely.

Warm Bearings. If the bearings commence to warm up this may be due to worn out oil or to clogged up oil supply passages; or some foreign matter may have got into the bearings and cut them.

The **Parsons elastic sleeve bearing,** Fig. 100, is still used on some small machines. It consists of an inner cylindrical brass shell surrounded by several concentric brass sleeves with about 0.002 to 0.006 in. clearance between each and the whole fitting into a cylindrical sleeve resting on the pedestal. Oil is forced into the clearance between these sleeves and acts as an elastic cushion, thereby reducing vibration. This bearing is expensive and is not needed on well-balanced machines.

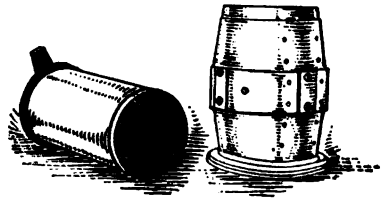


FIG. 100.

Main Bearing Design. Generally the diameter of the bearing is fixed by the design of the spindle. The given speed then settles the peripheral velocity of the shaft. This, however, should not exceed 110 ft. per sec. If W = the total load on a bearing of diam. d and length l, then

$$\frac{W}{ld} = 100 \text{ is a safe limit for bearing pressure.}$$

Higher velocities and pressures have been used but the losses in friction are

greater than with the figures stated. The ratio l/d varies from 1 to 3.5. The usual value is about 2.5. Many formulæ have been proposed for bearing friction but if the speeds and pressures are kept within the limits stated above, this loss may be neglected.

Thrust Bearings. Standard type of adjustable thrust blocks, as shown in Fig. 101 were formerly used on marine turbines. At the present time these have been largely superseded by single thrust collars and thrust blocks as shown in the following figures.

The **energy lost in friction** in thrust bearings can be calculated if the thrust coefficient of friction, speed and diameter of the bearing are known. (See p. 563.)

If the ordinary thrust-collar type of bearing is used or the horseshoe type on the main shaft, the allowable pressure should not exceed 50 to 60 lbs. per sq. in. and collars enough must be provided to take this amount of thrust.

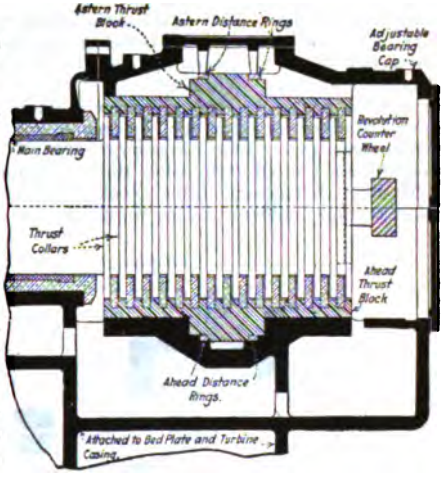


FIG. 101.—Standard Type of Collar Thrust Bearing for Turbines.

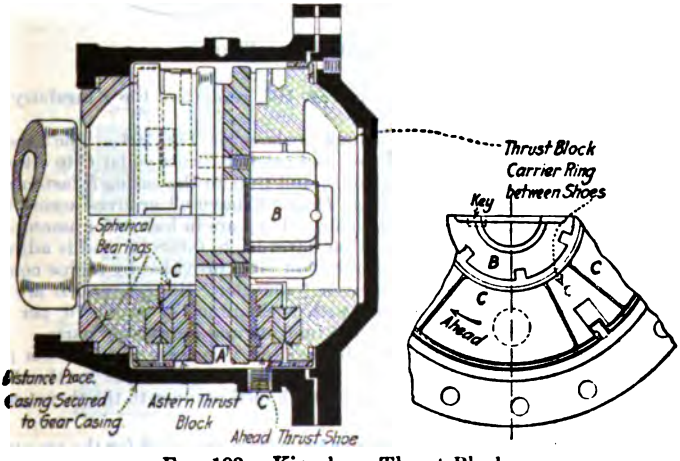


FIG. 102.—Kingsbury Thrust Block.

Fig. 102 represents the **Kingsbury thrust** used by the Westinghouse Co. It consists of a single collar *A* firmly keyed to the shaft and held in place

by a nut *B*. The thrust shoes *C* are in segments. They are babbitt faced and are fitted on the back with a spherically faced, hardened steel button. This button rests on a hardened steel piece inserted in the spherically seated base rings This enables the shoes to adjust themselves independently and thus allows the oil between the shoe and the thrust collar to take the wedge shape necessary for good lubrication, according to the theory of Osborne Reynolds Thrust blocks of this type can be designed to carry 250 to 300 lbs. per sq. in., though one manufacturer designs for only 150 lbs. per sq. in. to allow for dirty oil. In emergency this block can carry very much higher pressures. The Kingsbury thrust is much shorter and of lighter weight than the old style collar thrust bearing.

Modified Kingsbury thrust blocks are furnished under license by several other turbine builders besides Westinghouse.

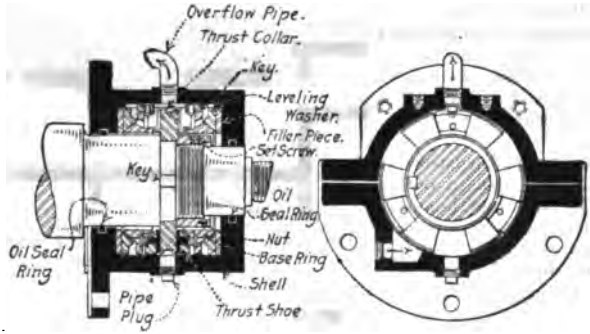


FIG. 103.—Kerr Thrust Bearing.

The **Kerr thrust block**, shown in Fig. 103, resembles the Kingsbury and functions in the same way.

The **General Electric thrust block** is shown in Fig. 104. The pressure is transmitted from the case-hardened and ground thrust collar *C* to either of two babbitt-faced bearing rings *D* and *E* and thus to the casing *F* fastened by bolts to the bearing shell. The rings *D* and *E* have oil grooves across their faces and are scraped to the thrust collar. They are in halves for assembling. The total clearance axially is only from 0.004 in. to 0.008 in. and is adjusted by the shims *G* and *H*. An ample supply of oil is provided by a pipe connection to the thrust at *I*. Emergency or "squealer" rings *K* and *L* are also provided. These should be spaced with a clearance of about 70 per cent of the minimum clearance between the buckets and stationary parts.

In England the Michell thrust is being used very extensively. Its principle is similar to that of the Kingsbury, as provision is made for the thrust segments to adjust themselves in a wedge-shaped manner to the thrust collar in order to lubricate satisfactorily.

Oil for Bearings. Various formulæ have been proposed for the amount of oil required for bearings and thrust. One builder uses the following equation: Gal. per min. of oil = 10 × (clearance area of bearings and thrust in sq. in.). Another states that the oil allowance is one-half of the theoretical flow of oil at 3½ lbs. pressure through the clearance area of the bearings and thrust.

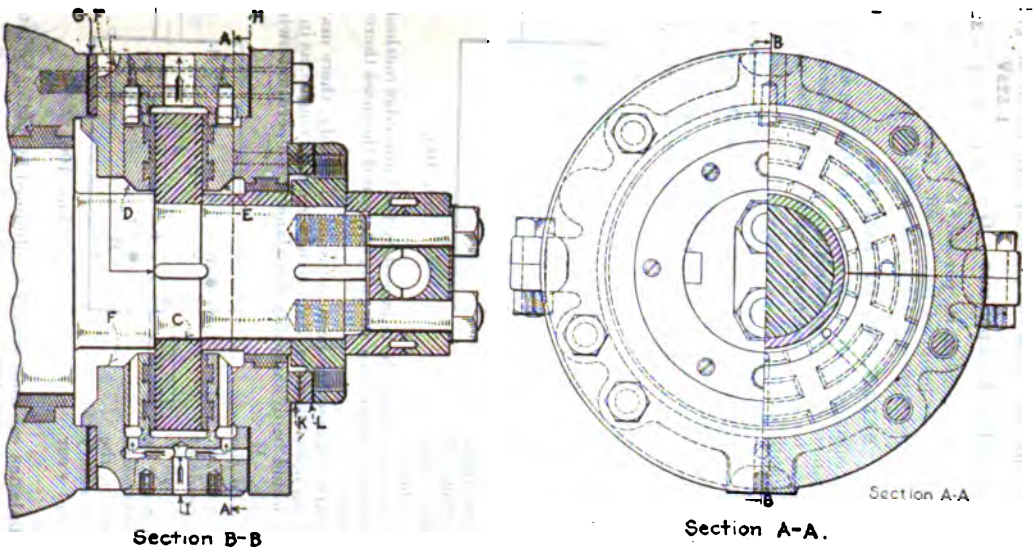


FIG. 104.—General Electric Co.'s Thrust Bearing.

Goudie in "Steam Turbines" gives the following formula for the **gal. of oil per sq. in. of projected surface per min.**: $G = \frac{1.825V}{S(t - t_1)(t - 32)}$, where V = surface velocity of journal in ft. per sec.; S = specific gravity of oil; t = temp. of oil supply; t_1 = temp. of leaving oil. He also states that 0.05 gal. per sq. in. of projected surface per min. has been allowed for lubrication.

BALANCING OF ROTORS

Rotors must be accurately balanced if excessive vibration at high speed is to be avoided. Symmetrical design about the axis of rotation does not guarantee perfect balance, for the metal may contain blow holes and the centre of gravity may be slightly shifted.

Static Balance. A rotating body may be statically balanced by so disposing its mass that the centre of gravity of the whole body practically coincides with the axis about which the mass rotates.

In the case of discs or rings these may be balanced in one of two ways. They may be mounted on accurately ground mandrels and placed horizontally on carefully leveled knife edges. The heavy side tends to go down. Metal, usually in the form of screws, can be added or removed until the disc shows no heavy side. This method can be applied successfully to discs and rings of moderate weight, but is not suitable for heavy masses. Heavy masses cannot be balanced by the knife-edge method because the heavy weight beds down into the knife edges and the mass therefore does not revolve freely.

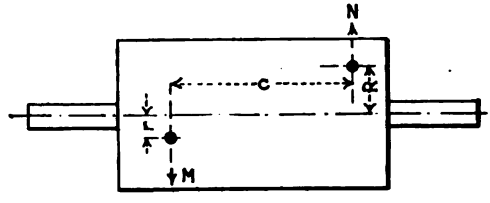


FIG. 105.

Even after static balance has been obtained in a body, there may be two excess weights, on opposite sides of the axis, forming a couple, as in Fig. 105.

Such a body would be dynamically unbalanced in operation owing to the centrifugal couple, M and N not being in the same transverse plane. At the high speeds of turbine rotors such a condition will result in vibrations and excessive bending stresses in the shaft and unnecessary friction, heat and wear in the journals.

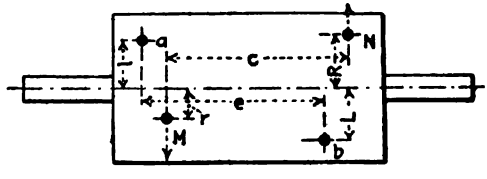


FIG. 106.

Dynamic balance can most readily be obtained by adding counterbalancing couples, as shown in Fig. 106.

Various dynamic balancing machines have been perfected (see *Jour. A.S.M.E.*, Aug., 1916). The Carwen and the Akimoff machines are the ones generally used.

Dynamic unbalance must be reduced to the smallest amount possible. Vibration on the ship is thereby reduced to a minimum.

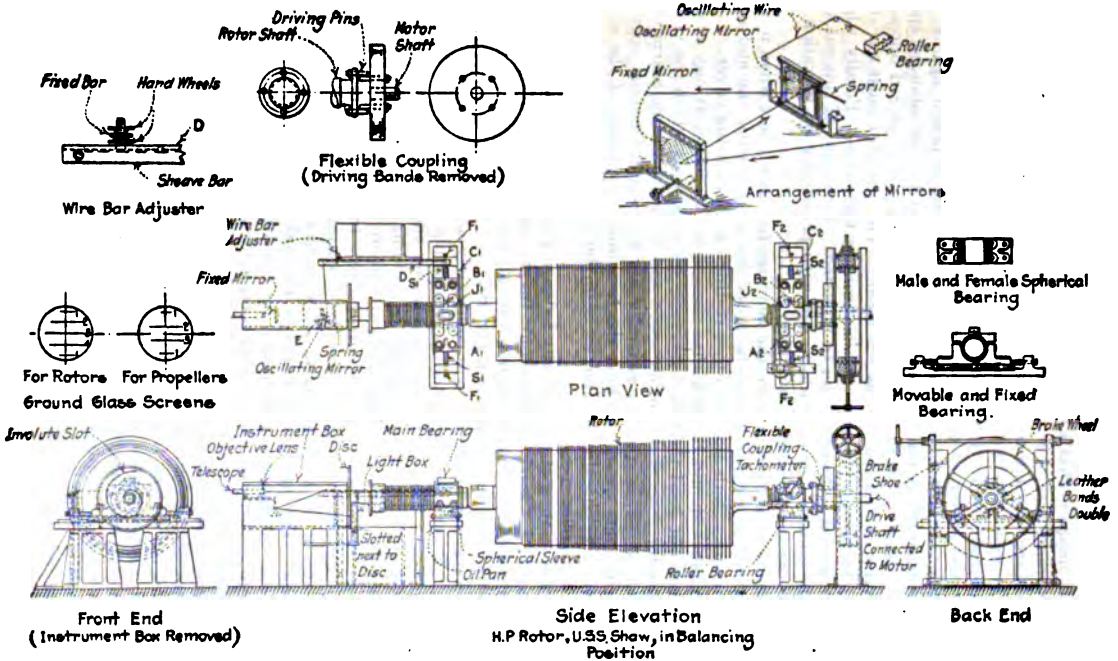


FIG. 107.—Cleary's Equipment for Balancing Large Rotors.

Propellers can be balanced both statically and dynamically in the same way as turbine rotors.

It frequently happens that a turbine rotor becomes unbalanced from alterations and repairs in port or at sea and no balancing machines are available. It is then necessary to balance the rotor in place. This is done by uncoupling the gears or the main shaft and running the spindle up to some chosen speed. It is not important what this speed is, provided the vibration is well in evidence, but the nearer to normal speed, the more satisfactory will be the balancing. The shaft is marked at speed by holding a pencil against it. This mark is generally a number of degrees ahead of the unbalanced spot. One end of the shaft is worked on at a time. A weight is placed opposite the unbalanced spot and the spindle run again. The operator must use skill and judgment in placing weights and in choosing their size. Finally when one end runs smoothly, the other can be adjusted. By means of this cut and try method the spindle can be made to run with practically no vibration, if time enough is taken to balance it carefully.

When a spindle is in good balance the bearings feel quiet, but when not in dynamic balance the vibrations can be noted by a stick with one end held lightly against one's teeth and the other resting on the bearing cover.

Dynamic balance should be such that there is no vibration at any speed. If the balance is not good there will generally be a speed at which the vibration will be greater than at any other speed and at times assume alarmingly great proportions. This speed is generally known as the "critical speed." It is the synchronous speed at which the rotational frequency of the shaft happens to coincide with the free frequency of the supports. Critical speeds are discussed on p. 605.

An interesting method of dynamically balancing large rotors was described by Commander F. J. Cleary, in "Dynamic Balancing" in the Jour. A.S.N.E., Feb., 1918, from which the following notes are abstracted.

The complete equipment is shown in Fig. 107.

The rotor weighing 14,500 lbs. was completely assembled with blades and finish turned. It was then mounted in spherically seated bearings to allow for free oscillation. The pillow blocks carrying these bearings were placed on rollers and were held to the supports by holding down bolts in elongated holes. These bolts were not tight but were merely to prevent the bearing from jumping off the frame. The bearings were free to move laterally but excessive side movement was prevented by springs which were all made exactly alike and of the same strength. The tension on the springs could be easily adjusted. The spindle was driven through a flexible coupling free from any strain. A brake was provided on the driving shaft to slow down the spindle quickly after a test run had been made.

An aluminum disc with an involute slot cut across its face was fastened to the opposite end of the spindle from the coupling. The origin of the involute was the centre of the shaft but the slot was only cut from the hub outward.

The disc was marked by four radial lines 90 deg. apart marked 1, 2, 3 and 4 in line with similar marks on the rotor. A light box with a long vertical slot on the side facing the disc, was placed between the disc and the bladed section of the rotor. This box contained two long-filament lamps and a mirror to throw all possible light on the slot. A wire was connected to the top of the turbine bearing, passed round two pulleys on the adjustable pulley bar *D* and was fastened at the other end to a lever at the back of the oscillating mirror *E*. Tension was maintained in this wire by a spring on the opposite side of the lever. The bar *D* could be adjusted when in use to make the

mirror oscillate about a central position. The range of oscillation of the mirror could be increased or decreased by altering the leverage of the link to which the wire is connected on the back of the oscillating mirror.

The disc shut off from the objective box all light shining through the vertical slot in the light box except at the point where the involute slot crossed the vertical slot. This ray of light was reflected by the fixed mirror onto the oscillating mirror and then into the telescope and was brought to a focus on the ground glass screen fastened to the end of the mirror. The point of light could be moved in a vertical direction on the ground glass screen by adjusting the fixed mirror by means of a thumb screw about its horizontal axis.

To adjust for a run, the rotor was turned over until the quadrant line 1 on the rotor was in a horizontal plane as determined by a trammel. The mirrors were adjusted until the point of light was at the intersection of the vertical line and horizontal line 1 on the ground glass screen. Then horizontal lines were drawn for 2, 3 and 4 and finally 1 again, as shown in Fig. 108.

When the motor was started the beam of light traveled up and down the vertical line on the screen and as the speed increased this becomes a continuous bright line. As soon as vibration started the mirror was oscillated and the beam of light followed a curved path depending on the displacement of the bearing from vibration. The curvature of this path is a measure of the dynamic unbalance of the rotor. The rotor speed was increased to its maximum and the critical speed (which shows the greatest displacement of the line) was noted. The rotor was brought again to this critical speed and the path of the beam was drawn on the ground glass screen, as in Fig. 108.

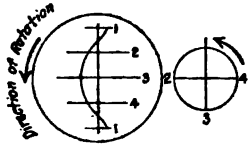


FIG. 108.

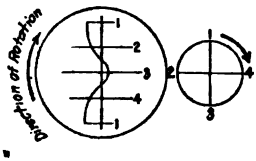


FIG. 109.

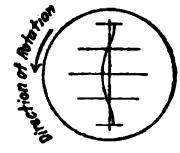


FIG. 110.

This curve indicated maximum displacement along the radial line 3. However in practice it has been found that the heavy spot is always in advance of the maximum displacement on the screen. The rotor was then run in the opposite direction at the same critical speed and Fig. 109 was obtained. This showed that the heavy spot was in advance of radial line 1. A study of the diagrams indicates the probable location at 4. Hence a weight of a few ounces was added near the periphery of the rotor on radial line 1 opposite the heavy point 4 and the spindle was run again at the critical speed and record Fig. 110 taken.

A comparison of this with the previous record indicated whether the dynamic unbalance has been increased or decreased. Similar runs and additions of weights were made until the weights causing the minimum displacement had been added. Then the operators proceeded to add weights to the opposite end of the spindle and to adjust it for the minimum displacement of the curve.

The speed was then changed and tests made to note whether the critical speed has been changed, also to record the displacements at all speeds.

Sometimes perfect balance can be obtained by the addition of a single weight at one or both ends. In other cases several weights are required. This procedure should be continued at both ends of the rotor if perfect balance is desired until the beam again travels in a vertical line at all speeds. Fig. 111 shows progressive indications with this method.

This method of balancing is of such a nature that it can be carried out in any well equipped engine building or repair shop.

Various machines and methods have been devised by Mr. N. W. Akimoff to secure proper dynamic balance. Several of these are described in "*Balancing Machines*" by *F. G. Heckler, Jour. A. S. N. E.*, May, 1919, from which article the following notes are abstracted.

Mr. Heckler states that an **ideal balancing machine should** (1) provide a sensitive and accurate method for determining static balance, (2) provide an efficient method for finding dynamic balance, (3) indicate the magnitude of the unbalanced force or couple, (4) indicate the axial plane of the unbalanced force or couple and (5) indicate the sign or direction of the unbalanced force or couple. He finds that only two machines adequately meet these requirements. These are described below.

The Carwen Machine is built by the *Carlson-Wenstrom Co.*, in nine sizes

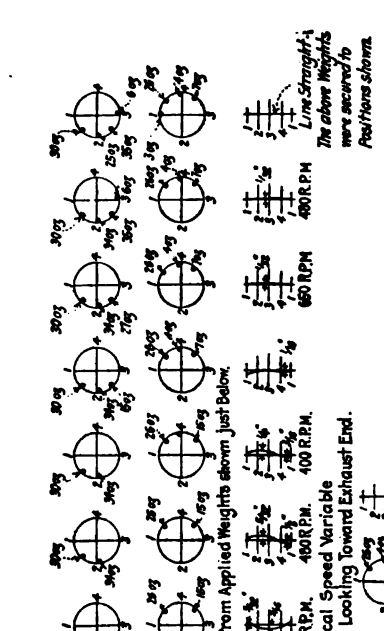

Curves Resulting from Applied Weights shown Just Below.
Critical Speed Variable
Both Ends Looking Toward Exhaust End.
Line Straight
The above Weights were secured in Positions shown.

FIG. 111.—Progressive Steps in Balancing by Cleary's Method.

having capacities from ¾ lb. to 14,000 lbs. Fig. 112 shows one of these machines diagrammatically. Two substantial box pedestals, one at either end of the machine, are rigidly connected by two rods, one on either side. These support a frame similar to a lathe bed. At one end the bed is hinged to the pedestal by a flat plate spring while the other end rests on helical springs. This allows the free end of the bed to vibrate up and down in a vertical plane and at the same time prevents motion in all other directions. A small electric motor with two balancing weights is suspended underneath the vibrating frame. These weights are splined to the supporting shaft so that a pair of forked arms moved by a right and left hand screw geared from a manipulating hand wheel, may move the weights away from or toward each other. When the weights are together they form a symmetrical body in perfect balance. When they are separated a compensating centrifugal couple is introduced.

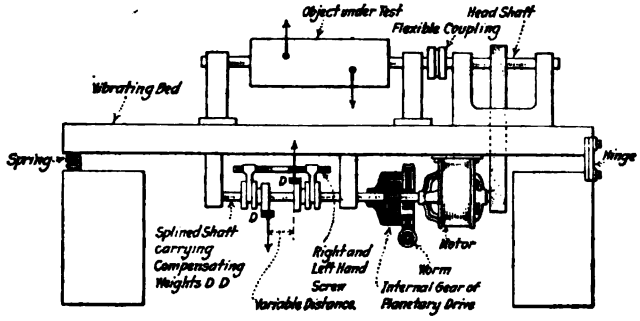


FIG. 112.—Carwen Dynamic Balancing Machine.

The body to be balanced is mounted either on adjustable pedestal bearings attached to the bed or in its own bearings on the bed. It is driven through a flexible coupling at exactly the same speed as the compensating weights below. The shaft on which the compensating weights are placed is driven through an adjustable planetary gear so that their angle may be adjusted to the plane of the disturbing couple without stopping the machine.

Before using this machine for dynamic balance, perfect static balance must first be secured. To obtain dynamic balance, the motor is started and the speed increased until the bed reaches its maximum vibration in synchronism with the springs. The position of the compensating weights is then adjusted until the dynamic unbalance of the body is neutralized.

The amount of this dynamic unbalance in inch-ounce-inches is indicated by a dial on the machine and the plane coincides with that of the compensating weights. Static unbalance cannot be found by this machine.

The combined static and dynamic balancing machine made by the **Vibration Specialty Co.**, consists of a cast iron base supporting a vertical frame by means of a steel-plate hinge. This permits the frame to vibrate horizontally in one direction only. Arms extend on each side of the frame and rest on helical springs whose tension may be varied by hand wheels to increase or decrease the natural period of vibration of the frame. Or the frame may be locked firm by adjusting screws when used for dynamic balancing. The frame carries a motor which drives the object to be balanced through a countershaft and belting.

The frame carries two pedestal supports on which the object to be balanced is mounted on rollers. One support is rigid; the other is supported by a phosphor bronze plate hinge which permits vibration in one direction only. This support has arms and springs just as the frame below so that the natural period of vibration may be altered in finding dynamic balance.

Dial indicators are provided to indicate the amplitude of vibration of both frame and pedestal.

Instead of a compensating couple, a balancing clamp is attached to one end of the object and rotates with it. It can be readily set at any angle and its neutralising effect is secured by sliding a weight in and out on the clamp.

The first step in using this machine is to secure **static balance** by rigidly locking the pedestal support and allowing the whole frame to vibrate freely from the base. The motor is started and the point of maximum vibration is noted. Then the balancing clamp is applied and adjusted until the unbalanced effect is neutralised. The clamp gives the angle and amount of unbalance which may be compensated for by adding or removing metal.

For **dynamic balance,** the frame is locked rigidly and the pedestal is allowed to vibrate on its springs. The motor is started and the synchronous speed of the object and springs determined. The balancing clamp is applied and adjusted until the vibration becomes zero. The unbalanced centrifugal couple is then numerically equal to the weight of the movable weight multiplied by the radial distance it has moved from its neutral position multiplied by the distance from the central plane of the clamp to the centre of the bearing on the left hand rigid pedestal. The position of the clamp shows the plane and the sign of the disturbing couple which may then be neutralised by adding or removing weight at two points in the same axial plane on two sides of the axis.

This machine can thus be used for both static and dynamic balancing. The *Vibration Specialty Co.* has also developed an instrument called a **"Vibroscope"** built on the principle of a seismograph to measure the actual amplitude of vibration in operating machines.

MISCELLANEOUS MECHANICAL EQUIPMENT

Hand Turning Gear. Two general types of hand turning gear are in use. General Electric Co. remove the cap from the end of the bearing casing and raise the trip finger from the emergency governor. A turning wrench, as shown in Fig.113 is inserted in axial slots cut in the collars of the emergency governor. A bracket is bolted on in place of the end cap to support and guide the extension of the turning wrench. The rotors may be revolved by a long-handled ratchet type wrench placed on the end of the turning wrench. The turbine should always be turned in the direction of running ahead. When the work is completed, the turning device should be removed and stored away and the cover replaced and made oil tight to prevent leakage.

The second general type is that used on Kerr turbines and is shown in Fig. 114. It consists of a worm wheel shrunk on the body of the high speed coupling and held in place by four screws. The worm is supported by a bearing at each end and is provided with two ball bearings to take up the thrust. One end of the worm shaft is squared for the ratchet for turning, or a motor may be used if desired, in which case the worm should be kept well lubricated. Extreme care must be taken that this turning gear is disengaged before starting the turbine. Otherwise the main shaft may be sprung. Fig. 114 shows the gear engaged and also shows how the parts are arranged and locked in the disengaged position.

Other marine turbine manufacturers use modifications of one or other of these two methods for hand turning gears.

Lining Up Shaft. The shafting from the gears to the propeller must line up very accurately with the gear shafting or undue strains may be thrown on

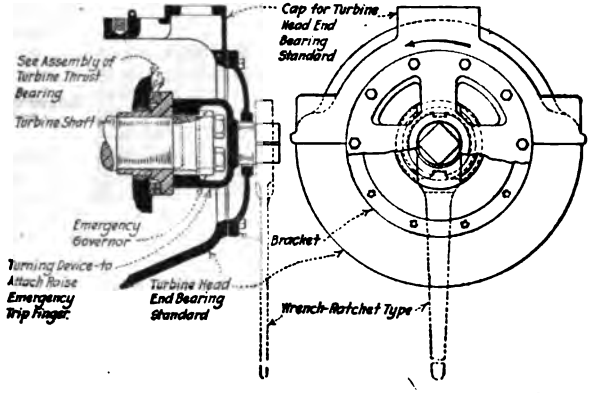

Cap for Turbine Head End Bearing Standard
See Assembly of Turbine Thrust Bearing
Turbine Shaft
Emergency Governor
Turning Device to Attach Raise Emergency Trip Finger
Turbine Head End Bearing Standard
Bracket
Wrench-Ratchet Type

FIG. 113.—General Electric Co. Hand Turning Device.

the gears. This lining up is first done by the usual method of running a line down through the bearings and stern tube and through an outer strut if such is used. If this is well done usually satisfactory results are obtained by squaring up the couplings.

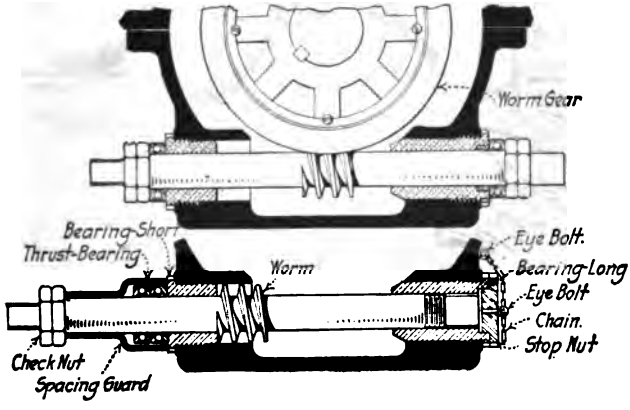


FIG. 114.—Kerr Turning Gear.

In naval practice an optical method of shaft alignment is sometimes used. This is described by Capt. Norris in Jour. A.S.N.E., Feb., 1919. He projects a beam of light from a target located in the strut through the stern tube

to a surveyor's transit located in the shaft alley. In order to permit checking afloat when the stern end may be liable to sagging, a 6-inch watertight tube is fitted between the forward face of the strut hub and the outboard stern tube bearing. The after face of the strut is blanked off.

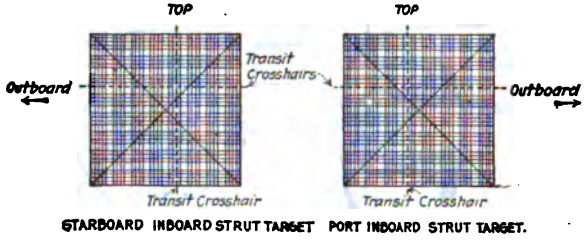


FIG. 115.—Target for Lining-up Shaft.

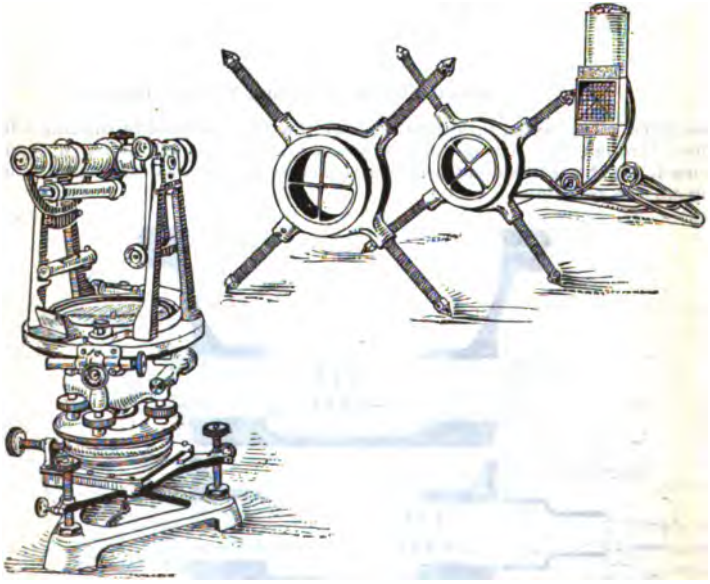


FIG. 116.—Transit, Spiders and Target for Lining-up Shaft.

In order to provide a proper line of sight, spiders with a hole in the centre of each carrying cross hairs are located at the exact centre of bore of both inboard and outboard stern-tube bearings. The transit is located so that its cross hairs fall in line with those of the two spiders, and the line of sight

is projected to the target in the strut. This target consists of a brass frame placed in a 3-inch brass tube in which electric lights are placed. The frame is drilled along the side with small holes through which waxed linen threads are drawn, forming a target as shown in Fig. 115. A white background fully illuminates the target which is placed in the strut so that the intersection of its diagonal lines coincides with the centre of the strut. Fig. 116 shows the transit spiders and target in their relative positions in use.

This method of lining up has proven quite successful and would seem to be suitable for other applications.

Micrometer Gear

Some turbines are furnished with a micrometer gear for checking the position of the spindle inside the casing. This is used more frequently on turbines having long drums than on turbines of the disc type.

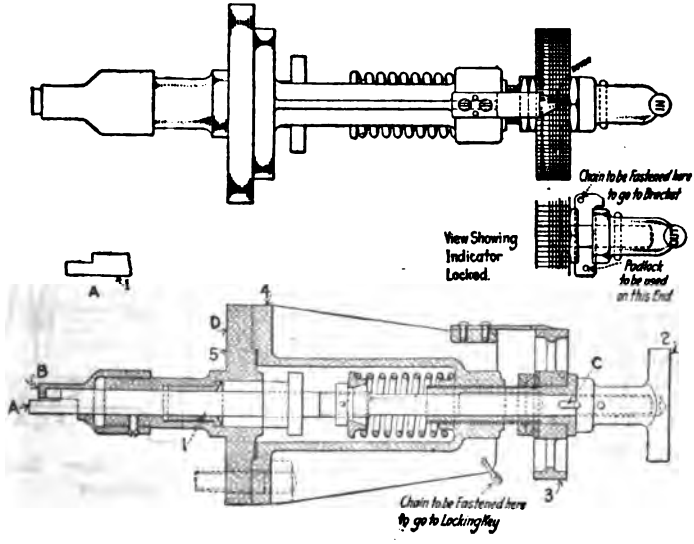


FIG. 117.—Micrometer Clearance Indicator.

This gear generally consists of some form of divided head or dial reading in thousandths of an inch fastened to the outside of the casing. A spindle to which this movable dial is attached passes through a stuffing box and has a hardened point which may be moved up to touch the spindle when it is stationary or turning over very slowly. In emergency it can be used at full speed but a touch is not so easily detected and the point will either heat up and expand the spindle or will wear away the softer drum end. Contact is usually made on the edge of the first or last blade ring or on the edge of the balance piston.

Fig. 117 shows one of these micrometer gears. The spindle (1) has a hardened point at *A* cut away as shown in lower view. A tee shaped handle (2) is fastened on the other end and is held firmly against the dial wheel (3) by

the spring as shown. The two ends of the handle of (2) are marked "IN" and "OUT." When in the "OUT" position the handle is drawn back compressing the spring and the spindle is turned 180 deg. The end *A* of the spindle then rests against the lip *B* of the holder placed on the inner end of the gland. It is locked in this position by inserting a locking key in *C* and attaching a padlock to this key. The outer bonnet (4) is fastened to a flange with stuffing box (5) which is bolted at *D* to the cylinder and forms on its inner end the guide gland for the spindle. The micrometer wheel (3) and its screwed end adjusts the position of the micrometer spindle relative to the fixed casing, as the spring holds the spindle tightly against the micrometer dial wheel. The micrometer spindle thus moves in or out as the micrometer wheel is turned. Adjustments can be made to the thousandth of an inch.

Usually this gear is used to check the position of the spindle just on shutting down after a long run when all parts are at working temperature. The readings of this micrometer gage are more reliable than the indications of the ordinary adjusting thrust screws for the thrust block may be subject to wear.

The Kerr Turbine Co. use a very simple form of indicator, Fig. 118, to show the position of the spindle in their marine turbines and to note wear in the thrust shoes. It is located on the forward end of the shaft beyond the emergency governor. Plate (3) is fastened to the end of the shaft to cover up the lathe centre and also to furnish a hardened surface for the end of the pointer that comes in contact with it when a reading is taken. The shorter arm of the pointer is at all times held away from the plate (3) by means of a flat spring. This prevents wear of the pointer tip.

As the pointer practically touches the centre of the shaft, readings can be taken at full speed. When it is desired to note the position of the spindle, the shorter arm of the pointer is pressed slightly until it touches plate (3). The position of the longer arm of pointer is noted on the scale at its end.

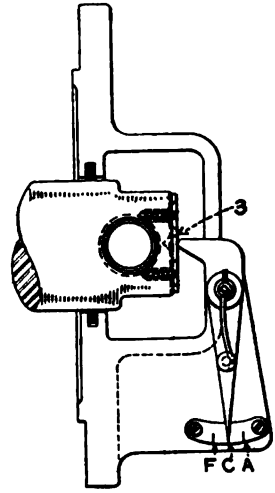


FIG. 118.—Kerr Rotor Indicator

The mark (*C*) on this scale represents the central position of the spindle when running ahead and when thrust shoes are in place. The pointer will be beyond mark (*C*) when running astern by an amount equivalent to the oil clearance in the thrust bearing. Mark (*F*) corresponds to the extreme forward position of the turbine shaft with thrust shoes out. This extreme forward position is reached when the forward end of the spindle strikes the casing. Mark (*A*) corresponds to the extreme aft position of the rotor with thrust shoes and emergency thrust rings out. This extreme aft position is reached when the aft end of the rotor strikes the cylinder body. When the turbine is in operation, the long end of the pointer should at no time approach the danger marks (*F*) or (*A*).

Bridge Gages

Bridge gages are frequently furnished with slow speed direct connected turbines, so that wear of the bearings may be detected in an easy manner.

These gages consist of cast iron yokes which can sit astride the shaft at some exposed point, usually between the gland casing and the bearing casing. The position of the shaft can be determined by a micrometer screw placed in the centre of the yoke and divided, as shown in Fig. 119, to read in thousandths of an inch.

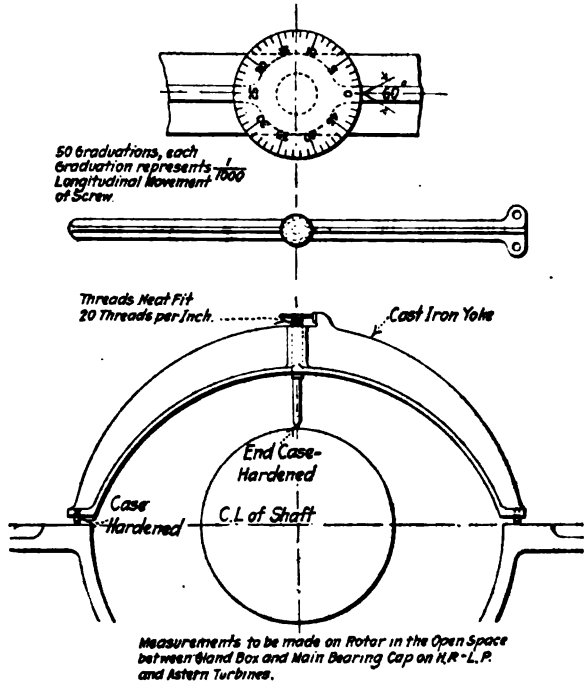


FIG. 119.—Micrometer Bridge Gauge.

Fig. 119 shows a bridge which simply rests on four supporting studs with hardened points. The bridge should preferably be firmly held by bolts to the cylinder body. If dowels are also provided, the bridge is certain to be fitted in the same place each time the bearings are checked.

Fig. 120 shows a second form of bridge gage where a fixed pin is fastened to the cast iron yoke. The position of the shaft is determined by inserting thickness feeler gages between this pin and the shaft. This can only be done where there is sufficient clearance to permit the use of the feeler gage. In cramped locations this micrometer indicator is not suitable. The yokes shown in Fig. 120 can be made to fit over any sized shaft by fastening steel packing blocks to the cylinder faces on which to rest the feet of the bridge. These blocks can be made of the proper thickness to permit the gage to be be used on large shafts.

Probable Mechanical Development. The probable thermodynamic developments, suggested on p. 589, will involve mechanical changes in construction. High pressures and high superheat will necessitate cast steel nozzle blocks and cast steel high pressure cylinders for turbines. Curtis wheels will probably be used in the first stage to reduce the pressure and temperature inside the casing.

Lower vacuum will require greater areas through the low pressure blading, particularly the last row. This may be secured by the higher blade speeds with larger wheels.

The losses due to partial admission in the early stages may be reduced by using smaller wheels in those stages and increasing the wheel diameters as the exhaust end is approached.

Higher blade speeds will involve the development of superior blading material and stronger fastenings. The discs will have to be of extra high grade carbon or alloy steels. Blading material should not only have the necessary increased strength, but should resist corrosion under all conditions of operation.

Considerable gains are still possible by reducing to a minimum a number of small losses throughout the turbine. Blade forms and the areas of passages deserve close study for the stage efficiency can still be increased. Blade forms require study also to determine which will withstand vibration best and how blades can be tied together by shrouds or binding wires which will offer the least frictional resistance to the flow of steam.

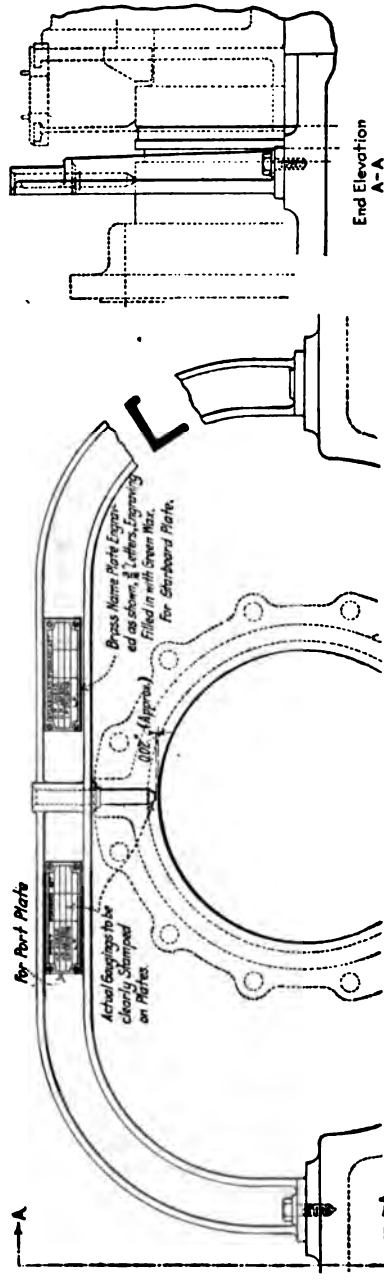

End Elevation
A-A
Bypass Name Plate Engraved as shown, ⅛" Letters, Engraving Filled in with Green Wax. For Starboard Plate.
for Port Plate
A
A

FIG. 120.—Large Bridge Gauge for Use With Feelers.

One manufacturing company has found it advisable, when high efficiencies are desired, to make their orifices of brass throughout the turbine and to provide a special fastening for these orifices in the diaphragms. This would indicate that the ordinary orifice construction, consisting of a thin-wall steel partition, results in measurable losses. The material and design of orifices will therefore merit further study.

Windage and whirling blade losses are of considerable magnitude in the first stages of impulse turbines. It is said that these losses have been noticeably reduced in England by the use of small clearances between diaphragms and discs. No exact data are available in this regard, though the idea appears promising.

Reduction gears have had a large measure of success considering the short time they have been in use. Some troubles have developed, sometimes due to faulty oil, sometimes to poor tooth form, and again to non-rigid supports on shipboard. It is apparent that shipbuilders must provide extra rigid and well braced framing under all turbines, gears and thrust blocks if trouble would be avoided. The high efficiency of the main turbine unit coupled with high gear efficiency makes this installation a very desirable one which will continue in vogue for some years to come.

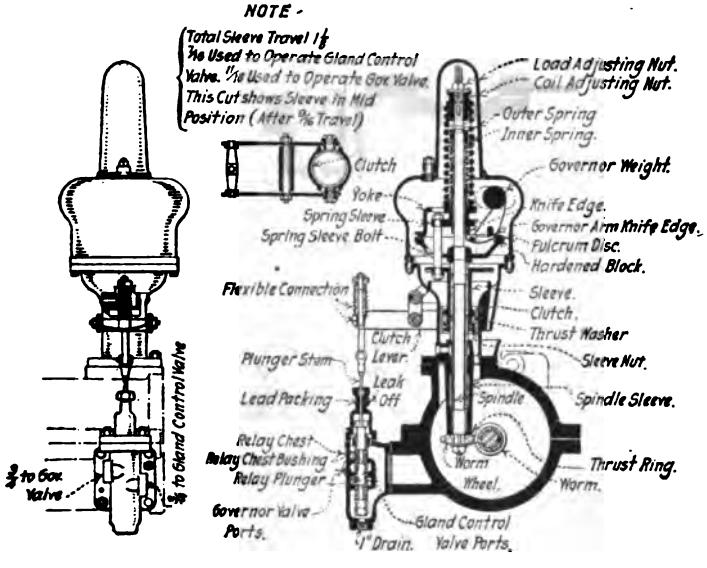


FIG. 121.—Westinghouse Marine Governor.

GOVERNORS AND VALVE GEAR

Most marine turbines are not fitted with any speed control except **safety governors.** No other equipment is needed with **direct-connected turbines** for the propellers are usually so well submerged that they seldom are thrown out of the water. With **geared sets** on the other hand, the propellers are large and in a rough sea may come out of the water and thus permit danger-

ously high speeds on turbine and gears. Speed control would therefore seem necessary. However, many turbine builders, even with geared sets, prefer to trust the safety governor to prevent any dangerous overspeeds rather than to

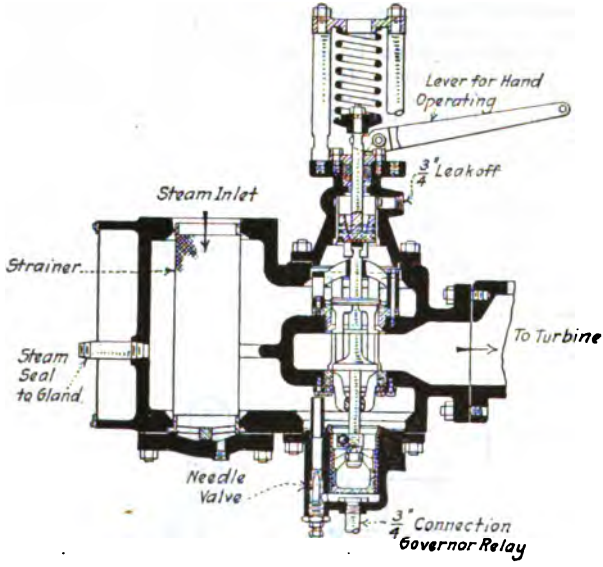


FIG. 122.—Westinghouse Marine Governor Valve.

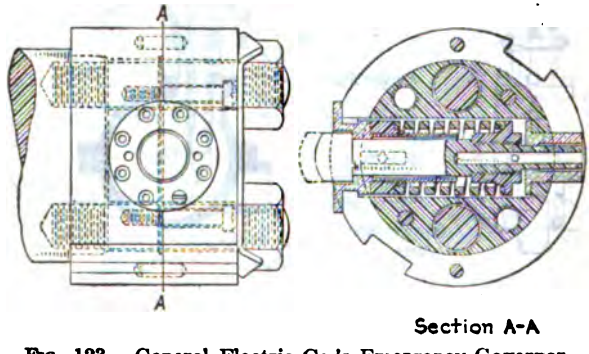


FIG. 123.—General Electric Co.'s Emergency Governor.

install regular speed control governors; but this results in frequent opening and closing of the emergency valve, particularly in a storm.

The Westinghouse Electric & Mfg. Co. furnish regular governing equip-

ment with their turbines to prevent such racing. The governor shown in Fig. 121 is of the ordinary fly-ball type used in land service in which the weights are carried on knife edges and all the working parts are arranged to be under forced lubrication. The governor controls the valve shown in Fig. 122 by means of a steam relay. The connection between governor and valve is merely a pipe and all connecting gear is eliminated. The operation of the relay can be checked on each watch by depressing the small spring above the relay, thus assuring the engineer that it is in operating condition. Cross-compound units are provided with a governor on each spindle, each connected to the relay valve so that the steam flow may be reduced by that side which is running fastest. The governors are usually set so that they come into action at 10 per cent over normal speed.

The governor valve, Fig. 122, is of the poppet type and actuated by two pistons above and below, in conjunction with the spring. Steam pressure is applied to the inner faces of both pistons at all times. The top of the upper piston connects to atmosphere. A needle valve admits steam to the chamber below the lower piston. This chamber is also connected to the governor relay and thus the position of the valve and the amount of steam admitted is under control of the governor at all times. This governing system serves very well at sea and is quite simple.

Turbines driving alternators **in electric drive installations** are all provided with a standard governor which is very sensitive to slight speed changes, as on land turbines. In marine service these governors must also be provided with special devices for varying the speed when in operation so that the speed of the propellers can be changed by altering the speed and frequency of the alternators as may be desired from the bridge.

Overspeed Governors. The form of governor described in the preceding section is virtually an overspeed governor for it only acts when the speed has exceeded normal. Its characteristic feature is that steam is not shut off completely when the governor acts as in the case of other overspeed governors that will now be described.

The emergency governor provided on General Electric turbines is shown in Fig. 123. A collar made in halves is bolted to the end of the shaft and contains a bored hole across its axis between the two halves. The governor mechanism is placed in this hole and consists of a plunger with a head which can pass freely through two guide bushings. The large striking end or head, being heavier than the other end, would fly out from centrifugal force if not counteracted by the helical spring one end of which bears against the nut on the small end of the plunger while the other rests against an adjusting nut on the collar at the head end. This nut is slotted and is prevented from turning by a dowel. The other nut and plunger are held firmly together by a cotter pin. Provision is made for oiling the device in operation. The spring is adjusted so that the centrifugal force of the head overcomes the spring at 10 per cent above normal speed. The plunger moves out and strikes the trip finger of the emergency tripping device (Fig. 124), and allows a flap valve in the steam line to close. A detail of the trip-finger mechanism is shown in Fig. 125. When the trip finger (25) is struck by the overspeed governor bolt, it causes the catch plate to be unsnapped. The spring on the trip rod of the emergency forces this rod down along the inclined surface of the catch plate and thus carries the trip finger out of striking range of the governor bolt. This trip rod releases another catch plate, and springs on the emergency valve rod close this valve.

The emergency valve itself is built up together with the strainer and is

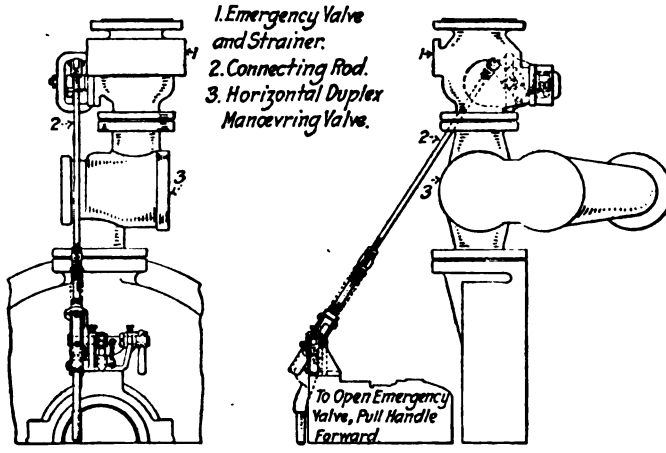


FIG. 124.

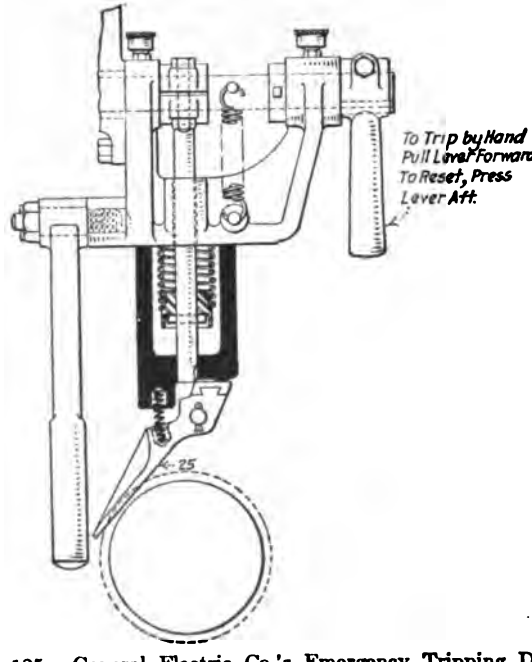


FIG. 125.—General Electric Co.'s Emergency Tripping Device.

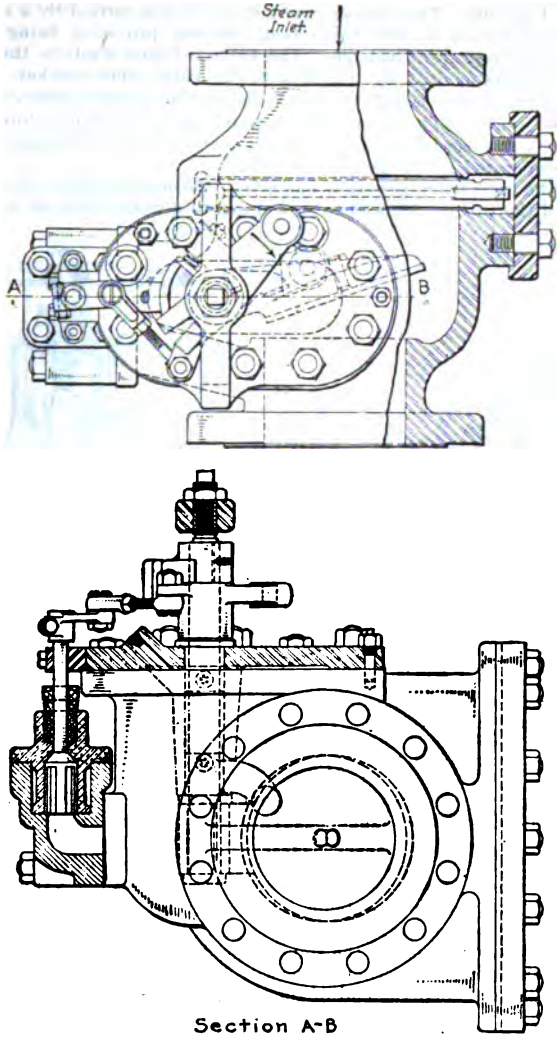


FIG. 126.—General Electric Co.'s Combined Emergency Valve, By-pass Valve and Strainer.

shown in Fig. 126. The clapper emergency valve is carried by a mechanism and bracket bolted to the valve body proper, provision being made by ground joints to prevent leakage. The valve is keyed firmly to the stem and when open rests against an extension of the valve stem bracket. A rotary slotted by-pass valve is provided to equalize the pressure above and below the flapper valve. The maneuvering valve must be shut before the main emergency valve can be opened. This by-pass valve and the main emergency valve are operated by the levers as shown.

When the emergency governor has acted, the maneuvering valve should be closed and the turbine allowed to slow down to normal speed or below before

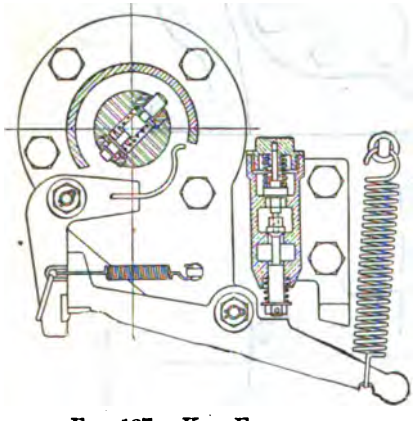


FIG. 127.—Kerr Emergency Governor.

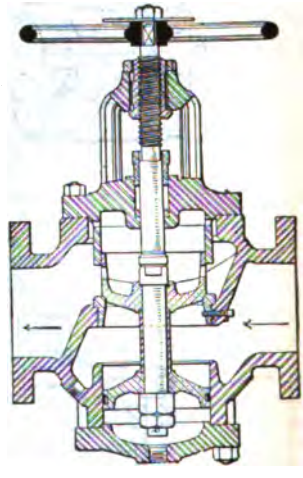


FIG. 128.—Kerr Emergency Stop Valve.

the tripping device is reset. This reduction of speed permits the governor to assume normal position again. Then the maneuvering valves can be opened again.

Kerr turbines use the steam relay overspeed governor, shown in Fig. 127 set to operate at 15 per cent overspeed. The Emergency valve is shown in Fig. 128. The pipe to the steam relay governor connects to the chamber below the bottom piston. The valve spindle is entirely free from the valve. When the valve has closed, the spindle should be screwed down before resetting the emergency governor, otherwise the valve will jump wide open at once and slam against the end of the stem.

Maneuvering Valves

Some types of turbines are furnished with separate maneuvering valves and handles for ahead and for astern usually with some interlocking device. Other types have both valves operated by a single hand wheel.

Westinghouse maneuvering valves are shown in Fig. 129. Both valves are operated by a single hand wheel. Motion in one direction gives steam to the ahead section and in the other to the astern elements. Balanced single disk valves are used as shown with only one seat to be kept tight. All steam must pass through these valves even if one portion of the unit is out of service.

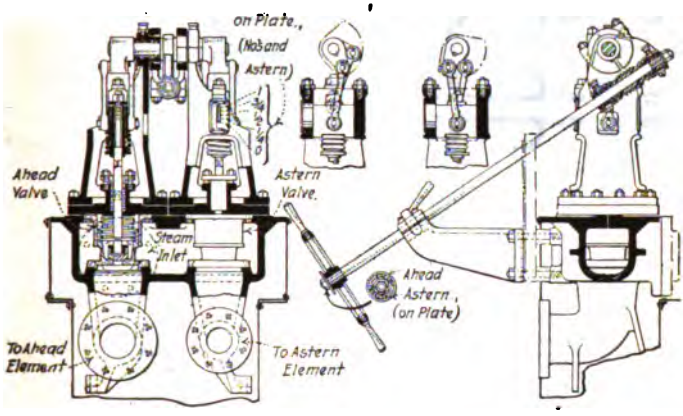


FIG. 129.—Arrangement of Westinghouse Maneuvering Valves.

General Electric maneuvering valves are shown in Fig. 130. The valve discs (1) are single castings with a single seat (2) and mounted on a guided valve stem (3) which forms with the disc a small equalising and warming up valve. On account of an unbalanced area on the valve stems the valve always has a tendency to shut. The valves are opened and closed by the two valve levers as shown, one for ahead, the other for astern, and may be held in any position by the ratchet stops. The two operating levers are interlocked by the devices shown in Fig. 131 so that when either valve is open the other cannot be moved from its closed position.

A globe valve is also placed in the connection to the astern nozzles. This is closed at all times except when maneuvering with the astern turbine. Any leakage to the astern nozzles is thus prevented. Such leakage acts as a brake on the main turbine. The heating of the astern blading from such friction may become serious.

The lift of the by-pass valve is limited to $\frac{1}{16}$ in. and serves only for equalizing and warming up. The valve casings are provided with proper drains to remove condensate.

Kerr maneuvering valves are very similar in principle to General Electric. Two levers are employed which are provided with interlocking devices so that only one can be opened at a time.

Maneuvering valves should be kept steamtight. This may involve regrinding at intervals. Care must then be taken to readjust all lock nuts and levers in suitable operating positions when assembling after regrinding.

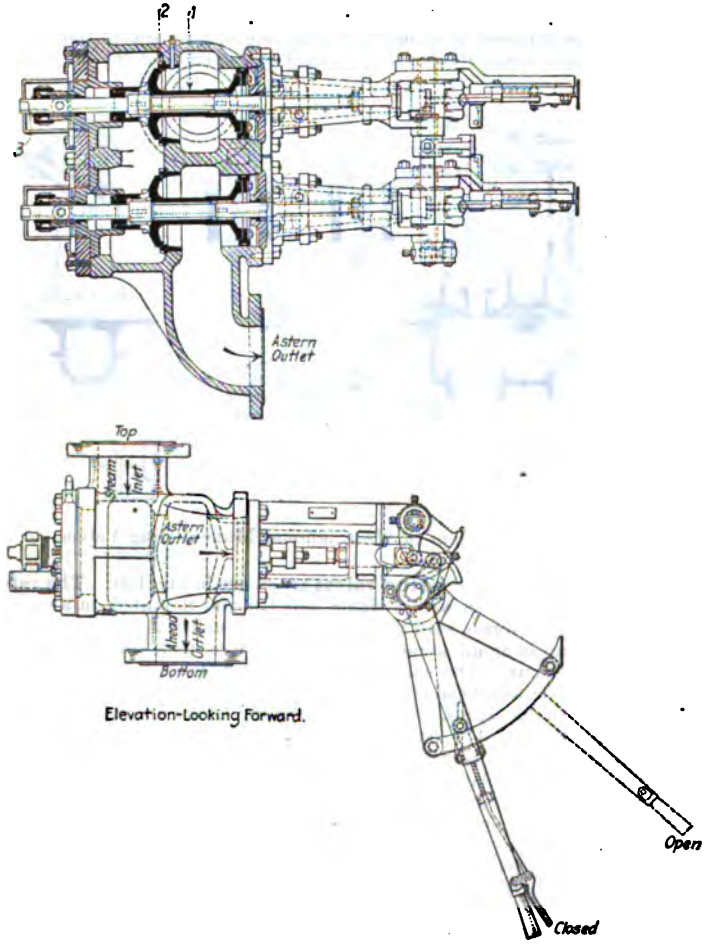


FIG. 130.—General Electric Co.'s Maneuvering Valves.

LUBRICATION SYSTEM

It is absolutely essential in all turbine and gear bearings where shafts revolve at high speed, that a perfect oil film be maintained between the journal and the bearing to insure good operation. This oil film picks up the heat generated by the friction of the journal on the oil and the oil on the bearing. The hot oil must be flushed out rapidly and must be replaced by fresh cool oil from the oil supply system. The lubricating system must therefore provide coolers and storage for oil, pumps to lift the oil from the drain tank to the supply system and also a filter to clear the oil of any impurities such as dust, dirt, casting sand, pipe scale, and metal particles.

The greatest care must be taken to keep the oil clean for dirty oil not only increases friction losses, but increases the liability of damage to gears and bearings.

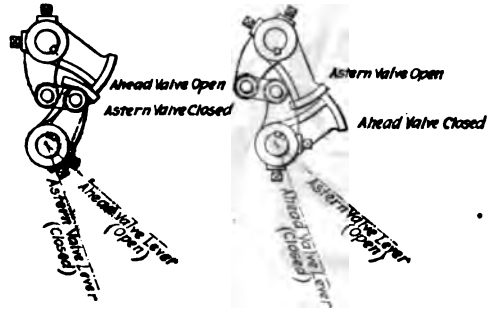


FIG. 131.—General Electric Co.'s Interlocking Gear.

Bearings

The film of oil under a high-speed journal must be of such body and viscosity that it absolutely prevents any metal to metal contact between the shaft and bearing. The allowable pressure per sq. in. must not be so high as to break down the viscosity of the oil. No grooves of any kind must be cut in the lower halves of bearings. This is contrary to former accepted engine practice and marine mechanics must be watched in fitting bearings to see that they do not try to cut grooves on their own responsibility.

For effective lubrication oil must be **admitted to bearings at a point** where there is the least pressure, and generally with turbines this is at the top or sides of the bearing.

The **rapid removal of the hot used oil** is a necessity for good lubrication. Hence a great quantity of oil is furnished to each bearing, of which a considerable amount merely serves in many designs of bearings to flush out the hot used oil. This is effected by cutting one or more open ended grooves clear down the whole length of the upper half of the bearing and supplying these grooves liberally with fresh oil. The Allis-Chalmers bearing, shown in Fig. 132 has several such flushing grooves.

This flushing out of hot oil prolongs its life and provides excellent lubrication.

If bearings commence to warm up, this heating may be due to impure or

emulsified oil or to clogged up oil supply passages. Or again, some foreign matter may have got into the bearing and cut it up.

Some oils emulsify if water is allowed to churn up with the oil. This destroys the lubricating value of the oil. The water can usually be separated out by heating the oil to 180 deg. F. and allowing it to stand at this temperature.

Sight oil vents or oil flow indicators should be provided above each bearing, as shown in Fig. 132. These are connected to the pressure oil supply and serve as air vents as well as to show the operating engineer that the oil pressure is being maintained on the bearing. The oil from such vents usually returns direct to the drain tanks.

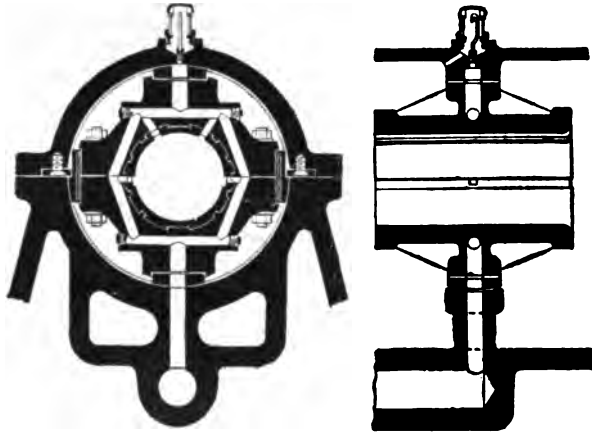


FIG. 132.—Allis-Chalmers Standard Bearing.

Lubricating Oils

Lubricating oils are fully discussed in Section 13 of this book. All turbine oils must be pure mineral oil free from animal and vegetable matter and sulphur. The following oils have been recommended for marine turbines and gears in Technical Order No. 75, of the Emergency Fleet Corporation.

Name of oil	Viscosity at 100 deg. F.	Maker's name
D.T.E., Heavy	271	Vacuum Oil Co.
Alcaid	322	The Texas Co.
Ursa	750	
Calol	275	Standard Oil Co. of California.
Atlantic Turbine Oil, Heavy	280	Atlantic Refining Co.

The heavy Ursa oil is recommended for Falk gears which require a high viscosity oil while the turbine bearings require a low viscosity oil. It is therefore recommended when Ursa oil is used that cooled oil be sent to the gears while the oil to the turbine bearings be by-passed around the oil cooler.

Oil should be supplied to the bearings at **temperatures of** 110 deg. to 130 deg. F. and should leave at about 20 deg. above this temperature.

The usual **oil pressure** at the bearings is about 15 lbs. per sq. in. Some turbines are furnished with **gages at the bearings** to indicate this pressure.

Oil becomes darker the longer it is used but this does not always indicate deterioration of the oil provided it is clear. If however it is cloudy, it probably contains much harmful water and should be heated in a storage tank and allowed to settle.

Oil should be **filtered** as much as possible to remove all fine foreign particles which may be carried in suspension and which would cut the gears.

Types of Lubricating Systems

One of the most important parts of the whole marine turbine equipment is the lubricating system. If it fails for any reason, the whole machinery must be stopped. This system must therefore be reliable, easily handled and as nearly as possible "fool-proof." It must be provided with every efficient device to give warning of trouble such as low oil level and closed strainer alarms.

The oil must be of high grade and must be maintained in as clean and effective condition as possible. This involves strainers, filters and coolers. Strainers may be placed in suction and discharge lines at the oil pumps. So much oil is circulated that all cannot pass through the filters, so fractional filtration or batch filters must be used. (See p. 671.)

The simplest lubricating system is the "**pressure**" **system** such as is used on all land turbines. The oil is delivered under pressure to the bearings by a pump geared to the main turbine shaft. An auxiliary steam driven pump is in reserve for starting up and for emergency use. A relief valve discharging into the oil supply tank maintains the desired pressure on the system. Strainers are provided on the suction or discharge side of the pump to catch dirt. Batches of oil are drawn off the system from time to time and clean oil added.

This system is **modified on shipboard** and reserve tanks above the turbines are furnished for gravity supply. Some systems are pure **gravity systems**; other systems use the **gravity tanks as reserves** only.

Gravity System. Technical Order No. 75 of the Emergency Fleet Corporation recommends the system shown in **Fig.** 133 for geared units up to 3000 S.H.P.

The oil is drawn from the drain tank through a coarse mesh twin strainer to the oil pumps. It is discharged through a fine mesh twin strainer and through the oil cooler to the gravity tanks, each of 600 gal. capacity, located 30 ft. above the centre line of the bearings. The supply pipe to the bearings is taken from 8 in. above the bottom of the gravity tanks, as shown, so as not to disturb water and dirt that has settled. The oil drains back to the drain tank from the bearings and gear casing.

The bottoms of the settling tanks are piped to an oil filter. The filtered oil also returns to the drain tank by means of the overflow line from the gravity tanks.

Three reserve tanks, each of 600 gal. capacity, are furnished for storage of oil and proper piping is provided for them.

In this system the gravity tanks are used alternately as settling tanks. In normal operation the tank in use should be continually overflowing to the drain tank.

The coolers are in duplicate. Piping to the coolers is furnished with necessary by-pass valves, gages, thermometers, etc. The oil velocities in the pip-

ing must not exceed 80 ft. per min. in the drain pipe system from the bearings, 125 ft. per min. in the oil pump suction and 250 ft. per min. in the discharge lines.

Oil flow indicators are placed in the oil pump discharge in the overflow line from the gravity tank to the drain tank and in the main supply line from the gravity tank to the turbine and gear unit.

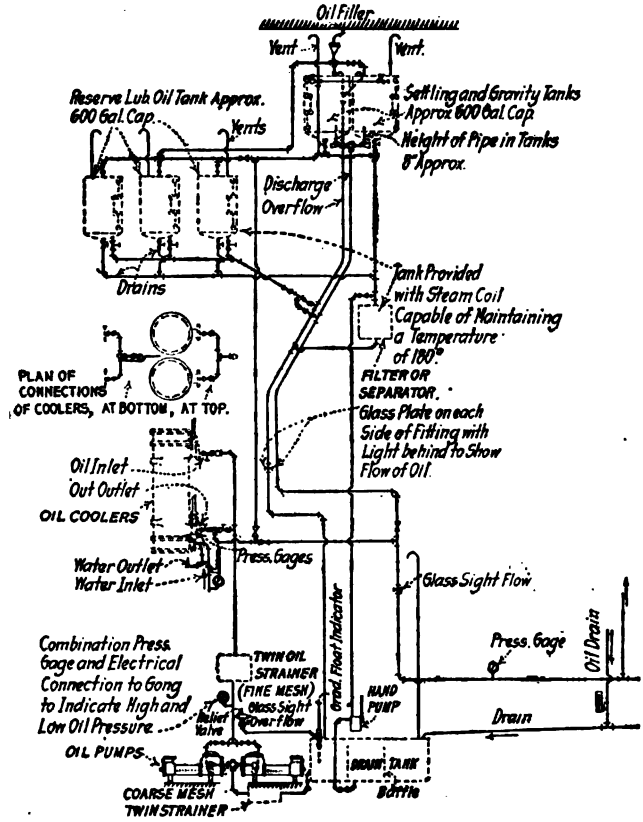


FIG. 133.—Oiling System Recommended by Emergency Fleet Corporation.

Thermometers are to be located at all bearings, oil cooler inlet and outlet and at other necessary points.

The gravity tanks have heating coils so that the oil may be heated to 180 deg. F. when in port so that all water and impurities will settle out. Each tank has gage glasses in series to indicate the oil level.

Float controlled electric alarms are furnished in each gravity tank to in-

dicate low level. An electric alarm pressure gage is also supplied between the oil pumps and the fine mesh twin strainer to indicate high pressure due to dirty strainer or racing pump or low pressure due to clogged suction strainer or too slow pumping.

Pressure-gravity Combination System. The system shown in Fig. 134, used on the Hog Island boats, is a combination of pressure and gravity systems. The used oil from bearings and gear case return to a drain tank

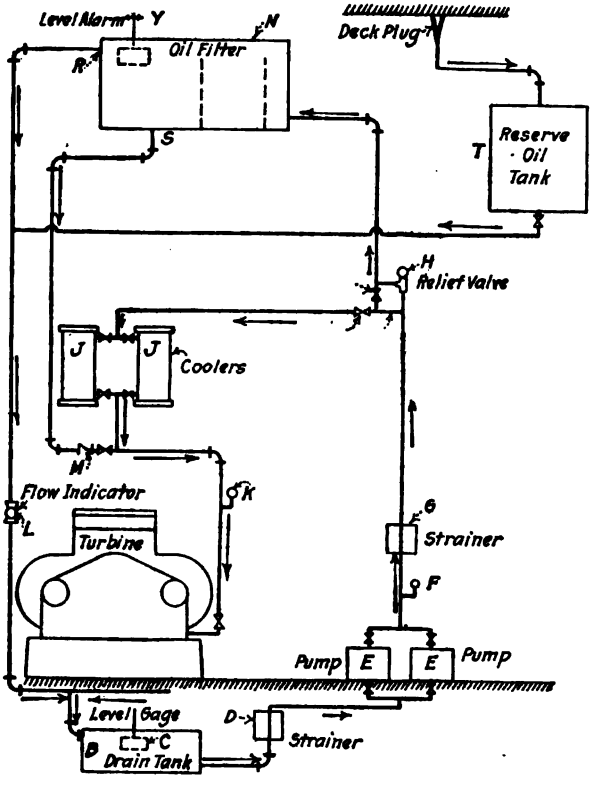


FIG. 134.—Lubricating System on Hog Island 7500 Ton Ships.

B. Either of two oil pumps *E*, *E* suck this oil from the drain tanks through a twin strainer *D* with ⅛-in. mesh to catch waste and coarse particles. The pumps discharge through a fine-mesh strainer *G* into the oil coolers *J*, *J* which are in duplicate and thence back to the oil supply line to the bearings. A pressure relief valve *H* set at the desired bearing pressure is placed on the oil line just before entering the oil coolers. The pumps are run at such a speed that more oil is pumped than required by the bearings. The excess

hot oil passes through this relief valve and up to the oil filter tank *N*. The filters only handle a portion of the oil and the remainder overflows into the storage tank. A line *S* leads from this tank to the oil supply line to the bearings. A horizontal swing check valve *M* connects the two lines. As long as the oil pump supplies an excess of oil at greater than bearing pressure this check valve remains closed. But if the pressure on the bearings fails for any reason such as slowed-down pump or clogged strainers, the swing check automatically opens and oil flows from the gravity tank. The normal overflow from the storage tank returns to the drain tank through pipe *R* with an illuminated sight flow indicator *L*. This device consists of two sight glasses on opposite sides of a fitting so that flow of oil may be noted.

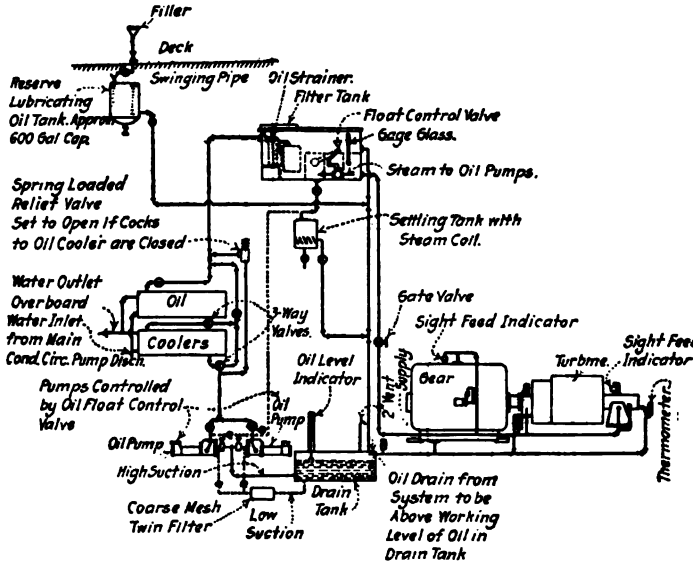


FIG. 135.—Westinghouse Oiling System.

An electric low level alarm *Y* is provided on the storage tank. A level indicator *C* is also placed on the drain tank.

A steam coil is placed in the storage tank *N* to heat the oil if necessary. Drains on the storage tank enable any water and other settled material to be drawn off. Reserve oil is held in tank *T*, which is filled through the deck plug. Oil pressures are shown by gages *F* and *K*.

The oil pumps are hand controlled for speed.

Fig. 135 shows another modified lubricating system recommended by Francis Hodgkinson in his paper "Progress in Turbine Ship Propulsion," Society of Naval Architects and Marine Engineers, Nov. 14, 1918.

The Westinghouse Company generally provides oil pumps directly driven by gearing connected to the main gears. With the double reduction gears

two pumps are generally furnished, each capable of delivering all the necessary oil with steam-driven pumps as relays.

The fine-mesh twin oil strainers of the two preceding systems are replaced by an easily cleaned strainer section in the gravity supply tank. The drain tank is furnished with a high suction for ordinary service and a low suction with a coarse-mesh twin strainer to draw off all dirt and muck from the bottom of the drain tank and discharge it to the oil filter. A settling tank, in which to separate out water, is placed so as to draw oil off from the bottom of the filter tank. This settling tank has a steam heating coil.

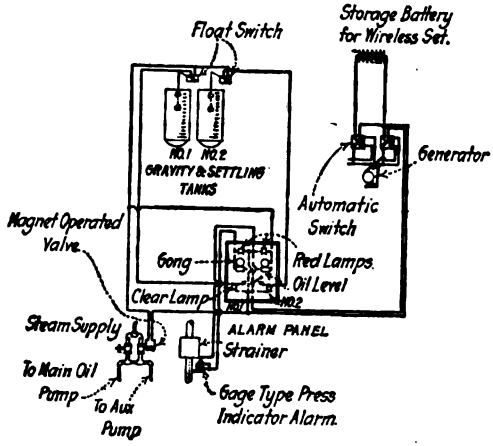


FIG. 136.—Automatic Electric Alarm and Control System.

Pressure Systems are used on certain marine installations, such as destroyers, where it is impossible to install overhead lubricating oil tanks and a gravity system. In this case reliance must be placed on fine-mesh twin oil strainers. These should be by-passed by a spring loaded check valve that opens at a very slight pressure above the normal pressure on the system. This prevents the system from being choked or the strainers from bursting and ensures a supply of oil even if the strainers become clogged. An alarm system should be installed to indicate when such a by-pass valve is working, so that the engineer can give attention to the strainer at once. One of these systems is shown in Fig. 136.

Oil Pumps

Oil pumps may be direct driven from the main turbine or may be independent. There are several styles of **direct-driven pumps.** Fig. 137 shows the **gear type** used by Westinghouse. The gear teeth carry the oil from the suction to the discharge pipe, as indicated by the arrows. The only leakage that occurs is the slip by the ends of the teeth and over the top of the gears. The latter clearance should be kept as small as possible by providing a tightly fitting cover.

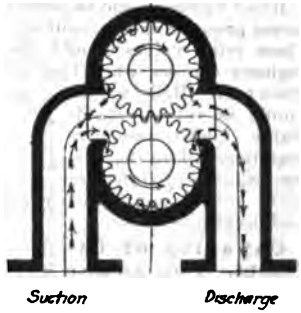


FIG. 137.—Gear Type of Oil Pump.

One of these gears is driven from the gear shaft of the first reduction gear

by spiral gearing. Check valves are provided on both suction and discharge oil pipes, so that the pump will function whether the turbine is running forward or backward. An auxiliary oil pump independently driven must also be provided so that oil can be circulated and cleaned when in port.

Independent oil pumps are either of the direct-acting duplex type or of the vertical simplex marine type, and must be in duplicate. The vertical type is preferable, for the oil cylinder can be placed lower down and if possible below drain tank level so that air pockets cannot form in the suction piping. The oil ends of these pumps should be fitted with metallic packing and metallic valves. No soft or composition packing should be allowed, as particles may get in the oil and cause bearings to overheat by stopping up passages. The piston on the oil end should be a standard steam piston fitted with snap rings preferably of bronze. These pumps must be of liberal capacity so that sufficient oil can be supplied without excessive speed even when a valve is not seating properly. Long stroke pumps are preferable to short stroke units, as they have longer life, do not churn up the oil so much and give less trouble. Excessive speed indicates trouble in the pump or on the system.

All these forms of pumps are notorious steam hogs. Small steam turbine driven centrifugal pumps for handling oil have been used for land service and there seems no reason why they will not operate satisfactorily on board ship. They have been built into the drain tank with the turbine element above the tank top on a vertical shaft and the centrifugal pump submerged in the oil. They cannot become air bound so long as there is a plentiful supply of oil in the tank and this can be assured by an electric low-level alarm. This unit is lighter, more compact and fully as reliable as the reciprocating unit. Turbine-driven rotary pumps are also used.

In some cases provision is made to automatically regulate the speed of the oil pump. Direct-driven pumps of course vary in speed with the speed of the main unit.

The Westinghouse system provides for automatic regulation of the auxiliary oil pump. DeLaval provides similar automatic regulation. The Hog Island system provides only hand regulation but the watch engineer must always see that oil is overflowing from the filter through the illuminated sight flow indicator.

Relief valves should be placed on the discharge side of each pump so that no excess pressure can be built up should valves be closed on the discharge line. These relief valves should discharge into open funnels so that the watch engineer can see when they are in operation. The overflows are piped to drain tank. Relief valves should be set at considerably higher pressures than required on the system, so as not to interfere with other apparatus, such as strainers, etc.

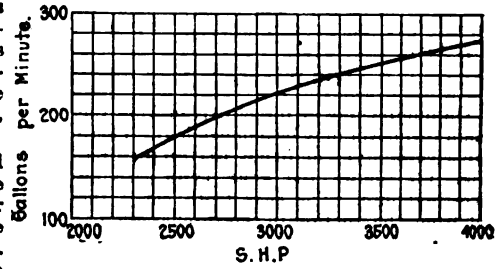


FIG. 138.—Oil Requirements of Geared Turbine Sets.

Capacity of Oil Pumps. Formulæ have already been given for the oil required by the turbine bearings alone. Gears, however, require great quantities of oil. The amount actually supplied depends on the individual ideas of the manufacturer and no definite rules can be stated. Fig. 138 is plotted from data supplied by one large builder and calls for very liberal quantities of oil.

Technical Order No. 75 of the Emergency Fleet Corporation states that duplex lubricating oil pumps 7½ in. × 7 in. × 10 in. built for 150 lbs. steam pressure shall be used on geared units up to 3000 H.P. The suction head must not exceed 2 ft. These pumps are to have brass valves, cast iron plungers and steel pump rods. The pistons and plungers are to have cast iron snap rings. Metallic packing must be used throughout on all rods. Paper packing is to be used on all joints. Vertical type of pump is preferred to the horizontal.

Strainers

The strainer shown on the Westinghouse system, Fig. 139, consists of three readily removable screens placed in a strainer box. The first is of per-

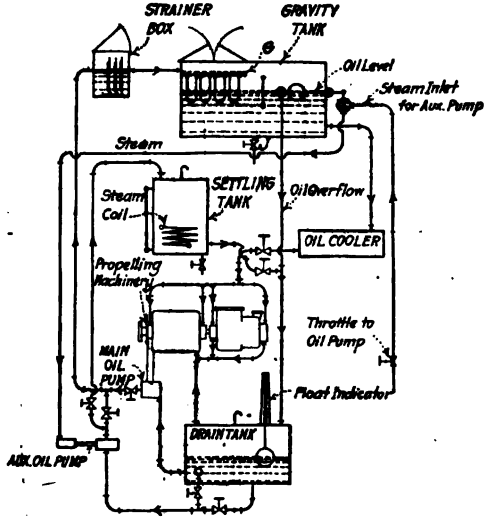


FIG. 139.—Westinghouse Oiling System.

forated sheet metal with 1/16-in holes, the second 35-mesh wire gauze and the last 70-mesh wire gauze. The frames carrying the screens are provided at their bottoms with lips to catch any foreign matter stopped by the screens and dropping down. If these screens become choked there is no interruption of service for the oil simply flows over the top of all of them. Hence they need frequent watching.

The system shown in Fig. 140 provides twin suction strainers with about 1/8-in. perforations. These need very frequent cleaning when starting up. DeLaval places a strainer basket over the suction intake in the drain tank. These pump suctions must have a reliable vacuum gage in sight of the watch engineer or otherwise the pumps may loose their oil due to choking up of the suction strainers.

Fig. 141 also shows a second strainer on the discharge side of the pump. This is a twin strainer with 32-mesh screens of brass wire. A differential

gage, or better still a U-tube preferably with an alarm, should be provided to note the choking up of these strainers. The ratio of the straining area to the inlet area in these strainers should be at least 6 to 1.

Duplex strainers must be provided with bevel gear controls or other lever devices so that both inlet and outlet valves to the strainer can be moved

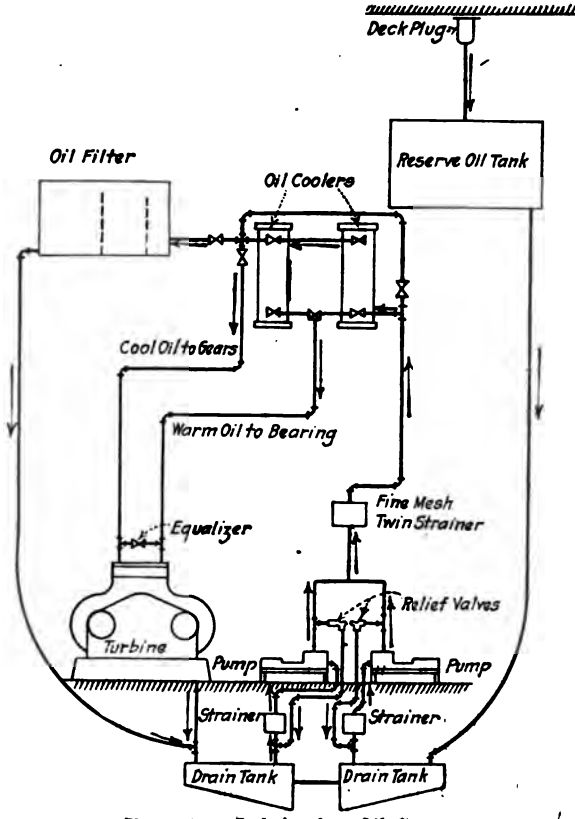


FIG. 140.—Lubricating Oil System.

simultaneously without interrupting flow. DeLaval places a duplex strainer between the oil storage tanks and the turbine so that any foreign material that may accidentally get into the storage tank can be caught before entering the turbines.

When new installations are first started up, an astonishingly large amount of dirt collects in the strainers, hence these require constant cleaning for some time; after the system is clean, once a watch is usually sufficient.

Strainers should be of such a size that the friction loss through them does not exceed a 2-ft. head.

Sometimes strainers are by-passed the by-pass being provided with a relief valve so adjusted that it opens when a fixed differential pressure occurs across the strainer. In this way the oil flow is uninterrupted, but an alarm must be provided to indicate when this valve opens or otherwise the clogged strainer will go unnoticed.

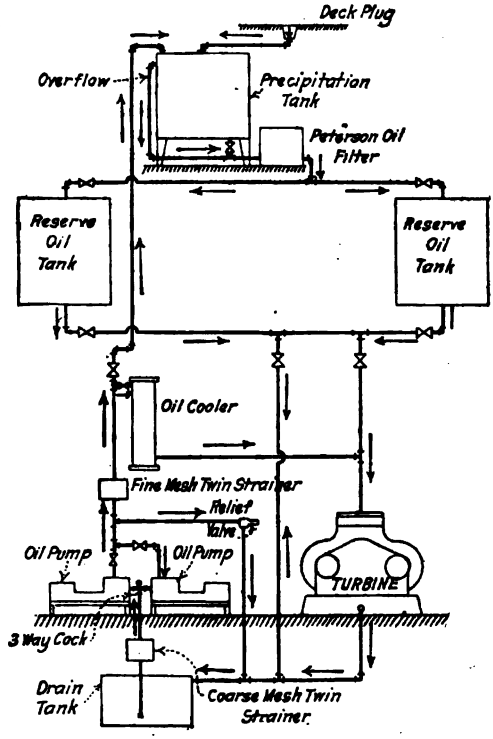


FIG. 141.—Oiling System for Ships of Ore Steamship Corporation.

Oil Filters

Oils used in turbines and gears become dirty from the formation of oxidation products, from scale off pipes and castings, from wear and tear of gears and bearings, and from water which has found its way past guards into oil chambers.

Purpose. Filters are provided to remove these impurities from the lubricating system. Water and muck can generally be separated from oil by heating the oil to about 180 deg. F. and allowing it to settle for several hours. Ample precipitating area must be provided to allow this settlement to take

place. The water and dirt go to the bottom and provision must be made to drain these off. Dirt and other material can be removed either by filters with cloth filtering mediums or by centrifugal separators. On account of the quantity of oil circulated in a turbine and gear, only a portion can be filtered at a time. Hence the fractional filtration and the batch filtration systems have been devised.

In the fractional system all the oil circulated passes through the filter tank. Only a portion of this passes through the filtering medium, the remainder overflowing back into the system again.

The percentage filtered depends on size of filter and storage tank provided, but usually ranges from 5 to 10 per cent. The oil that does not pass through the filters simply flows over a weir or baffle into the storage tank, as shown in the various figures. Considerable care must be taken in designing these filter tanks so that the oil does not wash sediment off the filter bags and into the storage tank when the ship is rolling heavily. A consideration of this feature may lead to considerable changes in the design of filters.

Westinghouse provides a fractional filter as shown at *G* in Fig. 139. A cloth filter bag is carried on a frame and is protected by a perforated sheet steel guard. The filter bag is held in place by a pipe guard and cannot be removed until same is above the surface of the oil. Bags should be removed periodically and cleaned.

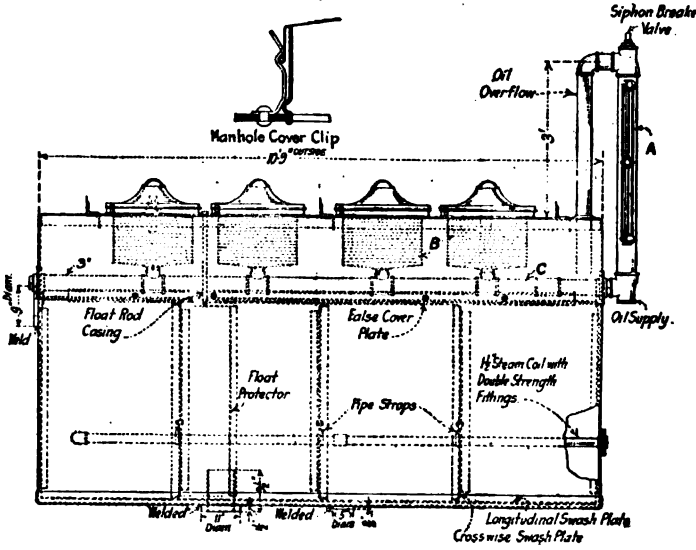
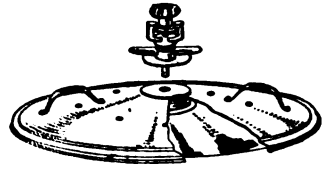


FIG. 142.—Bowser Fractional Oil Filter.

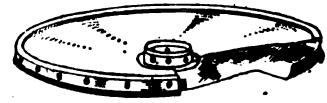
The Bowser continuous filter, shown in Fig. 142, works differently from the preceding one. The oil is supplied to a stand-pipe *A* which supplies oil to the filter bags *B*. These are in frames which sit on the oil supply line

C in the filter tank. When in place they hold open the oil supply valve. This automatically closes when the filter bag is raised for cleaning. This is shown clearly in Fig. 143, which gives the details of the filter section. This section contains 20 sq. ft. of effective filter service. The filtering medium is a



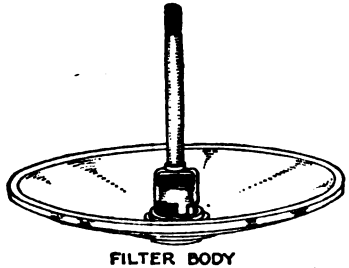
COVER DISC.

FILTER DISC SHOWING SCREEN WIRE AND FILTER CLOTH.

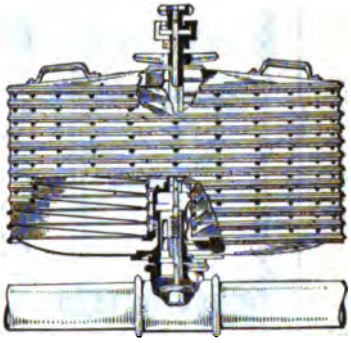


FILTER DISC SHOWING SCREEN WIRE.

FILTER CLOTH.



FILTER BODY



ASSEMBLED FILTER SECTION IN PLACE ON FILTER SUPPLY PIPE, SHOWING INTERIOR CONSTRUCTION, OIL PASSAGES AND AUTOMATIC OIL INLET VALVE.

FILTER SECTION ASSEMBLY

Fig. 143.—Details of Bowser Filter Section.

fine mesh muslin about 64 × 64 threads to the inch. The rating of the filter sections depends on the viscosity, temperature and the amount of foreign matter contained in the oil and hence varies from 100 to 300 gal. per hr. These sections can be removed and cleaned while in operation.

All oil above the capacity of the filter sections overflows from the standpipe

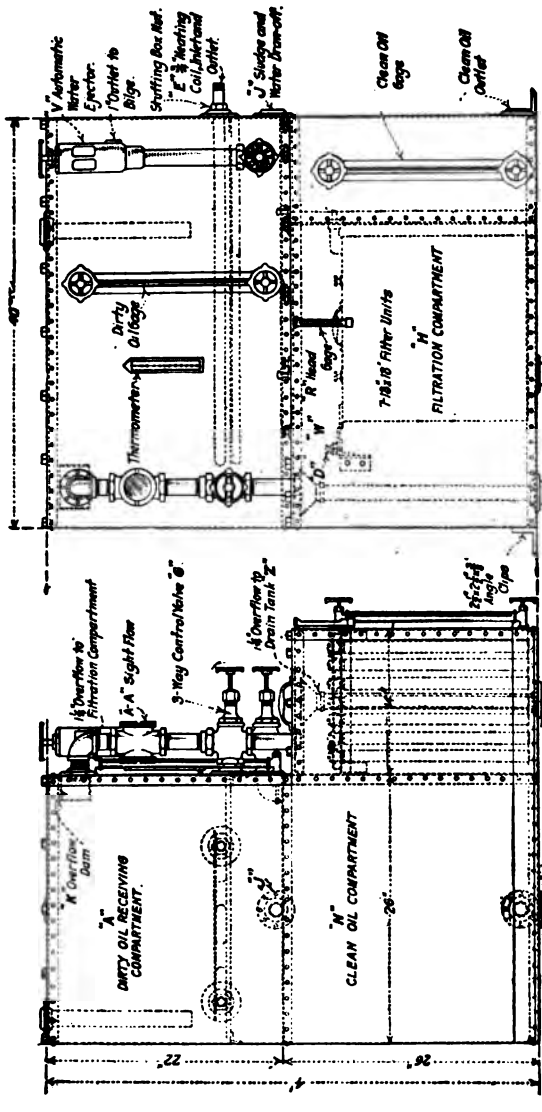
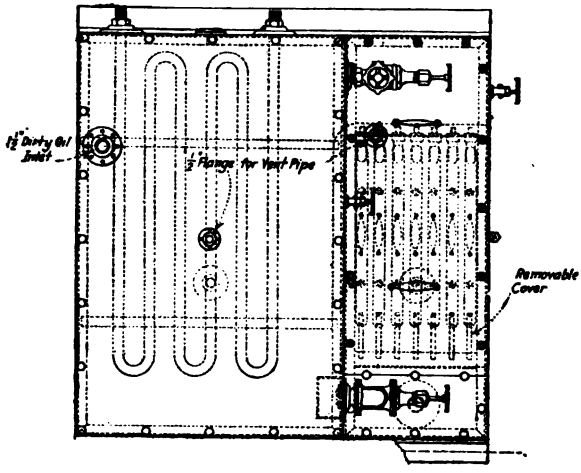


Fig. 144.—Richardson-Phenix Batch Filter.

into the main tank. A false cover is placed in this tank together with swash plates to prevent splashing of the oil when the ship rolls.

The Richardson-Phenix Co. have developed an excellent form of batch filter for marine turbines in which batches of oil of a given amount are drawn off from reserve tanks or pumped from the drain tank to the filter where the oil is allowed to rest and to filter at leisure and then to return to the system. This filter is shown in detail in Figs. 144 and 145. It consists of three com-



CAPACITIES

Receiving Compartment	100 Gallons	
Clean Oil "	100 "	to C.L. of Nozzles
Filtration "	35 "	" " " "

Effective Filtering Surface–31½ Sq. Ft.
Filtering Capacity–100 Gal. per. Hr. with 2" Head Over Units
Heating Surface–6 Sq. Ft.

FIG. 145.—Plan of Richardson-Phenix Batch Oil Filter.

partments, a receiving or precipitation compartment *A*, which contains the steam heating coils *E*, and a drain opening *J* for water and sludge, or automatic water ejector *V* is used. The oil in *A* rises over *K* and flows through sight flow indicator "*A-A*" into filtration compartment *H*, if used as continuous filter in port. When used on batches the whole batch is drained into the filtration chamber through three way valve *G*. The excess oil overflows through the overflow pipe *Z* back to the lubricating system. A portion of the oil passes through filters into the clean oil compartment *N* from which it may be returned to the oil system. The filter bags are shown in detail in Figs. 146 and 147. These filter units consist of two heavy galvanized wire screens placed about 1 inch apart and held in a suitable frame. The filter cloth in the form of a bag is drawn over this frame and held in place by a suitable clamping device and two thumb nuts. The clean oil leaves through a discharge spout in the partition of the filtration chamber. When the filter section is lifted out, a spring check valve automatically closes the opening of the discharge

spout into the clean oil chamber. Fig. 147 shows the path of the oil through this filter bag. The capacity of the particular filter shown in Fig. 144 is about 100 gal. per hour.

The shells of these filters consist of $\frac{1}{16}$-in. steel plates with lap seams closely riveted and caulked and covered with black enamel on the outside. Mill scale is removed by pickling before assembly.

Richardson-Phenix have also furnished a considerable number of continuous filters that operate on the same general principle as the Westinghouse continuous filter but with their own particular filter bags.

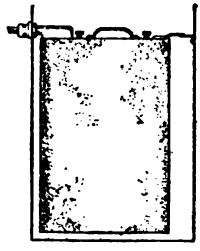


FIG. 146.

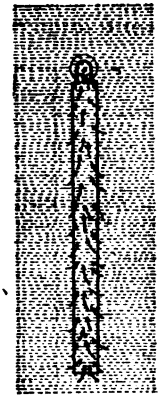


FIG. 147.

The DeLaval Separator Co. provide conical bottoms to their storage tanks and the oil supply to the turbines is drawn from above this conical bottom and passed through a duplex strainer on its way to the bearings. The coarser impurities, water, etc., settle in these conical bottoms. From here this matter is drawn off periodically and carried down to a DeLaval centrifugal oil purifier built on the principle of a cream separator.

Fig. 148 shows a DeLaval geared purifier. The gearing is replaced by a simple impulse turbine wheel on the spindle for marine use. The weight of bowl spindle and gear is carried by two adjustable tread wheels in the lower base. Two covers are placed above the bowl to receive the discharges of water and oil. A third cover, known as the "overflow cover" is provided above the other two covers. This receives the overflow if at any time the bowl becomes clogged and discharge ceases from the regular covers. This serves as an automatic signal that the bowl needs cleaning. All bearings are properly lubricated. Still another cover is placed above these three overflow covers. This receives the oil to be purified and regulates its rate of flow into the separator proper.

The bowl is the essential working part of the unit and is shown in Fig. 149. The oil to be purified is fed into the top of the bowl and passes down the centre to the bottom whence it flows out and up through the holes in the discs and is distributed in thin layers between the discs.

Separation takes place under the influence of centrifugal force. The material which is heavier than water, such as sand and metallic particles, is thrown outward to the periphery of the bowl and held there in a sediment pocket by centrifugal force. The water, being heavier than the oil, also passes outward and upward along the outer edge of the discs and from there to the discharge outlet.

The oil, being lighter, passes inward between the discs where the thin layers are subjected to tremendous centrifugal force which throws out the remaining traces of impurities and water. These impurities are forced along the lower surface of each disc toward the sediment pocket and the pure oil is forced toward the centre shaft and from there upward to the oil discharge outlet. The bowl automatically adjusts itself to varying proportions of oil and water and if no water at all is present, oil will discharge only from the pure oil

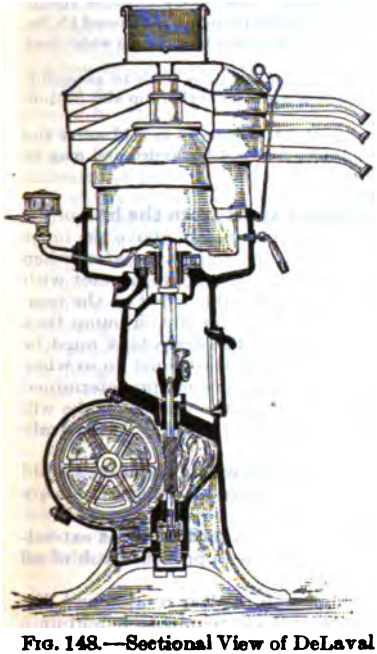


FIG. 148.—Sectional View of DeLaval Oil Purifier.

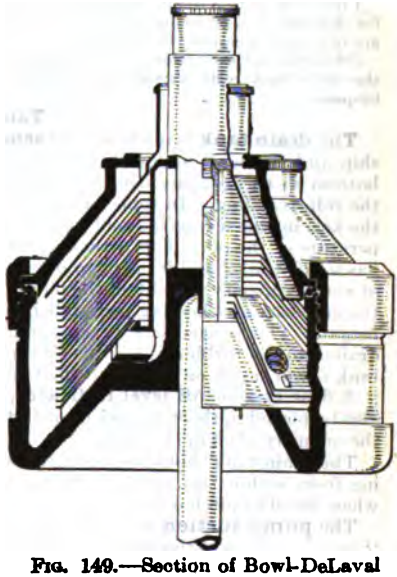


FIG. 149.—Section of Bowl-DeLaval Oil Purifier.

outlet. Conversely if no oil is present water will leave by the water outlet only.

These oil separators have shown up well under exacting conditions on U. S. Naval vessels.

Oil Coolers

The subject of oil coolers is treated fully in Section 10, p. 1111.

The Emergency Fleet requires coolers to be of such size that the oil circulated is cooled from 130 deg. to 110 deg. with cooling water at 70 deg., with a drop in oil pressure not exceeding 15 lbs. per sq. in. and a drop in water pressure not exceeding 5 lbs. per sq. in.

Coolers must be installed in duplicate and the piping must be so designed that either cooler may be put in service.

Heat transfer in an oil cooler is influenced by a number of considerations. It decreases as the viscosity of the oil increases so that heavy oils are not as easily cooled as the lighter oils. It varies with both the water and oil velocities. The temperature of the cooling water also affects heat transfer. With cold water of 40 deg., there is a tendency to

form a thick viscous film of oil on the cooling surfaces which materially decreases the heat transfer. With cooling water at 70 deg. F. and hot oil of about 250 viscosity, at 140 deg. F., the heat transfer with oil velocities of 2½ ft. per sec. and water velocities of 1½ ft. per sec. may, for estimating purposes, be assumed to be 60 B.t.u. per sq. ft. per deg. difference in temperature per hr.

Thermometers must be placed so that the temperature of the oil entering and leaving the coolers can be easily observed.

The cooling water should be supplied by a separate circulating pump with connections to the main or other circulating pump as an auxiliary. The water pressure should always be less than the oil pressure in the cooler and should therefore not exceed 15 lbs. per sq. in. The water side of the cooler should have a connection from which water may be drawn to test for oil.

Coolers should be so constructed that the water side of the tubes shall be accessible for cleaning, as they frequently become foul. Coolers with removable top and bottom are thus very accessible for cleaning.

Oil should never be allowed to cool below 90 deg. F. If the ship is in cold water and the drain tank rests on the ship's bottom, the cooler may not be needed and may be by-passed.

Tanks

The drain tank is made of steel **separate and apart from the hull** of the ship and is **placed** as low down as possible, usually directly above the inner bottom so as to insure complete drainage of gears and bearings even when the ship is rolling. In some cases it has been placed directly in contact with the keel in which case the cold sea water has an appreciable effect on the temperature of the oil. This arrangement is not desirable for the oil pump then has to lift the oil and it is apt to become air bound. The drain tank must be of ample capacity to hold all the oil in the system or at least must do so when the gear case and bearing pedestals have also filled up to some predetermined height. This is necessary because, on shutting down, the gravity tanks will drain into the turbine unless the oil supply is promptly shut off. If the drain tank is too small, oil wastes to the bilge.

A float type of **oil level indicator**, located as shown in Fig. 139, should also be placed in this tank, with the indicator stem so graduated as to indicate the quantity of oil in the tank.

The drain tank should be provided with two vertical **baffle plates** extending from within 1 in. of the bottom to 6 in. of the top to prevent wash of oil when the ship pitches or rolls.

The **pump suction** must be placed high enough to clear residue and dirt that may collect in the bottom. The Emergency Fleet requires a hand pump to remove this residue from the bottom of the tank and deliver it to the filter. **Manholes** must be provided to allow each section to be thoroughly cleaned.

The **drain pipes** from gears and bearings discharge into the top of the tank. A goose-neck pipe extending to the height of the centre line of the turbine serves as an **air vent** on this tank.

This tank must be **cleaned at regular intervals** and waste or linty materials must not be used for this purpose; water should never be used to wash down the tank.

Filter and storage tanks are usually mounted on the bulkhead or hatch at a sufficient height to give the necessary oil pressure at the bearings after allowing for friction in pipes, strainers, etc. This tank or tanks should always be provided with a gage glass to indicate the oil level as well as with a float-controlled low-level alarm.

The reserve oil tank is also mounted on the bulkhead or in the hatch and is available for emergency use. The capacity of this tank is generally about ½ the total capacity of the system though a larger supply is desirable. Sometimes several reserve tanks are used.

Oil Piping

The most desirable **material** for oil piping is seamless copper with brazed **flanges** and cast iron or brass **fittings.** Copper does not scale or corrode and as it can be readily bent, few fittings are needed.

Brass pipe is also extensively used. This has to be threaded to fittings and it is difficult to keep threads tight with hot oil. Brass is not so easily bent as copper.

On account of the shortage of these two materials, wrought iron and steel pipe have been used extensively of late. These materials are undesirable because they corrode and scale in service. Hence they should be cleaned by pickling in hot acid before installation, and after the system is finished, oil at 180 deg. F. should be circulated vigorously through all the system for several hours, to loosen and remove any remaining scale.

Threads must be clean cut and sharp and screwed up tight. Shellac is largely used on threads of oil pipes to ensure tightness. On account of the difficulty of making tight joints, **welding** has recently been extensively used with entire satisfaction on oil piping.

Heavy manilla paper or compressed fibre make the best **gaskets** on oil lines; some set these in brown shellac when tightening joints. This, while oil tight, **makes** the joint hard to break again if it becomes necessary.

Only **gate valves** should be used on oil lines except drip valves and other small valves. Globe valves on flow lines set up too much friction and act as collecting places for dirt.

Flanged joints should be provided on all tanks, strainers, filters, separators, pumps, gear casings and manifolds.

All screwed piping must be reamed at the ends to remove burrs.

PIPING CONNECTIONS

Steam Lines

Steam Piping. Formerly copper pipe was used exclusively for steam pipes. It was easily bent to conform with ship structure and had great flexibility. However, with high pressures and high superheat copper pipe had to be made very heavy. Furthermore, its reliability when subjected to superheat is open to question.

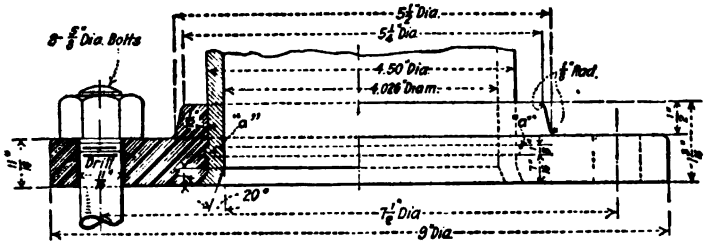


FIG. 150.—Lovekin Joint.

Open-hearth lap-welded or seamless steel pipe is now being generally used for **main steam lines.** The seamless pipe is best but is expensive and not always available. Lloyd's rules do not permit Bessemer pipe to be used, although this is largely used on land. Open-hearth lap-welded steel pipe is becoming the standard on **auxiliary steam lines.** When exposed to sea water this must be kept well painted to prevent corrosion.

Pipe Joints. A very popular form of joint on both steel and copper high-pressure lines is the "**Lovekin joint,**" shown in Fig. 150. The flange is of steel with grooves "*a*"*a*" turned in its inner bore and the pipe is rolled out cold into these grooves by a special expanding machine. The **screwed and-peaned flange** is largely used on auxiliary lines carrying up to 125 lbs. pressure. The **Van Stone joint** has been used for years in land practice with complete satisfaction under many adverse conditions. It has not been used on shipboard probably owing to prejudice on the part of shipbuilders. Both the Van Stone and welded joints will probably be in common use in a few years. Marine engineers prefer the **flat-face joint** to one with raised faces. The flanges can thus be interchangeable whether at bulkheads or at other points on the line. The flat-face joint can also be handled better when gaskets blow out at sea.

Long radius pipe bends, p. 1267, should be installed to take care of the expansion of the steam connection from the bulkhead to the turbines and also from boilers to bulkhead. These can easily care for all expansion in average ship installations, even when steel pipe, particularly seamless, is used.

Strainer. The main steam passes through a basket strainer in a strainer casing before entering the throttle valve. One form is shown in Fig. 151. The strainer is generally of perforated steel with holes $\frac{1}{16}$ in. in diam. Care must be taken to see that the clear opening through the holes is greater than the area of the pipe, or an excessive drop in pressure may occur. The basket should be made of brass if wet steam is used.

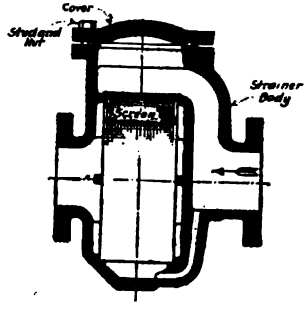


FIG. 151.—Steam Strainer.

Steam Speeds. In designing steam mains for turbines using **wet steam** at 200 lbs. boiler pressures, 8000 ft. per min. steam velocity can be allowed. With **superheated steam** at this or higher pressures, steam velocities of 10,000 ft. per min. are considered good practice with steam turbines.

Gaskets may be of any high-grade material with asbestos base suitable for high superheat and high pressure.

The steam connection to the astern turbine is usually taken off the main steam chest, and is under the control of the main astern valve. A stop valve is usually placed in this connection beyond the astern control valve to prevent any leakage to the astern turbine during forward operation. Such a connection is shown in Fig. 152.

When high- and low-pressure turbines are used, provision must be made to take high-pressure steam from a point below the strainer and in front of the main "ahead" valve to the low-pressure cylinder for emergency service in case of an accident to the high-pressure turbine or its gears. A stop valve should be placed in this line as shown in Fig. 152 to prevent leakage to the low pressure. This pipe should be designed sufficiently small so that no excessive pressure can accumulate in the low-pressure cylinder even if the valve is full open. A similar connection must also be provided to the low-pressure astern unit also for emergency use. As these connections are for emergency service only they do not need to be manipulated from the main control

valves, for the valves can be adjusted by hand under such exceptional conditions.

Exhaust Piping. The exhaust from marine turbines usually leaves either the top or upper side of the casing and enters the condenser at the top or side. The condenser generally has to be set higher than the turbine in order to get sufficient head room for a hot well and condensate pumps. **The exhaust pipe** may be made of cast iron, in which case it must be quite heavy and have substantial ribs to withstand the collapsing effect of the vacuum. If flat surfaces are used, internal spacing plates are often used to stiffen these sections. Steel exhaust pipes are usually built up of riveted steel plates. They weigh less but are more apt to leak air at high vacuum.

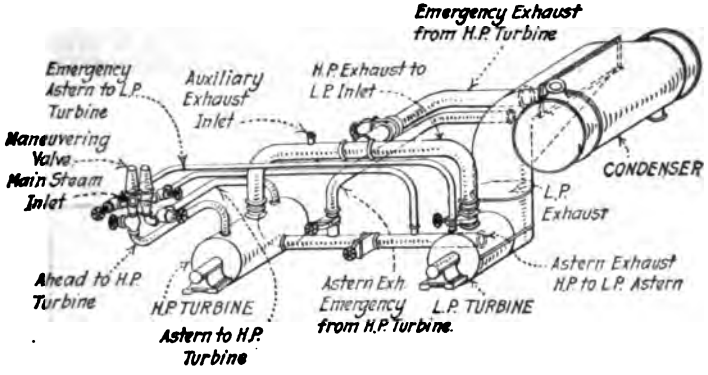


FIG. 152.—Arrangement of Pipes and Valves Westinghouse Cross Compound Marine Turbine.

Gaskets on exhaust lines are usually of red rubber, cloth inserted with vulcanized joints.

Steam velocities in exhaust piping may range from 18,000 to 24,000 ft. per min. at full load. Higher velocities have been allowed in land practice.

Copper **expansion joints** or bellows-type welded-steel joints are placed on all exhaust and low-pressure pipes where expansion may occur that cannot be taken care of by bends.

Traps and Drains

When wet steam is used, a separator should be placed on the main steam line with a trap of the pivoted bucket type, see p. 1341, which discharges into the feed and filter tank. Other high-pressure drains can discharge through traps to the same place.

It is necessary to drain the exhaust end of a single-cylinder turbine. This is usually quite close to the inner bottom. If a reciprocating wet air pump or a wet vacuum pump with a cylinder below this level is used with the condenser, this drain can be led directly to either of these pumps with a metal disc swing check valve in the drip line, see p. 628. This cannot be done with a centrifugal hot-well pump; in that case a receiver is generally placed directly below the turbine cylinder and the condensate is allowed to collect in this. It is

lifted periodically to the condenser by a steam ejector operated by the watch engineer. A gage glass must be provided to indicate the amount of water in the receiver.

When high- and low-pressure turbines are used, the draining of the high-pressure end presents a difficult problem. It can be dripped directly into the low-pressure end by a pipe without valves but this involves a constant loss of steam.

The best plan is to drip as shown in Fig. 89. Receivers and ejectors, similar to the one described above, are placed under both turbines and drained as shown in the sketch with swing check valves in each discharge line.

The condensate can be bled directly from the high-pressure turbine to the condenser when in normal operation for there will be a considerable pressure in the exhaust end of the high-pressure cylinder. When running astern the high-pressure casing will contain a vacuum and the receiver can then be discharged by the ejector. The low-pressure cylinder would be drained as already described for the single-cylinder machine in all conditions of operation.

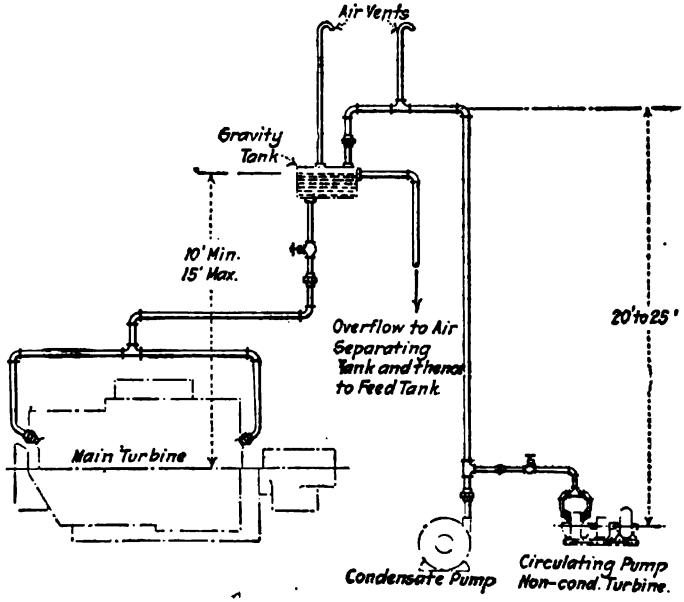


FIG. 153.—Gravity Gland Water Supply System when Space is Available for Gravity Tank.

Gland Piping

Turbines with steam-sealed glands have a number of pipes for supplying steam for this service. In some impulse machines a portion of steam leaking from the gland on the forward end is used to seal the astern end and when running astern the reverse action occurs. The remaining leakage is fre-

quently led to one of the low-pressure stages of the turbine. At low speeds the valve in this connection must be closed and live steam must frequently be supplied to the glands in addition. Hence a connection must be made from the main live steam line to the gland casing.

Considerable attention must be given to the operation of all steam-sealed glands. Turbine manufacturers are prepared to give detailed instructions for handling their machines and watch engineers should be made thoroughly familiar with all adjustments for all loads, if highest economy is to be secured. Every change of speed of the ship makes a change of gland steam necessary. Valve handles on gland steam lines should be plainly marked and should be made very accessible by extension handles and universal joints.

A supply of water under a pressure of about 5 lbs. above the pressure in the casing must be furnished when water glands are used. This water must be fresh and condensate is usually used. Fig. 153 shows such a system where an overhead tank can be supplied. In some cases this room is not available, as in destroyers, and a plan such as Fig. 154 has been used. An orifice in a

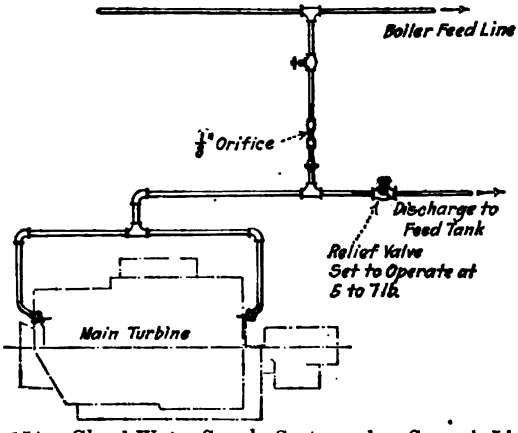


FIG. 154.—Gland Water Supply System when Space is Limited.

branch from the boiler feed line furnishes just a little more water than the glands need. The remainder is discharged back to the feed tank. Where glands combining water seal for high speed and steam seal for slow speed are used, a pipe must lead to the steam seals from the high-pressure steam main. Valves must be provided so that either supply may be cut off or turned on quickly and easily.

FOUNDATIONS

Geared steam turbine sets require very strong and rigid framing under all turbines, gears, thrust blocks and shaft. Otherwise troubles occur due to changes in alignment and distortion of the machines. Some of the troubles experienced with gears have been laid to this cause. The **hull framing** must be strong and well braced and amply heavy to carry the load imposed

on it. It must be designed so as to conform to the general framing of the foundation under the turbine and gear base. A typical foundation for turbine, gear case and thrust block is shown in Fig. 155.

The foundations should be of **continuous sections under the turbine, gear casing and thrust bearing,** and should be strongly stiffened with fore and aft and athwartship brackets. Every effort should be made to have athwartship bracketing in alignment with ship's floors and to have foundations tied into ship's structure in a substantial manner. Transverse girders should be placed under each foot pad of the turbines and gears.

No plates in these foundations **should be less than ⅝ in. thick.** The vertical plates should extend the full distance from the sole plate to the tank top. The **sole plate** should be not less than 1 in. thick and should be continuous over the entire length of the foundation. The sole plate must be connected to the foundation by double angle bars. These must also be used to connect the foundation to the ship's floor. All **rivets** in the sole plates should have countersunk heads and should be made flush with the face of the sole plate.

The method of lining up turbines, engines or gearing is to use forcing screws or wedges so as to obtain the proper location. Where supporting screws are used the overhang of the sole plate can be supported by small screw jacks. Adjusting washers or chocks are then fitted between

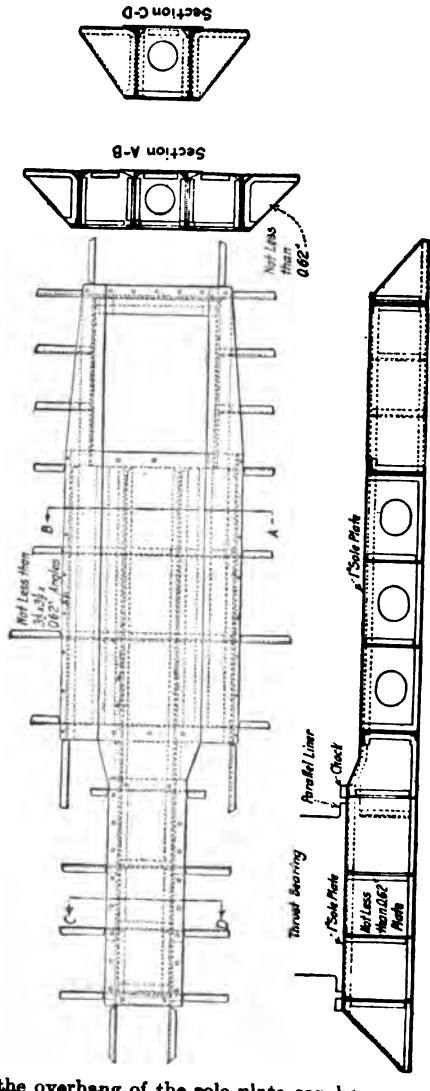


FIG. 155.—Turbine, Gear and Thrust Foundation.

the bed and the sole plate, the sole plate having previously been ground off to a smooth even surface. This fitting must be done mostly by hand and by high priced skilled labor. The liners must fit to the thousandth part of an inch so each will carry its proper proportion of the load.

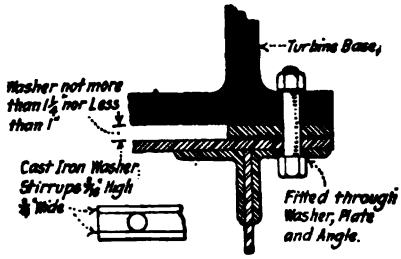

Turbine Base
Washer not more than 1¼ nor Less than 1"
Cast Iron Washer
Stirrups ⅝" High
⅝" Wide
Fitted through Washer, Plate and Angle.

FIG. 156.—Method of Bolting Turbine and Gear Casing to Foundation.

The rectangular **washers** or chocks, between the bottom of the casings and the sole plate should not be less than 1 in. thick nor more than 1¼ in. thick. They should have two chipping strips ¾ in. wide and 5/16 in. high on the top and bottom surfaces as shown in Fig. 156. The holes in these chocks

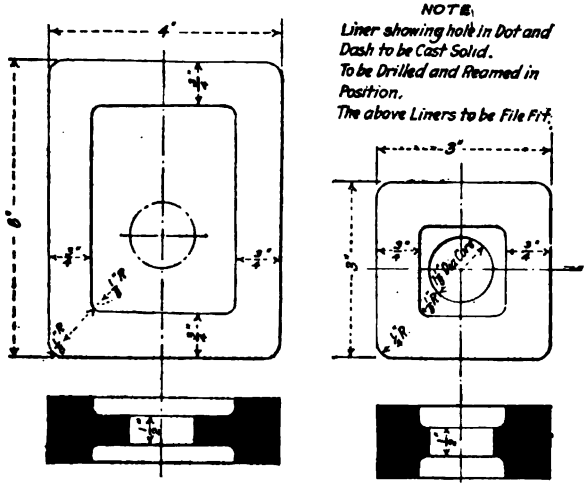


FIG. 157.—Solid Liners for Turbine Foundations.

must be reamed, not cored. If thin chocks are used, these should be of steel. Care should be taken to see that these washers extend well over the vertical plates of the **seatings**.

Some other forms of solid liners used on ship sole plates are shown in Fig. 157. Mr. Luther D. Lovekin has developed a new form of liner, shown in Fig. 158, which reduces the time of leveling and bolting down bed plates to about one-tenth that necessary with hand fitted chocks. The sole plate around the bolt hole is counterbored flat by using a bar in the bolt hole with an inserted cutter, or by means of a rose reamer. This counterbore insures that the liner will rest on a smooth parallel base. Its depth need only be sufficient to insure a flat surface. When the bed plate is lowered in place and leveled by means of wedges, the adjusting liners are placed in the counterbores.

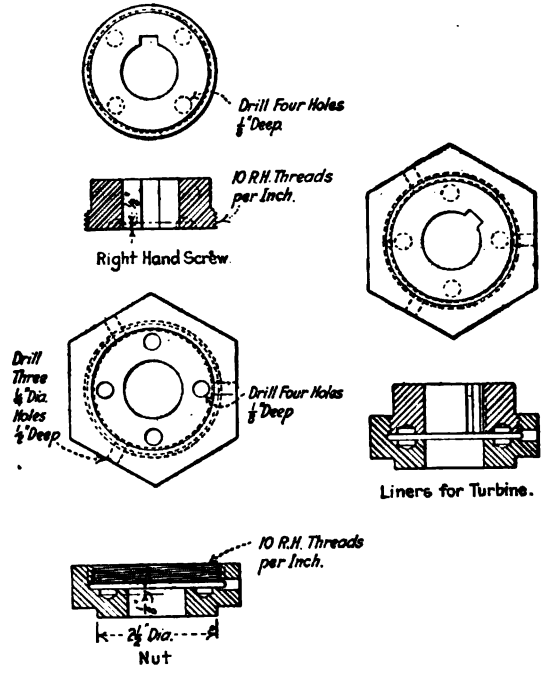


FIG. 158.—Details of Lovekin Foundation Liners.

The upper section is held by a spanner and the lower part turned until the liner is tight. A wooden plug is then put through the liner and type metal or babbitt poured under a slight head through the hole in the lower ring. This runs into the dowel holes in the two halves and locks them so that they cannot turn. A reamer is then run through bed plate liner and sole plate and fitted bolts are driven in place and screwed down. These rings can be made to have adequate surface to carry the load. After the bolt is in place a forcing screw is placed in the small hole through which the white metal was poured. This liner is a great time-saver on erection work. Fig. 159 shows the general arrangement of the assembled liner.

Unless impractical to do so, every through bolt connecting casing flange, washer and angle-bar should be a **fitted bolt.** All bolt holes must be spot-faced.

Holes for fitted bolts should be reamed through casing flange, cast iron washer and sole plate after alignment of unit.

The sole plate of the thrust bearing should have chocks either as part of the sole plate forging or securely riveted to it on both forward and after end of the foundation. A parallel **liner** should be fitted between the ends of the

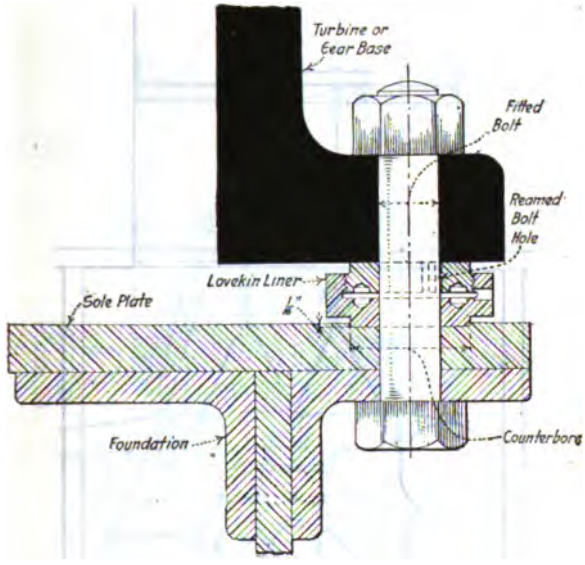


FIG. 159.—Assembly of Lovekin Foundation Liner.

bottom flange of the thrust bearings and the chocks to prevent fore and aft movement of the pedestal. The adjustment of thrust collars will be made by the use of jaw nuts on collar bearing screws, but the pedestal should be fixed permanently in its correct position. **The use of wedges is not allowable.**

LIFTING GEAR

Lifting gear must be provided on each ship to make repairs and to permit inspections of turbines and reduction gears. This hoisting equipment must be of small bulk which can be readily assembled and easily handled. It must be such that no possible accident can happen to the workman or damage occur to the turbine or gear parts when the gear is in use. Frequent careful inspection of all lifting gear parts must be made to avoid such accidents. All turbine parts must be handled with the greatest care for they can easily receive serious damage.

A great variety of lifting gears are in use. These vary from the simple block and tackles, and triplex hoists to elaborate lifting gear equipment

forming part of the turbine construction, such as is used in some British Transatlantic Steamships.

The simplest lifting gear is the chain hoist with suitable shackles, wire cables, ropes and slings. The framing of the decks above the turbine and gears is either made sufficiently heavy to allow a hoist to be connected to it by means of shackles, or special I-beams are fitted across the engine-room hatch. The chain hoists are fastened to these shackles or special beams and the parts are lifted by means of ropes and slings. Fig. 160 shows the use of such

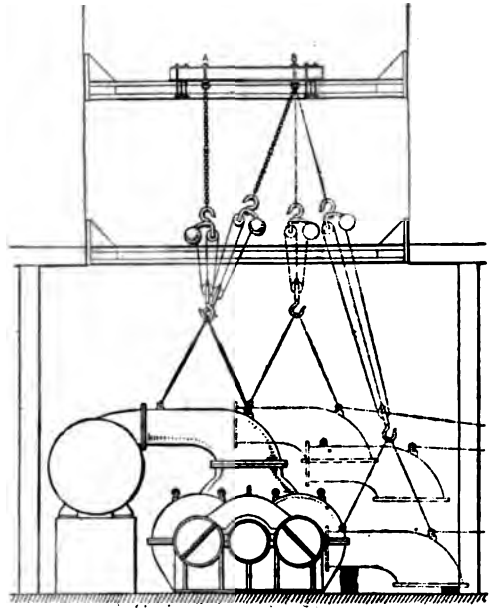


FIG. 160.—Lifting Gear for Exhaust Connection.

hoists and beams across the hatch when used to lift the **exhaust connection** from the turbine. When movable cross-beams are used, these must be securely held in place, preferably by bolts, when lifts are made. Care must also be taken to avoid excessive side pull on these beams or they may turn over.

The **turbine and gear case covers** can be lifted by hoists in a similar manner. In the case of turbine covers, it is the practice of many builders to provide a number of **jacking bolts** by means of which the joint may be broken before attempting to lift by the hoists. This prevents overstraining the hoist to break this joint. It is also advisable to provide four large **guide bolts,** one for each corner of the cylinder which fit certain reamed holes in the bottom half of the casing when the regular bolts are removed. The guide bolts hold the upper half of the casing central during lifting and thus prevent fouling of blades or diaphragms. Guide bolts are often marked in inches to enable the

engineer to note whether the casing is lifting evenly at all corners. If these lifted parts are placed on the engine-room floor, they should be set on heavy

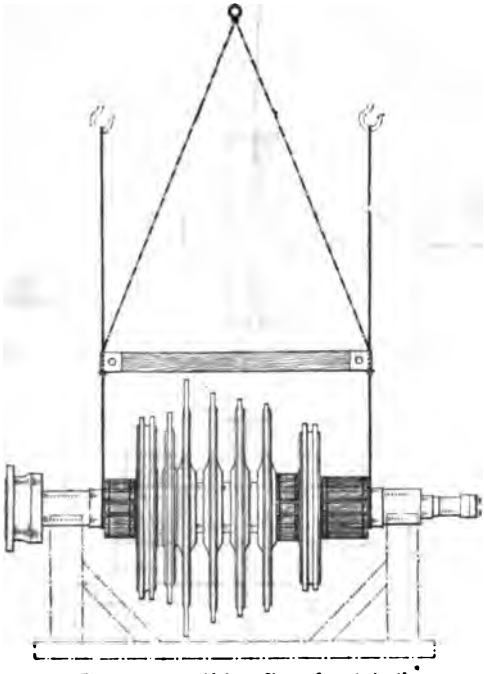


FIG. 161.—Lifting Gear for Spindle.

timbers to distribute the load over a considerable area of the inner bottom. It is frequently simply necessary to lift the cover for inspection. When raised to the desired height standards should be placed on the lower casing and the weight let down on these supports so as to relieve the lifting gear and ensure safety to the engineers during inspection.

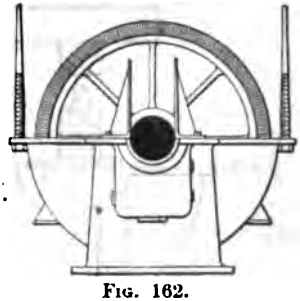


FIG. 162.

Turbine rotors are usually lifted by means of wire rope slings and a strong-back, as shown in Fig. 161. Great care must be taken in lifting the rotor so that no damage is done to blades, guides or packings. This can be best assured by providing four **cast iron guide frames** fastened to the lower half of the casing or bearing pedestals and thus forming a perfect guiding channel for the spindle. Fig. 162 shows both guide bolts for cover and guide channels for spindle. The packing rings and labyrinth packing should be carefully protected during handling by strips

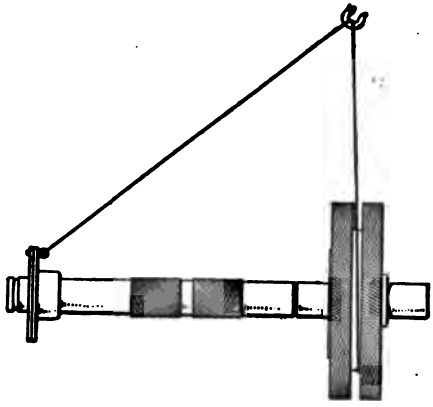


Fig. 163.

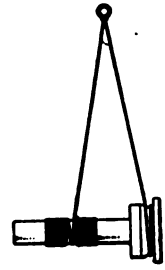


Fig. 164.

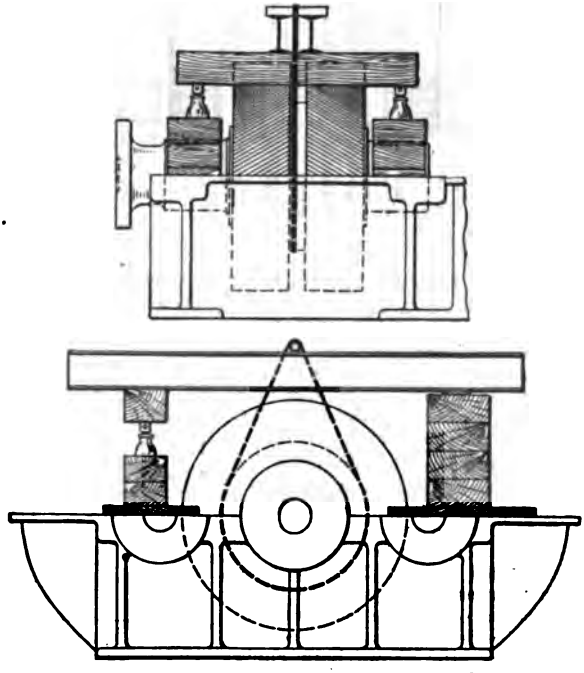


Fig. 165.—Method of Lifting Low Speed Gears.

of soft wood held in place by wires. If the rotor is removed from the casing it should be placed in a properly lashed supporting rack where care must be taken not to injure the journals.

When gears are to be lifted it is essential that the marked teeth be brought into mesh or else teeth then in mesh should be marked to ensure proper mesh on reassembling. Methods of removing a gear and pinion are shown in Figs. 163 and 164. Wood must be placed between the sling and the gear to prevent injury to the gear teeth. The gear should be lowered only on clean soft pine to prevent injury to the teeth and then must be lashed firmly in place.

Various devices may be employed to lift heavy low-speed gears to change bearings, etc. Screw jacks using beams as a lever are suitable for this purpose. Such an arrangement is shown in Fig. 165.

Some vessels provide a small **trolley hoist and crane-way** above turbine and gears that will permit the removal of all but the heaviest parts. This is a decided improvement over hoisting by temporarily rigged-up lifting gears as it is always available for use. Provision must be made for firmly lashing this trolley to a bulkhead or hatch when not in use.

Equal care must be taken in reassembling turbine and gear parts as in dismantling them. Particular care must be taken to lower the spindle and cover slowly and evenly into place or blades will foul each other or foul the diaphragms.

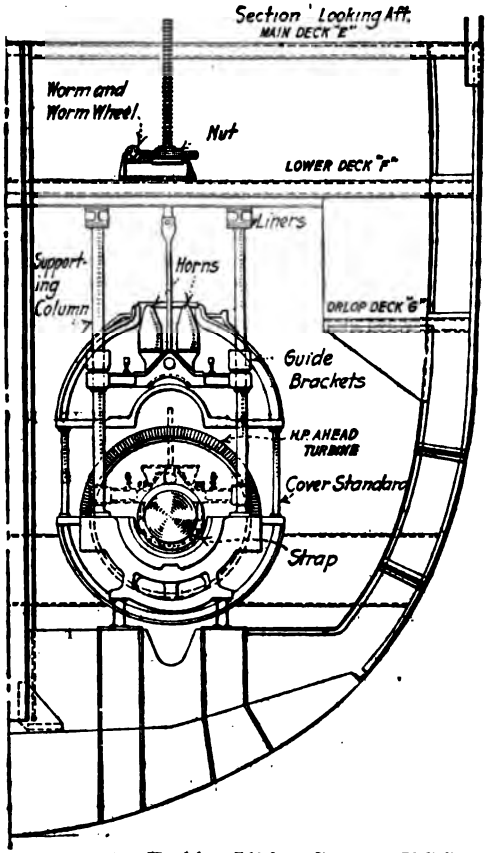


FIG. 166.—Turbine Lifting Gear on T.S.S. Aquitania.

A more elaborate **lifting device** is that shown in Fig. 166 which was **furnished on T.S.S. Aquitania** and is typical of the equipment on many British vessels. Heavy beams are built over the ends of the turbine into the framing of the deck above. A casting is fastened above these beams

carrying a vertical nut with square threads. A worm wheel is fastened to the outside of the nut and is driven by a worm on a motor-driven shaft. Large vertical eye bolts with square threads pass through these lifting nuts. A steel cross-beam is fastened to the eye at the lower end of this bolt. Each end of the cross-beam is bushed and slides up and down on a vertical column which acts as a guide. The lower ends of these columns rest on the top of the lower half of the turbine casing. The top end is fastened to the heavy beams of the deck above, by means of liners when lifting is to be done. These columns act as supporting struts to the framing carrying the lifting nut. When not being used, the liners are removed so that any distortion of the hull at sea is not communicated through the columns to the turbine casing.

The upper half of the casing is provided with horns which rest on the cross-beam when lifting takes place. The cover is guided by brackets encircling the guide columns. When the cover has been lifted the desired height, its weight is taken on steel standards resting on the joint of the lower half of the casing.

The beams may then be lowered to lift the spindle. This lifting is done by passing a strap around the shaft and fastening it by bolts and nuts to the steel beam. The rotor is supported after lifting to the desired height by wooden blocks set under the bearings.

The worm and worm wheel drive from a single shaft ensures a uniform rate of lifting of each end of the shaft or cover which is a very desirable feature. This gear is operated from a variable speed motor which gives easy and suitable gradation of control of lifting speeds.

MECHANICAL REDUCTION GEARING

This subject is very fully treated on p. 818.

ELECTRICAL GEARING

A second method of securing the advantages of the combination of high speed turbine and slow speed propeller is by means of electrical gearing. The high speed turbine in this system drives a turbo-alternator which furnishes current through proper controlling, regulating and reversing switch gear to specially wound slow speed alternating current motors, or in some late installations to high speed induction motors geared to the slow speed propeller shaft by mechanical toothed gearing. Future installations will probably embody the mechanical toothed gearing between the induction motor and the propeller shaft with the turbine direct connected to the generator.

As compared with other methods of speed reduction, such as mechanical gears, etc., the electric drive appears to have the following advantages: (*a*) the ratio of speed reduction can be varied by winding the motors so that different numbers of poles may be used, as desired; (*b*) at low powers only a part of the plant may be used, for instance if two turbo-generators are installed, one may be shut down at low speeds; (*c*) there is no need for a reversing turbine for the motors can easily be reversed with the main generator continuing to run in the same direction. This decreases the weight and size of the main turbine and the elimination of the idle reversing wheels at the astern end increases its over-all efficiency. It has been stated that the friction losses in a turbine wheel when operated in the reverse direction is at least eight times as great as in the normal direction. Hence the importance of the elimination of the reverse elements is apparent.

The full power of the motors is available for reverse which is not the case in the ordinary reversing turbine, where usually only 60 per cent of full power is available on the astern wheels on reverse. The turbo-alternator continues to

run in the same direction for both forward and reverse operation of the motors. Reversal is taken care of by the switching gear. Furthermore, it is possible to purchase and install the standard land types of turbo-alternators and motors for this service with but slight alterations and the cost is thereby reduced.

Most of these advantages are of more importance in the case of naval vessels that require a wide range of speed and power than in the case of merchant ships, although the electric drive has been applied successfully to both purposes.

Electric propelling machinery will be generally heavier and more expensive than the best developments of geared turbine machinery. It overcomes the mechanical difficulties of high speed high reduction ratio mechanical gearing. Experience with this use of electricity on shipboard has been quite satisfactory and with ordinary care it should be possible to operate the electric drive with no more difficulty than with motors on land.

In the case of mechanical gearing, if trouble occurs with any part, it usually means a shutdown of the whole interconnected system. In the case of the electric drive any unit can be shut down without seriously interfering with the other units in the case of several screws and two or more turbo-generators. As regards efficiency, while the mechanical geared set may be slightly more efficient at full load, it is claimed that the electric drive is superior at slow speeds and particularly at cruising speeds. High superheat can also be used in the turbines and the steam consumption is lowered by its use.

Data is lacking to enable a comparison to be made of mechanical geared and electric drive installations. It is probable that each will find its best field of service as experience with each is broadened. The electric drive seems well suited for cruising services at variable speed, while the mechanical gear is most efficient on long full-speed runs. Recent developments indicate little difference in efficiency between the two systems at average power.

U. S. S. Jupiter

One of the earliest installations of the electric drive was in the U. S. Collier "Jupiter," fully described by Lieut. S. M. Robinson, *Jour. A.S.N.E.*, Nov., 1913, and May, 1914, from which the following notes are abstracted:

The engine room layout is shown in Fig. 167.

A General Electric turbo generator of 5450 kw. rated capacity with a normal speed of 1990 R.P.M., produces three phase current at 2300 normal volts and 33.2 frequency.

The turbine is of the standard stationary type with one Curtis and eight Rateau stages. It is supplied with steam at 190 lbs. per sq. in. gage and exhausts into a 10,000 sq. ft. surface condenser. The generator is totally enclosed. Its rotor has fans on each end which take air at the bottom of the casing and blow it out through the windings. Such cooling air is discharged into a duct on top of the generator casing. This duct leads to the suction of the forced-draft fans for the boilers, thus recovering the heat generated by electrical losses. Exciting current is furnished by the 35 kw. ship lighting sets.

The turbo-generator furnishes power to the motors, the function of the generator and motors being solely that of a reduction gear. All changes of speed are made by changing the speed of the turbine. This varies the frequency of the generator and consequently the speed of the motors. The generators are 2 pole and the motors 36 pole, practically giving a ratio of 1 to 18. In this case the motors are wound for 36 poles only. In other ships

the winding is arranged so fewer poles may be used, thus giving a lesser speed ratio and greater ship speed.

Two induction motors are furnished, one on each propeller shaft. Each motor is of 2750 H.P. and runs at a normal speed of 110 R.P.M. The motor frames are wholly enclosed and ducts carry away the heat from the windings to the forced draft fans. The motors are placed in water-tight pits. As an extra precaution the windings were made waterproof and can operate partly submerged.

The standard windings of a variable speed induction motor have been modified to give suitable control for starting and for reversing by means of short-circuiting devices and two water-cooled rheostats.

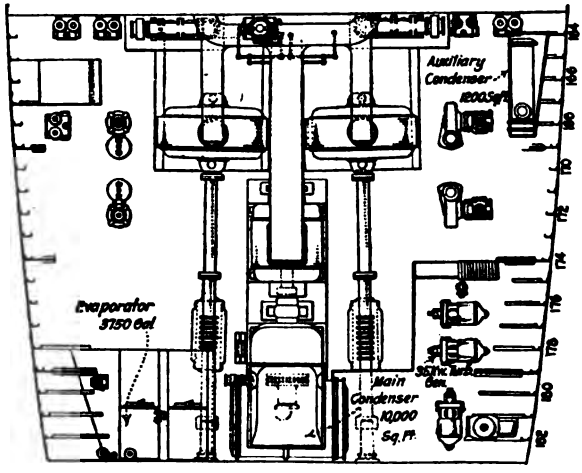


FIG. 167.—Arrangement of Engine Room U.S.S. Jupiter.

The main switchboard carries the ahead and astern oil switches, together with all electrical meters, speed counters, etc. The control stand is placed in front of the switchboard and consists of a small hand wheel working around a dial. All operating levers and switches are interlocked.

Both motors must necessarily run at the same speed and as the governor of the main turbine is sensitive, there will be no racing of the propellers, should these throw out of the water in a heavy sea. The speed can always be adjusted by a control wheel which increases or decreases the tension on the spring of the turbo-generator governor, thus changing the frequency of the set.

The performance of this particular unit is given on p. 721.

U. S. S. New Mexico

The electric drive appears to have very distinct advantages for vessels that run frequently at cruising speed. After the successful performance of the U. S. Collier Jupiter, the U. S. Navy Department contracted for an electrically propelled battleship, the U. S. S. New Mexico, 624 ft. long and with a displacement of 32,000 tons and a speed of 21 knots. This ship is

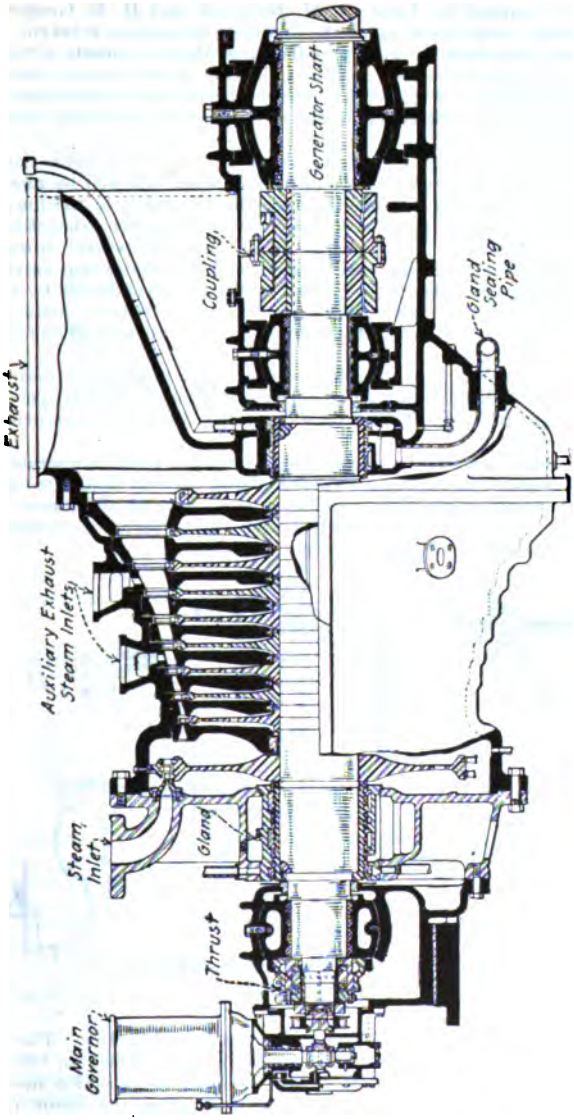


FIG. 168.—Turbine for U.S.S. New Mexico.

described in a paper by Lieut. S. M. Robinson and H. B. Gregory, *Jour. A.S.N.E.*, May, 1919, from which the following description is taken.

The main propelling machinery of the New Mexico consists of two turbo-alternators supplying current to four induction motors direct connected to the propeller shafts. Two 300 kw. direct current generators furnish excitation and power to various auxiliaries. Two motor generator boosters are also provided.

The turbo-alternators are of the standard General Electric land type. The turbine, Fig. 168 consists of one Curtis stage followed by nine Rateau stages. The casing has two openings for the admission to the lower stages of exhaust steam either from the exciting units or from the exhaust line of the other ship auxiliaries. This exhaust steam is under perfect control by a suitable arrangement of stop check valves and automatic trip valves. The auxiliary exhaust can also be by-passed to the main condenser by means of a spring loaded valve. Provision is also made so that the main governor opens this valve and sends the steam direct to the condenser when main steam is cut off the turbine.

The main governor is adjustable and can be set to give any desired speed from 700 to 2200 R.P.M. and when once set will maintain that speed regardless of changes in load. The governor operates through a compensated oil relay mechanism.

An emergency governor is also placed in the main shaft to operate at overspeed. It trips the main throttle valve and also the valves admitting exhaust steam to the lower stages of the turbine, thus shutting off all sources of supply of steam to the turbine. Experience with this governor indicates that it should be so modified as to also trip the nozzle valves.

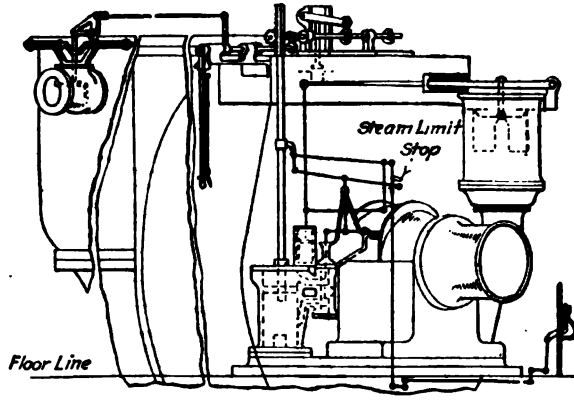


FIG. 169.—Arrangement of Governor Mechanism, U.S.S. New Mexico.

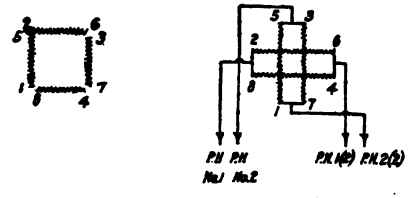
The governor gear is shown diagrammatically in Fig. 169. The governor weights operate a small valve of a hydraulic relay which in turn admits oil pressure to a piston driving a rack which drives a pinion mounted on a cam shaft. The cams open successive valves on the steam chest, admitting the desired amount of steam to successive groups of nozzles.

Adjustment of this control mechanism can be altered from the control

board of the whole system by means of bell-cranks and rods, as shown in Fig. 169. The fulcrum about which the governor weights act can be altered, requiring the turbine to run at a different speed for each position of the fulcrum.

Another system of bell cranks and levers from the control board operate a steam limit stop which limits the amount of steam that the turbine can take at any setting. This steam limit is necessary in connection with the manouvering of the ship.

The generators are normally rated at 11,500 kw. each at 80 per cent power factor at a normal speed of 2100 R.P.M. and have an overload rating of 25 per cent.



For the 3000 Volts connection, so called "Diametrical" connection Windings 2-6 and 8-4 are connected in multiple and 5-1 and 3-7 are connected in multiple so that phases are as follows:—

Phase-1 2-6 and 8-4

Phase-2 5-1 and 3-7

For the 4242 Volts, so called "Square" connection, windings are connected in series as shown

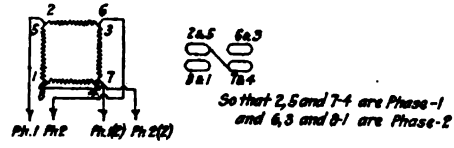


FIG. 170.—Main Generator Connections, U.S.S. New Mexico.

The windings are quarter phase and may be arranged as shown in Fig. 170. Both ends of each independent winding are brought to the terminals of a double throw switch. This switch when in one connection puts the windings in diametral or parallel connection and, with the generator operating at designed field and speed, gives 3000 volts. With the switch in the other connection the windings are thrown into the square or series connection and this arrangement at the designed field strength and speed gives 4242 volts. The windings are arranged in this way to secure the maximum efficiency of each generator when driving four motors when the first connection is used, and also when it is driving two motors and the high voltage connection is made. This change is made by the generator disconnecting switches.

The generator is well ventilated by fans to care for the heavy field currents when over-excited. All windings are mica insulated.

Exciting current comes from the 300 kw. generator but is passed through the booster to obtain the desired range of field control on the main generators.

The general arrangement of the engine room is shown diagrammatically in Fig. 171.

The motors are of special design. They are rated at 7250 H.P. each at 167 R.P.M. They are arranged for pole-changing. The number of poles can be changed from 24 to 36 by a suitable pole changing switch. This gives

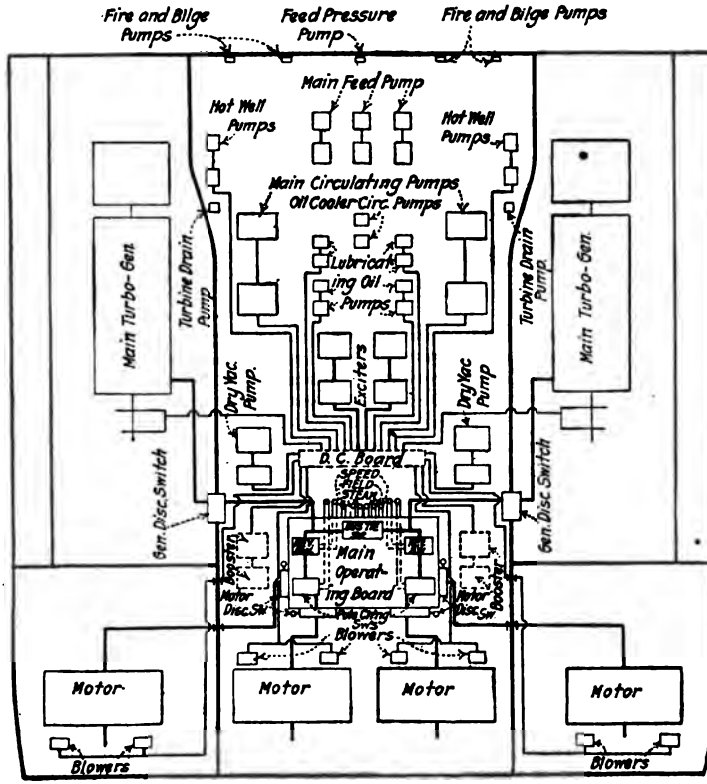


FIG. 171.—General Arrangement of Machinery, U.S.S. New Mexico.

speed reductions of 12 to 1 and of 18 to 1 on the propeller shafts, and enables the turbine to operate at full speed both at 15 knots and at 21 knots. Such operating conditions add materially to the economy of the turbine. The rotor is provided with a double squirrel cage winding. The outer slots are filled with German silver high resistance bars; the inner slots with low resistance copper bars. Each squirrel cage has the end of its bars connected by short-circuiting rings. The object of this construction is to provide a winding which will give with the simplest construction the necessary torque for reversing, and still retain the advantages of the simplicity of a squirrel-cage

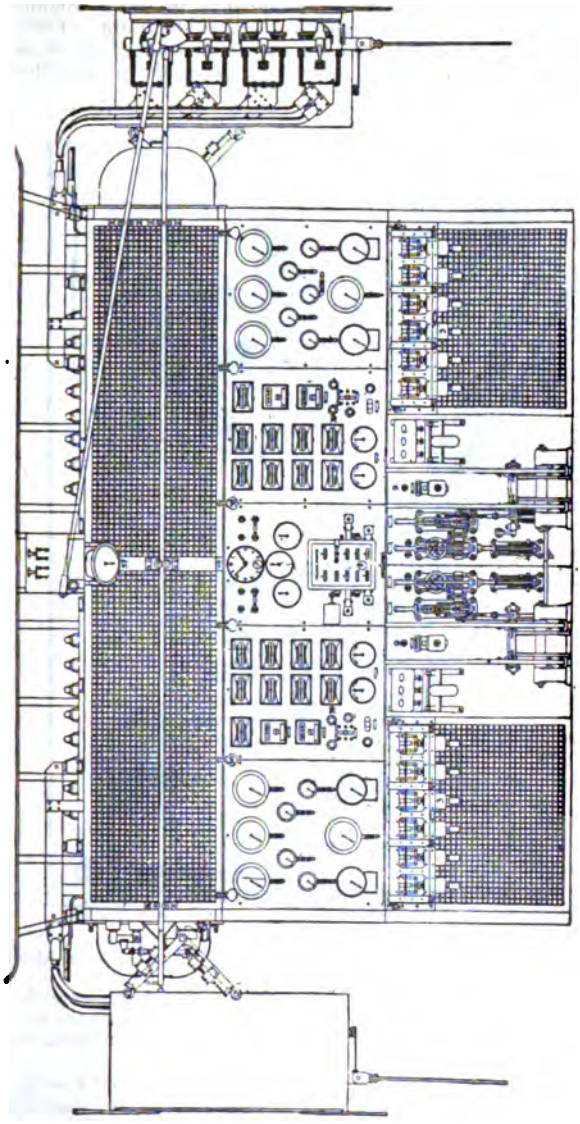


FIG. 172.—Main Switchboard, U.S.S. New Mexico.

motor. Furthermore, no changes are required in the rotor winding when shifting from 36 poles to 24 poles, and vice versa in the stator. The reversal of the motor is accomplished by crossing two of the leads of one phase of the motor and this is done by reversing switches located in the centre of the engine room.

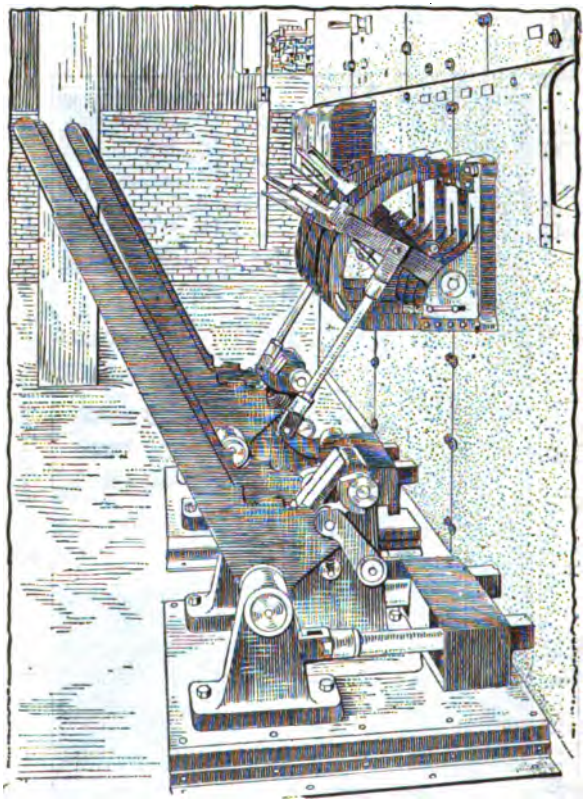


FIG. 173.—Control Levers, Main Switchboard U.S.S. New Mexico.

The motors are ventilated by suction blowers located one on each side of each motor on top of its casing. These fans draw air through the motor and discharge it into ducts leading to the deck, where the hot air from the main generators is also discharged.

Double buses are furnished for exciting current on the exciter switchboard. The usual field switch for the turbo-generators and the accompanying discharge resistances are placed on the exciter switchboard. The field switch is solenoid operated and is provided with a magnetic blow-out.

The main switchboard is a cell structure and a front view is shown in Fig. 172. The operating levers are all placed on the front of the board while oil switches, interlocking gear for the same, etc., are placed behind the board and enclosed by grill-work.

A partial view of the switchboard, showing operating levers, is shown in Fig. 173.

The disconnecting switches from the main generators are mounted on the outboard bulkheads and are of the knife-blade type. The motor disconnecting switches are mounted on top of the cell structure at the rear of the switchboard.

Reversing switches are provided to change two leads of one phase when reversing. Pole changing switches are also furnished to change from 24 to 36 poles. All these switches are interlocked with each other so that the reversing lever cannot be thrown into the astern position except when the pole changer is in the 36 pole position.

The speed of the turbo-generator and motors is varied by means of a worm and worm-wheel on the main switchboard which varies the governor links by means of the bell-cranks and rods. The field is varied automatically with the speed.

Steam limit levers are also mounted on the switchboard. These levers can be set to allow any desired number of nozzle valves to open on the main turbine and when once set, the amount of power that can be developed by the turbine is limited by the amount of steam that the given number of valves will pass. This prevents overloading of the main turbine.

Various types of relays and the usual indicating instruments are also mounted on the switchboard.

The New Mexico is operated from slow speed up to 15 knots by using one generator and the motors connected for 36 pole connection; from 15 to 17 knots, one generator is used with motors on the 24 pole connection; from 17 knots to full power both generators are used and motors are arranged for the 24 pole connection.

Data from the trials of the New Mexico are given on p. 721.

U. S. S. Tennessee

The U. S. S. Tennessee is another battleship being fitted with electric drive, although not yet in service. An account of its equipment appeared in "Power," Aug. 7, 1917.

The ship will have a displacement of 32,600 tons and a rated full speed of 21 knots. Its propelling machinery will consist of four motors, one on each propeller shaft. Current for the motors will be furnished by two standard Westinghouse turbo-alternators each of 10,000 kw. capacity. The steam turbines are of the impulse-reaction semi-double flow type. The H.P. steam is first expanded in suitable nozzles and passes through a two row velocity stage. It then passes through the first reaction stages which are single flow. This type is shown in Fig. 197.

The turbine speeds will be controlled from the engine room by a mechanically operated throttle valve and hand or automatic operated emergency valve. Or the speed can be controlled from the operating room by automatic governor control operated by an adjustable speed control valve while the emergency valve may be operated by pull wires.

The alternating current motors have rotors wound for 3400 volts and have two independent windings for 24 and 36 poles. The rotor windings will have a three-phase connection to three slip rings and will be arranged so that with the 24 pole connection the motor will act as an ordinary wound-rotor

machine and by means of a suitable cross-connection it will act as a squirrel cage motor with a 36 pole connection. The motors will be separately ventilated.

The motors will have a normal capacity of 6700 H.P. at 175 R.P.M. on the 24 pole connection and 2125 H.P. at 118.3 R.P.M. on the 36 pole connection and an overload capacity of 25 per cent or 8375 H.P. for four hours at a maximum speed of 185 R.P.M.

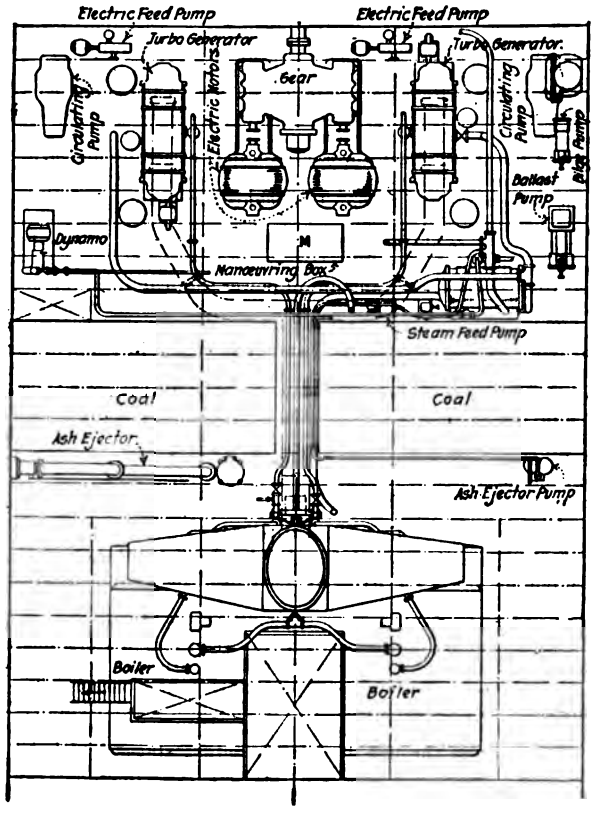


FIG. 174.—General Arrangement of Machinery, S.S. Mjölner.

Up to 118.3 R.P.M. on the motors all the power will be furnished by one turbo-alternator and the speed will be varied by altering the speed of the turbine. Above this speed the motors will run on a 24 pole connection.

The direction of rotation is changed by reversing switches. Liquid rheostats will be used for starting the motors. All switches and rheostats

will be operated by levers mounted on the front of the instrument boards. All switch levers will be interlocked.

Six 300 kw. direct current turbo-generator sets will be installed, two for excitation and four for general service. The excitation sets are non-condensing and exhaust into late stages of the main turbines. Exciting current is passed through boosters as on the New Mexico, p. 697.

The guaranteed steam consumption of the main generators and auxiliaries is as follows:

Speed in knots	R.P.M. propellers	S.H.P. total	Steam consumption lbs. per S.H.P.-hr.
21	175.0	26,800	11.90
19	154.4	18,850	11.65
15	118.3	8,500	12.10
10	81.0	2,600	15.45

S. S. Mjölner

One of the first applications of the electric drive to merchant ships was in the S. S. Mjölner, built by the Svenska Turbinenfabriks Aktiebologet Ljungström, Finspong, Sweden. Two sister ships were built, the Mjölner with turbo-generators and electric drive and the Mimer with triple expansion engines, each of about 1100 tons dead weight capacity built for coast-wise trade. Each is 225 ft. long, 36 ft. beam and 14 ft. 4 in. draft. The Mjölner had 900 S.H.P. installed and the Mimer 1000 I.H.P. corresponding to 860 S.H.P.

Fig. 174 shows a plan of the engine room of the Mjölner.

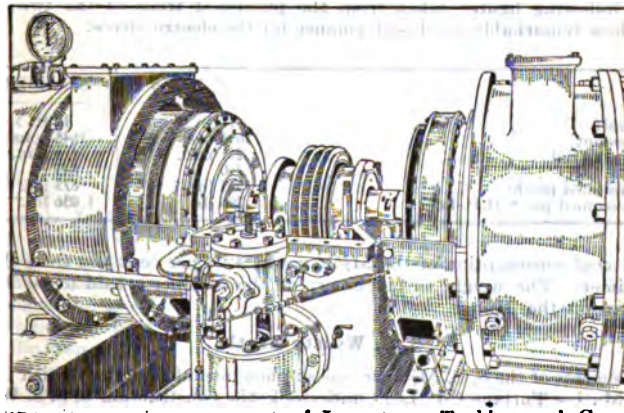


FIG. 175.—General Arrangement of Lungstrom Turbine and Generators.

Two turbo-generator sets were installed, each of 400 kw. capacity at 7200 R.P.M., providing 3 phase current at 500 volts. Each set consisted of a Ljungström type of reaction turbine, each half of which rotates in the opposite direction and drives an alternator. These two alternators are tied together electrically and function as one unit. The whole set, turbine and alternators, is mounted directly on top of the condenser and supported by it. The hot air from the generators is delivered by ducts to the Howden system in the boiler

room. Fig. 175 shows one complete set with the upper half of the turbine casing removed. The steam supply is controlled by the turbine governor operating through a compensated oil relay. Superheated steam is furnished these turbines.

The two motors are of the usual commercial type of three-phase induction motors with slip rings running at 900 R.P.M. They are geared by helical gearing to a single propeller shaft. The pinions are of chrome nickel steel and the gear of carbon steel. This arrangement permits the use of standard low priced high speed induction motors instead of specially built high priced less efficient slow speed motors.

Specially designed slip couplings connect the motors to the pinions so that if the propeller strikes ice the coupling will slip and prevent damage to propeller or gear.

The switching gear is very simple and is enclosed in a single cabinet.

In the American electric drive installations previously described, the motor speed is controlled by altering the main turbo-alternator speed and also by pole changing. This Swedish installation does none of this but depends on resistances of the liquid type in series with the motor feeders to regulate the current in the motor windings. This resistance consists of one cone terminal which can be lowered by means of the control hand wheel into another cone terminal which is filled with liquid. In the stop position the upper cone is out of contact with the liquid. In the full speed position the two cones are in metallic contact. Intermediate speeds are controlled by the varying distance between cones.

All maneuvering is done by a single hand wheel mounted on the switchboard which forms the front of the cabinet.

The following figures taken from the published trials of the two sister ships show remarkably good performance for the electric drive:

	S. S. Mimer	S. S. Mjölner
Mean draft	11 ft. 9 in.	11 ft. 5½ in.
Displacement	1696 tons	1648 tons
Duration of trial	251 min.	176 min.
S.H.P.	730	843
Coal consumed per hr.	1313 lbs.	873 lbs.
Coal consumed per S.H.P.-hr.	1.798 lbs.	1.036 lbs.

The coal consumption of the Mjölner was 42.3 per cent less than that of the Mimer. The weight of machinery on the Mjölner is said to be 50 tons less than on the Mimer.

S. S. Wulsty Castle

The success of the S. S. Mjölner was so phenomenal that the British Ljungstrom Marine Turbine Co., Ltd., undertook the construction of S. S. Wulsty Castle, a cargo vessel. This ship is fully described in Engineering, May 3, July 12, 26 and Aug. 9, 1918. It is 356 ft. 3 in. long and 48 ft. 9 in. breadth with a molded depth of 25 ft. 11 in.

Superheated steam at 220 lbs. gage and 625 deg. F. total temperature is furnished to two standard constant speed Ljungstrom turbo-alternators of 625 kw. each which deliver 3 phase current at 650 volts when running at 3600 R.P.M. As in the Mjölner, these turbo-alternator sets are carried by spring supports on the top of their own condensers.

The power is delivered to two induction motors each of 785 H.P. capacity. The performance curves of these motors is shown in Fig. 176.

The efficiency of these motors from these curves is 95 per cent at full load with a power factor of 87.5 per cent. The two motors are connected through double helical gearing to a single propeller shaft. The normal speed of the motors is 714 R.P.M. and of the propeller 76 R.P.M. A specially designed flexible coupling connects the motor to the pinion shaft. The pinions are 9.2 in. diameter at the pitch line and have 23 teeth. The gear is 86 in. in diam. and has a ratio of reduction of 9.4 to 1. The effective width of face is 22 in.

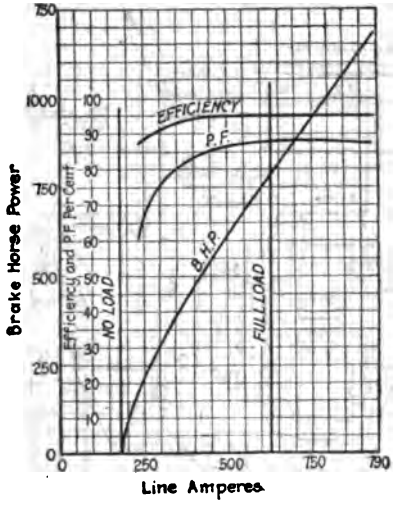


FIG. 176.—Characteristic Curves of Motors, S.S. Wulsty Castle.

As in the Mjölner the speed of the motors is regulated by resistances in the circuit and not by varying the speed of the main turbine. The current is controlled by liquid resistances, the electrolyte being a solution of potassium hydrate. The electrodes have the form of cones. The lower cones are stationary and are filled with the solution, while the upper ones are mounted on spindles which are raised or lowered by the maneuvering hand wheel.

In the stop position the cones are lifted entirely free of the solution. The hand wheel also operates the reversing switches of the motors when first moved either side of its mid position. The maneuvering wheel can be locked firm in any position. The electrolyte is circulated by a motor driven pump in the base and is cooled by the coolers provided. This arrangement is said to give very satisfactory speed regulation. The general arrangement of machinery in the Wulsty Castle is shown in Fig. 177.

The turbines and electrical equipment for this ship was furnished by Brush Electrical Engineering Co., Loughborough, England.

General Notes

In a recent copy of Marine Engineering, W. L. R. Emmet points out that careful tests on a 2500 H.P. double reduction gear set indicates a mechanical

efficiency of 94.5 per cent. This gearing is further handicapped by the friction losses of the reversing turbine sections in the main turbine. He states that with high speed turbines and the most improved electrical apparatus, a transmission efficiency of 93.5 per cent can be secured. Hence with high speed turbines in both types of merchant ships the difference in efficiency will not exceed 1 per cent.

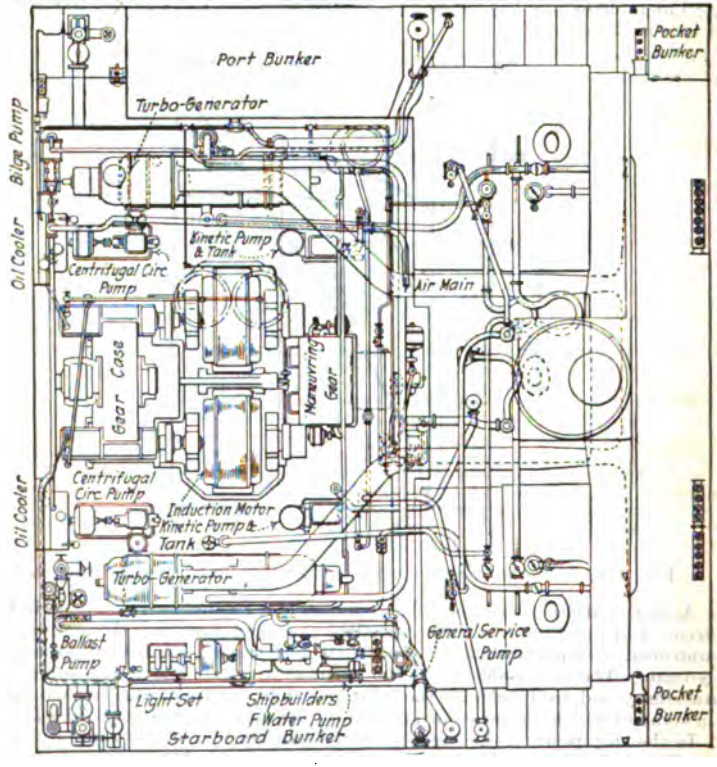


FIG. 177.—Arrangement of Engine Room, S.S. Wulsty Castle.

In the same article the heat troubles encountered in reversing impulse types of turbines are discussed. The temperatures due to friction become so high that the forward blades become blue. Local heating and irregular heat distribution may thus become a source of danger in reversing turbines, particularly if superheat is used. These troubles are overcome when the electric drive is used. It is stated that a guarantee of 10 lbs. of steam per S.H.P.-hr. on the main turbine alone could be made on a 4000 H.P. merchant vessel

to be equipped with electric drive, at a propeller speed of 110 R.P.M., the steam conditions to be 265 lbs. gage at throttle, 150 deg. superheat and 28.5 in. vacuum. The water rate per shaft horsepower, including auxiliaries, is estimated to be 11.1 lbs. per shaft horsepower hour. It is proposed to drive all auxiliaries by motors, furnishing current by either of two 150 kw. direct current sets which exhaust either into the feed water heater or into one of the lower stages of the main turbine.

The electric drive has proved quite satisfactory for naval vessels where slow speed motors are direct connected to the propeller shafts and operate a considerable amount of the time at cruising speed.

In merchant service the standard type of induction motor has been geared to the slow speed propeller as described above.

As experience accumulates, systems of electric drive will be modified and improved. The system has certain advantages over the mechanical gearing. Against these must be placed the increased cost of such a plant.

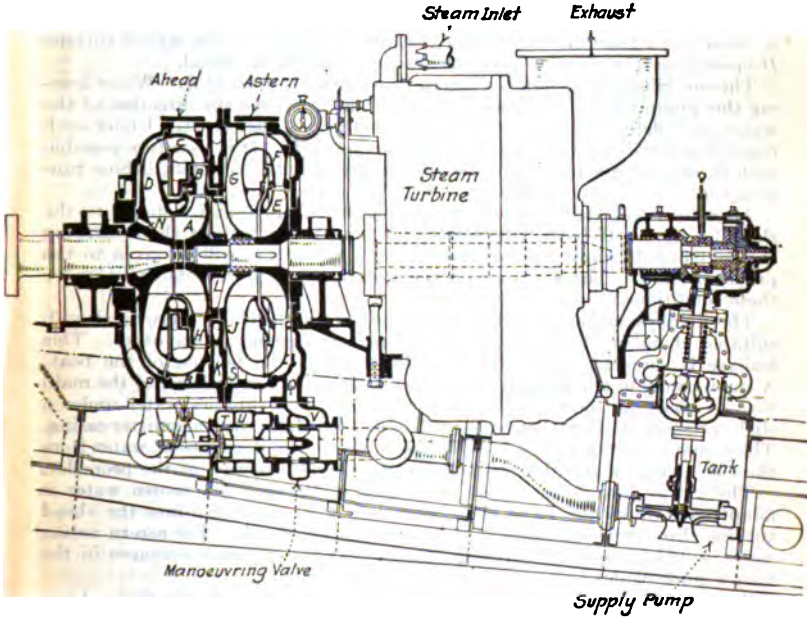


FIG. 178.—Föttinger Hydraulic Transmission Gear.

HYDRAULIC GEARING

A third means of reducing the high speed of the main turbine to that desired on the propellor is by means of hydraulic gearing. The Föttinger hydraulic transmission has been the only one of this kind developed. Fundamentally it consists of a centrifugal pump direct connected to the main shaft of a high speed steam turbine and delivering water to a water turbine direct connected to the propeller shaft in the same axial plane. By arrangement of guide

vanes and buckets it is possible to have a reversing water-turbine transmitter of somewhat similar design.

The Föttinger transmitter, attached to a Curtis turbine, is shown in Fig. 178. It consists of two chambers of cast iron, the after one for driving the propeller shaft ahead at reduced speed, and the right hand one for driving it astern also at reduced speed. The ahead transmitter shown on the left consists of a centrifugal pump marked *A* on the (primary) turbine shaft; this delivers water directly into the first water turbine *B* on the (secondary) propeller shaft where part of the energy of the water is absorbed; the water leaving *B* passes through stationary guide blades *C* fastened to the casing which deliver it to the second (secondary) turbine wheel *D* also fastened to the slow speed shaft; the water leaving the second turbine still has some velocity and is carried directly into the intake of the centrifugal pump *A*. The water thus flows in a compact body through a closed circuit.

The centrifugal pump has the usual impellers with backward slope. The blades on the first water turbine resemble those on an impulse wheel and absorb principally the velocity of the water. The guide blades on *C* are similar to those on a reaction water wheel, while the blades on the second turbine *D* operate on the reaction principle and are practically radial.

The centrifugal pump on the astern transmitter is marked *E*. Water leaving this pump passes into guide blades *F*, which reverse the direction of the water and deliver it to the turbine *G* on the propeller shaft. Under such conditions *G* will turn in the opposite direction to *E*. It is therefore possible with these arrangements to reverse the propeller with the main turbine running continuously in one direction.

All wheels and blades are of bronze, highly polished, and fastened to the shaft by keys. If both transmitters are empty the turbine shaft carrying the centrifugal pumps will revolve idly and no motion will be given to the propeller shaft. The governor on the turbine must control its speed under these conditions.

The diaphragm between the ahead and astern elements is provided with suitable chambers to catch leakage from one element to the other. This leakage is carried back to the receiving tank in the bottom of the boat. A small centrifugal pump, driven by gears from the forward end of the main turbine shaft as shown, is placed in this tank and supplies water under a slight pressure to the maneuvering valves located under the transmitter casing. These maneuvering valves control the admission and release of water from the two elements and thus control the speed and direction of the propellor.

The valves are of the piston type, and in the position shown water is admitted through the valve chamber *V* into the port *P* and into the ahead turbine through the passage *N* in the second turbine *B*. For astern action water would be admitted through the port *Q* and through passages in the hollow guide blades *F*.

At the same time that water is being admitted to *V* or *K*, the drain chambers *T* or *U* are opened and water drained from the other element through the drain passages *R* or *S*. The valves are so arranged that the drain chamber is fully closed off when the corresponding admission chamber is full open. In maneuvering the steam turbine is kept running but the maneuvering valve only admits a small amount of water to the transmitter which is then only partially filled with water and hence the water turbines and propeller shaft run at reduced speeds. Either the forward or astern element can be put in use or can be drained by the manipulation of the maneuvering valves.

The Föttinger transmitter permits a very compact engine room layout.

It takes less room than the ordinary reduction gear. Its straight axial drive and compactness are similar to the Poole Turbo-gear though not nearly so efficient. No astern turbine section need be provided in the main unit. Thus the weight and first cost of the turbine are reduced and its efficiency is slightly increased as the whirling losses of the idle astern blading is obviated. As the astern element of the Föttinger gear is somewhat simpler and less efficient than the forward element, the reverse power is only about 85 per cent of the forward power.

This gear is a combination of a high efficiency centrifugal pump and a water turbine. There are thus certain hydro-dynamic losses that cannot be avoided. Hence, the efficiency of the Föttinger transmitter is much lower than in the geared transmission described on p. 818.

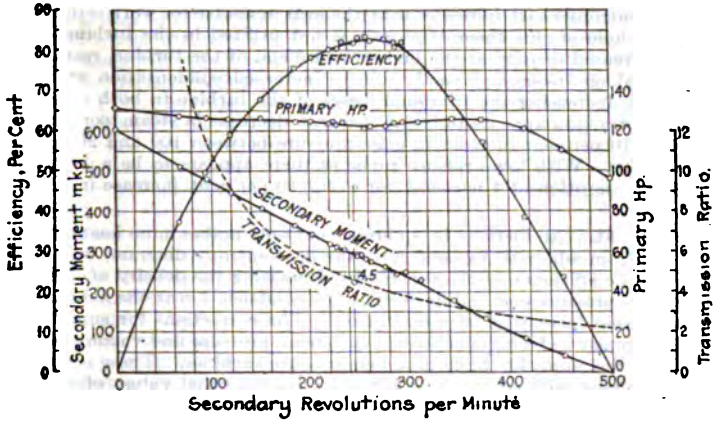


FIG. 179.—Tests of Föttinger Hydraulic Reduction Gearing.

Tests in Engineering, Nov. 5, 1909, shown in Fig. 179, indicate on a small set a maximum efficiency of 83 per cent with a speed ratio of 4.5 to 1. The efficiency decreases rapidly as this ratio is increased, indicating very poor steam economy for the set at slow speeds.

In Engineering, April 7, 1912, Dr. Föttinger is reported as saying that 85 to 90 per cent efficiency had been obtained on an 800 H.P. gas engine driven boat in reducing the speed from 270 to 100 R.P.M.

Engineering, Dec. 13, 1912 contains a description of a 10,000 H.P. Föttinger transmission for a German ship to reduce the speed from 850 to 170 R.P.M. An efficiency of 88 to 90 per cent was guaranteed and this was said to have been met on test.

It will be apparent from these figures that the efficiency of these devices does not compare with that of the toothed reduction gear transmission. However, Dr. Föttinger proposes to use the feed water in the gear before supplying it to the feed and filter tank. The friction losses in the form of heat will thus be largely recovered and the coal consumption of the ship decreased. He therefore claims a thermal efficiency of 97 to 98 per cent for his hydraulic transmitter. However, the hydraulic system has never been adopted outside of Germany and will probably only be used for special purposes.

PERFORMANCE

Effects of Variation in Operating Conditions

All corrections for pressure, vacuum or superheat are subject to variation on different turbines depending on the particular conditions for which the turbine was designed to run and also on its ratio of blade speed to steam speed in operation. No absolutely fixed rules for such correction can be stated which will apply to every case. All corrections therefore must be agreed upon by manufacturer and purchaser, before tests are made. However in order to enable engineers to compare turbines operating under conditions other than the assumed standard conditions, the following corrections, based on average performance, are submitted.

Pressure. Obviously as the pressure is increased adiabatic expansion on the heat-entropy chart indicates that the heat available for work is increased. But the whole of this theoretical gain is not utilized in the turbine, partly due to increased density at the high-pressure end of the turbine, resulting in greater leakage losses and partly due to increased condensation at the exhaust end, increasing the frictional losses of the turbine in both cases. An average correction appears to be 1.5 per cent decrease in steam consumption for every 10 per cent increase in gage pressure between 100 and 200 lbs. per sq. in. gage. With low-pressure turbines there appears to be a decrease of steam consumption of 4 per cent for every 10 per cent increase in absolute pressure.

Superheat. An increase in superheat not only makes more heat available for work from adiabatic expansion on the heat-entropy diagram, but it also increases the efficiency of the turbine by decreasing the density of the steam at the high pressure end and decreasing the condensation at the exhaust end, in both cases decreasing frictional losses. The corrections for superheat in practice appear to be independent of the steam pressure and vacuum and are therefore the same for high- and low-pressure turbines. These corrections vary somewhat with type. The following are the usual values of such corrections: 1 per cent decrease in steam consumption for every 12 deg. F. increase in superheat between 0 deg. and 100 deg. F. superheat; 1 per cent decrease in steam consumption for every 14 deg. F. increase in superheat between 100 deg. and 200 deg. F. and 1 per cent decrease in steam consumption for every 16 deg. increase in temperature from 200 deg. to 300 deg. F. Marine turbine practice does not usually exceed 100 deg. F. superheat so that corrections above 200 deg. F. are seldom used.

Moisture. The usual correction for moisture is to multiply the total condensate by the per cent moisture. A study of adiabatic expansion on the heat-entropy chart will indicate that this is not sufficient, for an increase of moisture means a decrease of heat available for work. Furthermore it increases the wetness of the steam in all stages of the turbine and thereby increases friction losses and decreases the efficiency. No data are available of this correction factor by actual tests. It has been found that the efficiency decreases approximately 1 per cent for every 1 per cent increase in wetness with nearly dry steam. Hence the steam consumption will increase 2 per cent for every added 1 per cent of moisture under such conditions. This correction emphasizes the necessity of absolutely dry steam for turbines without superheat.

Vacuum. Corrections for vacuum are considerably influenced by the initial pressure. It is apparent that such will be the case by a glance at the heat-entropy chart, taking expansions to vacuum from high pressures and

from about atmospheric pressure. The gain from increased vacuum will be a much larger percentage in the latter case than in the former where high pressure is used.

Vacuum corrections naturally vary with the initial conditions of pressure and superheat and also with the type of turbine, *i.e.*, whether the turbine is properly designed for high vacuum or not. The following corrections represent average practice for pressures of 100 to 200 lbs. per sq. in. gage and 0 deg. to 200 deg. F. superheat when turbines are properly designed: 4 per cent decrease in steam consumption for 1 in. increase in vacuum between 26 and 27 in. vacuum; 5 per cent decrease in steam consumption for 1 in. increase in vacuum between 27 and 28 in. vacuum; 7 per cent decrease in steam consumption for 1 in. increase in vacuum between 28 and 29 in. vacuum; some designs of turbines show much greater improvements than indicated above, others less.

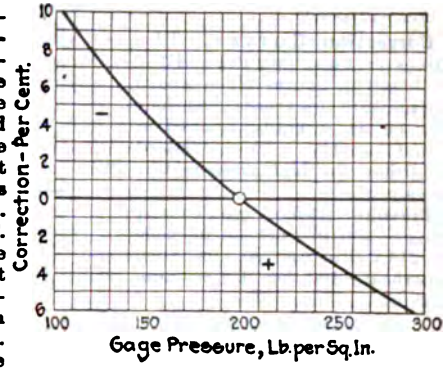


FIG. 180.—Correction Curve for Variations in Initial Steam Pressure.

For low-pressure turbines Baumann gives the following corrections for pressures from 14–18 lbs. per sq. in. absolute and 0–100 deg. F. superheat: 11.5 per cent decrease in steam consumption for 1 in. increase in vacuum between 26 in. and 27 in. vacuum; 13 per cent decrease in steam consumption for 1 in. increase in vacuum between 27 in. and 28 in. vacuum; 14.5 per cent decrease in steam consumption for 1 in. increase in vacuum between 28 in. and 29 in. vacuum.

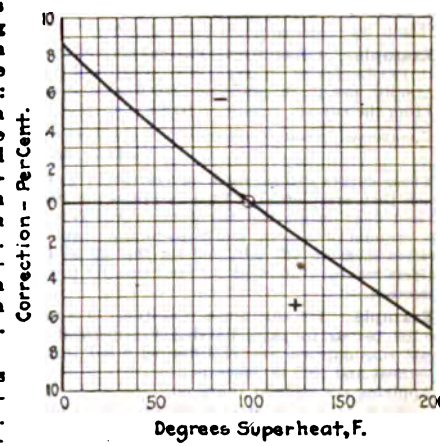


FIG. 181.—Correction Curve for Variations in Initial Superheat.

Speed. In marine turbines the horsepower output increases rapidly with the speed. The steam consumption decreases with increase of load. No definite rules can be stated for the relation of steam consumption and speed and reference must be made to performance curves given later.

In general, the turbine blading will be designed for best efficiency at about

rated load, and at any other load the ratio of blade speed to steam speed will be altered and the efficiency decreased. One can naturally expect therefore increased steam consumption at decreased speeds.

Correction Curves. The curves in Figs. 180, 181 and 182 are based on figures given above and indicate how these corrections can be most easily applied in actual practice. The standard conditions for high-pressure turbines are:

Pressure (before governor valve)—200 lbs. per sq. in. gage or 214.7 lbs. per sq. in. abs.; superheat—100 deg. F.; vacuum —28 in.

To use these curves the corrections are read from each curve and the algebraic sum gives the correction to be applied to the water rate. Thus, let w_t = steam consumption under test conditions and w_g = steam consumption corrected to standard or guaranteed conditions and let C_p, C_s and C_v be the corrections for pressure superheat and vacuum. Then $w_g = (1 + C_p + C_s + C_v)w_t$.

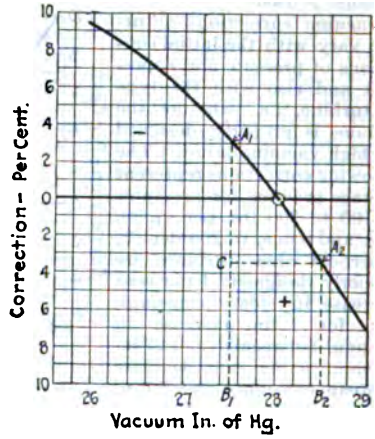


FIG. 182.—Correction Curve for Variations in Vacuum.

Example. A 3000 Curtis Parsons turbine under test required 11.1 lbs. per S.H.P.-hr. with 190 lbs. per sq. in. gage press., 120 deg. F. superheat and 28.7 in. vacuum. Correct to 200 lbs. per sq. in. abs. gage, 100 deg. F. superheat and 28 in. vacuum.

From the curves, $C_p = -0.008$ $C_s = +0.015$ $C_v = +0.049$

$\therefore$ corrected steam consumption $= 11.1(1 - 0.008 + 0.015 + 0.049)$

$= 11.72$ lbs. per S.H.P.-hr.

These curves can be used for correction to conditions other than the standard for which they are plotted. Thus, suppose the standard vacuum was 28½ instead of 28 in. and that during test the vacuum was 27½ in. Let A_1B_1 represent a line through test vacuum of 27.5 in. and A_2B_2 be line through standard vacuum of 28.5 in. With dividers measure off A_2B_2 and set this off as B_1C from B_1 on line A_1B_1. Then with dividers measure intercept A_1C and transfer this to the correction scale and read off the correction.

Example. The guarantee conditions of a marine geared set of 2500 S.H.P.-hr. are 200 lbs. per sq. in. gage, 100 deg. F. superheat and 27.5 in. vacuum. On trial trip the steam consumption was 11.3 lbs. per S.H.P.-hr. with steam at 190 lbs. gage, 90 deg. F. superheat and 28.5 in. vacuum.

Applying the method above outlined,

Steam consumption at guaranteed conditions,

$$w_g = (1 - 0.008 - 0.008 + 0.065) \times 11.3 = 11.85 \text{ lbs. per S.H.P.-hr.}$$

Turbine Testing

The testing of marine steam turbines presents many problems not encountered on turbines for land purposes. The latter usually operate at fixed speeds and may be tested by water brakes or electric generators. Even if the speed is variable the load on the brakes can be adjusted to any desired amount at any speed and in general the method of test is simple.

Marine steam turbines operate generally at variable speeds. Even those which employ the electric drive operate at variable speed. The propeller acts as a brake on the unit and it is not easy to reproduce on the test floor the exact load of the propeller at various speeds of the main shaft. This, however, has been attempted and much valuable data has been derived from such tests. In the case of the marine electric drive the electric power input to the motors can be calculated, and the shaft horsepower can be calculated after allowing for motor losses. Where tests are to be carried out on shipboard with direct-connected or geared sets, some form of **torsion dynamometer**, as described in Section 14 of this book may be used.

The Westinghouse **floating frame reduction gear** can also be **used as a dynamometer**. When oil pistons are used to support the pinion frame, the pressure of the oil on these pistons is an exact measure of the force that is being applied at that instant at the pitch line of the gears. The power transmitted can be found by multiplying the pressure gage reading by the area of the supporting pistons and by the pitch line speed of the gear in ft. per min. and dividing by 33,000. Only one gage indicates pressure in either forward or reverse operation. Recording pressure gages and speed recorders give a continuous record of power transmitted with such a gear.

The following **observations** should be taken **during a test:**

The steam pressure before the control valve read from an accurate gage should be recorded during the test. The temperature of the steam if superheated, or the quality if wet should also be taken. The pressure beyond the control valve and before the first nozzles or the first blades in a Parsons unit should be read from a pressure gage.

Some turbines are fitted up so that steam pressures at the various stages may be read. This information is useful for checking design and also to check operation from time to time, but is not absolutely essential.

If the unit has two cylinders the receiver pressure between these should be accurately read on a compound pressure gage.

The vacuum should be read on a mercury column, for vacuum dial gages are, in general, unreliable. The height of the mercury column taken together with the barometer gives an accurate measure of the absolute pressure in the exhaust. It is also advisable to record the temperature of the exhaust and the temperature of the air pump suction. Excessive air leakage can be detected by a comparison of these readings, the greater the difference, the larger the air leakage.

The pressure drop through the condenser should also be read on a mercury column. In good marine practice this should not exceed 0.1 in. of mercury. If it exceeds this it may be necessary to increase the steam passages in the condenser.

All gages should be carefully calibrated against a standard gage or by a gage tester both before and after use. Care must also be taken to see that the reading of the gage is corrected for any water column that may exist above or below the gage.

In cases of reversal all steam gages on the "forward" section of the turbine will be subjected to vacuum and unless they are of the compound type the needle of each will be forced against the stop pin. This action will cause the gage to indicate incorrectly in a short time. The same condition holds for the astern gages when running ahead. Hence it is good marine practice to insist that all gages beyond the main control valves shall be of the compound type. This is required on all vessels of the Emergency Fleet Corporation.

Thermometers should be immersed as deeply as possible in thermometer wells filled either with cylinder oil, mercury or soft solder if the temperature

is high. These wells should be heavily ribbed inside the steam pipe and should have a thin neck connection to the head of the well. This reduces the loss due to conduction to the steam pipe.

The amount of lubricating oil circulated may be estimated by counting the strokes of the oil pump.

Speed is usually measured on the propeller shaft. The turbine speed can be found when the gear ratio is known. Various speed-measuring devices are shown in Section 14.

The measurement of condensate is rather a difficult matter at sea. The best plan is to weigh it. When this cannot be done some form of metering device may be employed between the hot-well pump and the feed and filter tanks. Steam-jet air pumps also complicate water measurement in some cases.

On test floors it is usually quite easy to make all necessary weighings. On some tests it is also desired to know the fuel consumption per shaft horsepower. This is particularly easy to do with oil-fired boilers by metering the oil supplied to the furnaces. (See p. 318.) This measurement gives the overall performance of the main unit and all auxiliaries. Frequently one finds heavy uneconomical direct-acting pumps installed with condensing equipment instead of lighter weight and often more economical turbine-driven auxiliaries.

Readings should be taken on tests every fifteen minutes although shorter intervals may be employed if sufficient help is at hand. Some signal system should be provided to indicate to observers the time for taking readings.

Forms for taking such necessary readings should be ruled up previous to the test and bound in suitable covers to keep them clean and intact during the test. Either board covered books or sheets held on clip-boards are satisfactory for this work.

Small turbine auxiliary sets can be tested according to standard methods in land practice. The readings that must be taken will depend on the type of auxiliary and the purpose of the test. The necessary readings can usually be easily determined by the engineer in charge of the test.

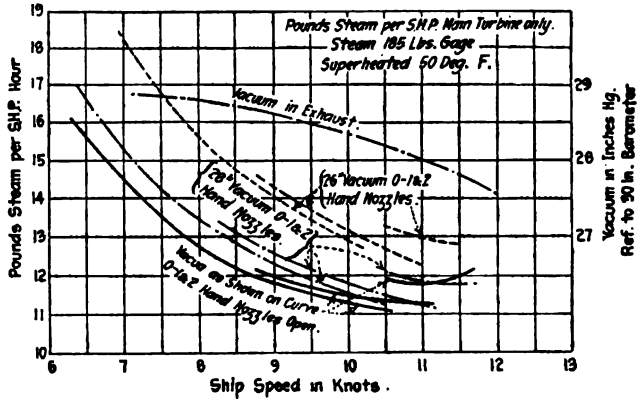


FIG. 183.—Performance Curves, Westinghouse 3000 S.H.P. Cross-compound Marine Turbine.

Test Data and Trials

Very little information is available in regard to performance of marine turbines under varying conditions, particularly in reference to geared sets and to electric drive. Statements of over-all performance appear from time to time but few comprehensive tests have been run of late.

Fig. 183 gives the performance curves of a 3000 S.H.P. cross-compound Westinghouse turbine furnished with double reduction gears for a single screw propeller such as is now being furnished on 8800 ton merchant ships.

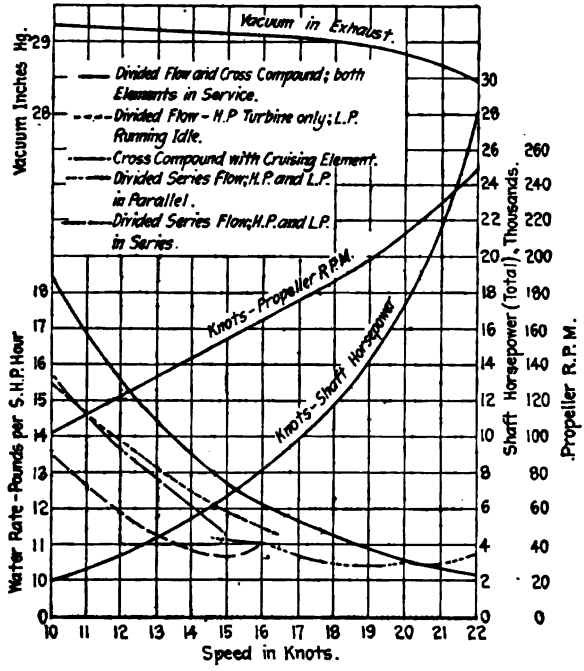

Vacuum in Exhaust.
Divided Flow and Cross Compound; both Elements in Service.
Divided Flow - H.P Turbine only; L.P. Running Idle.
Cross Compound with Cruising Element.
Divided Series Flow; H.P. and L.P. in Parallel.
Divided Series Flow; H.P. and L.P. in Series.
Knots - Propeller R.P.M.
Knots - Shaft Horsepower
Vacuum Inches Hg.
Water Rate - Pounds per S.H.P. Hour
Shaft Horsepower (Total), Thousands.
Propeller R.P.M.
Speed in Knots.

FIG. 184.—Estimated Performance Curves, Westinghouse Marine Turbines with Gears.

Standard conditions are 185 lbs. gage, 50 deg. F. superheat and vacuum as shown. The heavy line gives the actual performance. The light line gives the performance when corrected for a vacuum of 28 in., while the dotted line gives the calculated performance with only 26 in. of vacuum.

Fig. 184 is taken from F. Hodgkinson's paper "Notes on Progress of Turbine Ship Propulsion," Transactions, Society of Naval Architects and Marine Engineers, Vol. 26, 1918. It gives the estimated performance curves of Westinghouse geared turbines with reduction gears under 250 lbs. gage steam

pressure, dry saturated steam and with the vacuum as shown. The performance would be materially improved if superheat had also been used.

These curves refer specifically to a turbine consisting of specially built high and low pressure elements capable of operating on the principle of divided flow, as shown in Fig. 185.

It will be apparent from a study of these two figures that there are appreciable savings possible by operating under the variable conditions indicated. This would not be of much importance in ocean-going merchant ships for these operate all but a small portion of the time at full speed. The scheme would have manifest advantages for naval vessels which run a great portion of the time at cruising speeds.

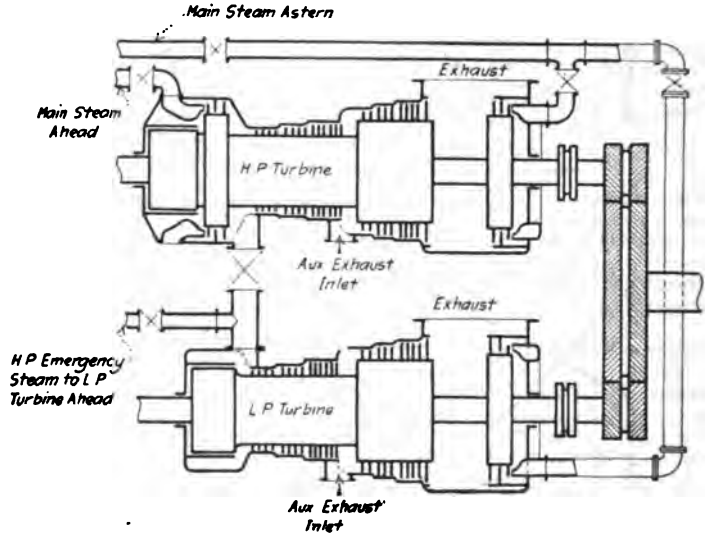


FIG. 185.—Arrangement of Westinghouse Divided Flow Turbines.

The test performance of the 2500 S.H.P. turbines furnished by Poole Engineering & Machine Co. for the U. S. Eagle boats is shown in Fig. 186. The upper curve gives the gear efficiency alone. The gear ratio was 8.4 to 1. The second set of curves gives the steam consumption of the set based on delivered S.H.P. at various speeds under test conditions and when corrected to 250 lbs. per sq. in. and 28.5 in. vacuum. The steam consumption of the turbine alone is also shown. The two lower curves give the relation between speed in knots, R.P.M., of propeller shaft and delivered S.H.P. The unit is apparently at its best economy at full rated load. It should also be noted that the gear efficiency remains high over a wide range of loading.

Mr. W. W. Smith, in a paper in the *Jour. A.S.N.E.*, Aug., 1917, gives the following performance data for two Swedish battleships furnished with Westinghouse geared turbines. The propelling machinery consisted of two

units of 11,000 S.H.P. each. Each unit consisted of two turbines of the Westinghouse divided-flow type driving through a double pinion reduction gear.

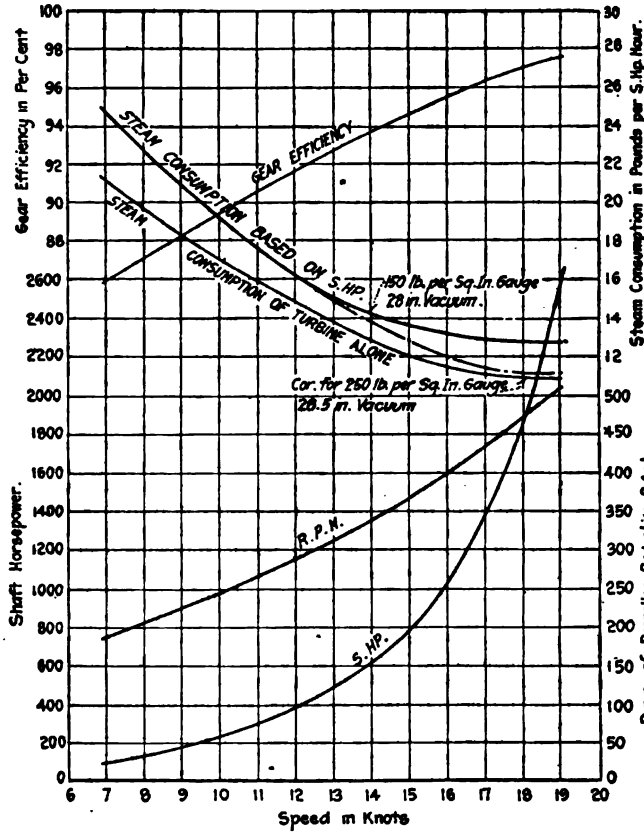

Gear Efficiency in Per Cent
Steam Consumption in Pounds per S.Hp. Hour.
Shaft Horsepower.
R.p.m. of Propeller. Reduction 8.4:1
Speed in Knots
GEAR EFFICIENCY
STEAM CONSUMPTION BASED ON S.HP.
STEAM CONSUMPTION OF TURBINE ALONE
150 lb. per Sq.In. Gauge 28 in. Vacuum.
Cor. for 250 lb. per Sq.In. Gauge 28.5 in. Vacuum
R.P.M.
S.HP.

FIG 186.—Test Curves, Poole Turbines with Gears, U.S. Eagle Boats.

At 22,000 S.H.P. a speed of 22.5 knots was guaranteed. The performance data is as follows:

Speed (approximate) knots	22.5	20.0	12.5
R.P.M. propeller	200.0	177.0	115.0
R.P.M. turbine	3,600.0	3,186.0	2,070.0
S.H.P.	22,000.0	15,000.0	3,150.0
Water rate of turbines, lbs. per S.H.P.-hr. at propeller shaft	10.9	11.1	13.3

The astern turbines will develop 41 per cent of the ahead power with the same flow of steam or 9000 S.H.P. at full power. The operating conditions for the above were as follows:

Steam pressure at turbine	265 lbs. absolute
Quality	Dry and saturated
Vacuum in exhaust	28.27 in.

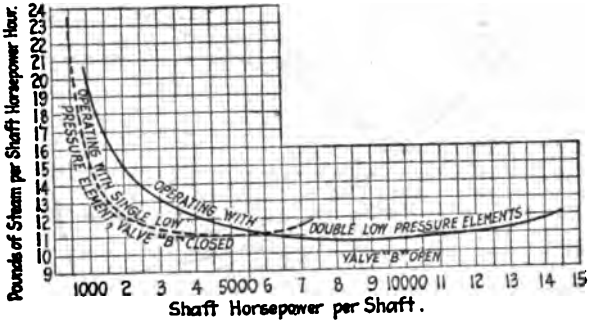


FIG. 187.—Performance Curves of 11,000 S.H.P. Westinghouse Turbines.

The curves in Fig. 187 show the steam consumption guarantees for the main turbines only in lbs. of steam per S.H.P.-hr. at the propeller shaft with the divided-flow principle applied to the turbines. In the same paper the curves in Fig. 188 show the relative performances of reciprocating engines, direct driven and geared turbines for a battleship of 21 knots and 28,000 S.H.P. based on trial data of recent battleships under the following operating conditions: Steam, 250 lbs. gage, dry and saturated; vacuum, 28½ in. The curves show the marked superiority of the high speed turbines with the reduction gears as far as the steam consumption of the main unit alone is consumed.

In another paper on "Geared Turbines for Propelling Machinery," International Marine Engineering, Oct., 1914, the following estimated data is given for the performance of various types of engines alone.

(1) Large freight steamer, speed 14 knots, R.P.M. of propeller 90, S.H.P. 5500, steam pressure 200 lbs. gage:

	Triple expansion engine	Geared turbine
Steam consumption per S.H.P.-hr	16.30	12.10
Coal consumption per S.H.P.-hr	1.63	1.21

Saving in fuel by geared turbines, 25.7 per cent.
Saving in total weight of machinery by geared turbines, 29 per cent.

(2) Slow freight steamer, speed 10 knots, R.P.M. of propeller 70, S.H.P. 2500, steam pressure 180 lbs. gage:

	Triple expansion engine	Geared turbine
Steam consumption per S.H.P.-hr	16.75	12.80
Coal consumption per S.H.P.-hr	1.68	1.28

Saving in fuel by geared turbine, 23 5 per cent.
Saving in total weight of machinery by geared turbine, 27 per cent.

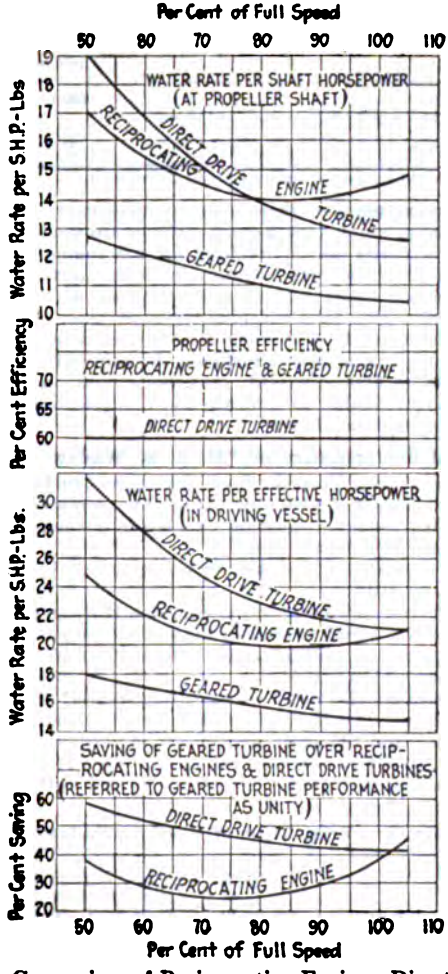

Per Cent of Full Speed
50 60 70 80 90 100 110
Water Rate per S.H.P.-Lbs
WATER RATE PER SHAFT HORSEPOWER (AT PROPELLER SHAFT)
RECIPROCATING ENGINE
DIRECT DRIVE TURBINE
GEARED TURBINE
Per Cent Efficiency
PROPELLER EFFICIENCY
RECIPROCATING ENGINE & GEARED TURBINE
DIRECT DRIVE TURBINE
Water Rate per S.H.P.-Lbs.
WATER RATE PER EFFECTIVE HORSEPOWER (IN DRIVING VESSEL)
DIRECT DRIVE TURBINE
RECIPROCATING ENGINE
GEARED TURBINE
Per Cent Saving
SAVING OF GEARED TURBINE OVER RECIPROCATING ENGINES & DIRECT DRIVE TURBINES (REFERRED TO GEARED TURBINE PERFORMANCE AS UNITY)
DIRECT DRIVE TURBINE
RECIPROCATING ENGINE
Per Cent of Full Speed

FIG. 188.—Comparison of Reciprocating Engines, Direct Driven and Geared Turbines.

(3) Battleship, speed 21 knots, S.H.P. 28,000, steam pressure 250 lbs. gage:

	Direct drive turbine	Geared turbine
R.P.M. propellers	320.0	125
Steam consumption in lbs. per effective H.P.-hr. in driving vessel at full speed (turbine only)	21.2	15
Same at cruising speed	27.8	17

Total saving in weight by geared turbines about 30 per cent.

(4) Destroyer, speed 30 knots, S.H.P. 18,000, steam pressure 250 lbs.

	Direct turbine	Geared turbine
R.P.M. of propellers	600.0	450.0
Steam consumption in lbs. per effective H.P. at full speed (turbine only)	19.5	15.5
Same at cruising speed	28.5	18.0

Total saving in weight by geared turbines about 28 per cent.

In all of these comparisons there is an unquestioned gain in steam consumption of the main units when geared turbines are used. But in practice these comparative results are not realized for the engine room as a whole. Experience shows that the high vacuum condenser auxiliaries and the oil pumps needed on the steam turbine unit, together with circulating water pumps for oil cooler, etc., take considerably more than the auxiliaries of a triple expansion engine with 26 in. vacuum. Hence while there is a considerable gain in weight by using geared turbines, the gain in fuel burned is not so great as would be at first apparent. Complete data on sister ships with both types are not available to indicate the relative overall efficiencies of engine and geared turbine drives.

Trial Performance of "U. S. S. Wadsworth"

The "Wadsworth" is a Torpedo Boat Destroyer 310 ft. long, 30 ft. 6 in. beam, with a displacement of 1050 tons. It has twin screws driven by Parsons turbines fitted with reduction gearing capable of developing 17,500 S.H.P. It was built by Bath Iron Works, Ltd.

The following information is taken from an article by H. B. Gregory, *Jour. A.S.N.E.*, Aug., 1915:

Each shaft is driven through gearing from one H.P. and one L.P. turbine and one astern turbine which is built in the same casing as the L.P. turbine. The H.P. turbine runs at 2494.5 R.P.M., the L.P. at 1509.8 R.P.M. when the propeller is turning 450 R.P.M.

The H.P. turbines are also fitted with cruising stages. The reduction gear is of Parsons design. The H.P. pinion is 11.947 in. diam., at pitch circle, the L.P. 19.738 in. and the shaft gear 66.227 in. with helical teeth of 44°–23.7′ angle and 3.8504 in. diametrical pitch. Fig. 1, p. 1441, gives the relation between speed, R.P.M. of propellor and S.H.P. Table 16 is abstracted from the data on the trial runs.

Table 16. Trial Runs of U. S. S. Wadsworth

Duration of test, hr.	4	4	4	4	4
Date of trial	6/22/15	6/22/15	6/23/15	6/23/15	6/24/15
Speed in knots	30.68	24.99	20.08	16.00	11.99
Boiler pressure gage, lbs.	250.10	248.20	202.00	150.00	151.20
Average vacuum, in. mercury	28.71	29.12	29.15	28.96	28.55
Mean R.P.M. main shafts	459.60	349.60	263.70	208.40	154.60
Total S.H.P.	16,100.00	7448.00	2930.00	1436.00	551.00
Total water consumption per S.H.P.-hr., lbs	12.53	12.72	15.45	21.20	35.92
Fuel oil (19,500 B.t.u. per lb.) per S.H.P.-hr., lbs	0.955	0.919	1.063	1.443	2.536

Electric Drive Performance

U. S. Fleet Collier "Jupiter" has received much attention on account of her electric drive. The boat is 520 ft. long, 65 ft. 2 in. beam, 27 ft. 6 in. draft and with a displacement of 19,230 tons designed for 14 knots. Her engine

equipment consists of a Curtis-Rateau turbine driving a turbo-generator which has a normal output of 5450 kw. at 1990 R.P.M. Alternating current is furnished by the generator at 2300 volts, at normal load to two specially wound induction motors direct connected to the two propeller shafts. Each has a rated capacity of 2750 H.P. at a normal speed of 110 R.P.M. These motors are placed in water-tight pits that cannot easily be flooded. The motors can be controlled by two water cooled rheostats through the switchboard. The following data given in Table 17 are taken from the trial data of the Jupiter.

Table 17. Performance of U. S. Fleet Collier "Jupiter"

Duration of test, hr	48	24
Date of test	2/14/14	2/18/14
Speed in knots	14.99	10.01
Displacement in tons	19,452.00	19,350.00
Boiler pressure gage, lbs	193.80	193.90
Vacuum in. of mercury	28.20	28.50
R.P.M. motors	116.72	77.077
R.P.M. main turbines	2,130.00	1,410.00
Total kw. generated	5,526.00	1,464.00
Total S.H.P.	7,152.00	2,015.00
Water consumption data		
Total lbs. per hr	105,764.00	40,066.00
Lbs. per hr. auxiliaries	22,212.00	15,246.00
turbine	83,552.00	24,820.00
per S.H.P. (turbine only)	11.68	12.316
per S.H.P. (total)	14.78	19.88
Lbs. of coal per S.H.P.-hr	1.662	2.506
Knots per ton of coal	2.82	4.43

Trials of U. S. Battleship "New Mexico"

The U. S. S. New Mexico is fitted with electric drive which is fully described on p. 694. The following data gives its trial performance.

The ship was given two sets of official trials under differing conditions. The water rates which include the steam used by the 300 kw. exciters were as follows for the first trials:

Speed	Lbs. per S.H.P.-hr.
Full speed	12.290
19 knots	11.926
15 knots	11.657
10 knots	14.223

The average trial data for the second set of trials are given in Table 18.

Table 18. Performance of U. S. S. "New Mexico"

	4 hr. full power trial	4 hr. 19 knot trial	4 hr. 15 knot trial	4 hr. 10 knot trial
Steam at boilers, lbs. gage	278.60	274.80	273.80	
Steam at turbines, lbs. gage	272.10	274.20	274.60	277.50
Vacuum	29.00	29.50	29.80	30.00
Barometer	30.83	30.77	30.92	30.79
Feed water temp. deg. F	182.80	189.70	203.40	208.90
R.P.M. main generators	2,042.00	1,825.50	2,012.50	1,450.00
R.P.M. motors	167.69	152.20	115.35	88.49
Slip of propellers, per cent	16.00	14.97	13.24	14.83
Speed in knots	21.68	19.37	14.98	10.26
S.H.P.	31,197.00	23,233.00	9,648.00	3,690.00
Lbs. steam per S.H.P.-hr	12.01	12.33	12.475	13.96

Performance of Small Steam Turbines

Small steam turbines have a wide range of efficiency depending on the care taken in design and in the construction and finish of the machine. Cheap small units are generally less economical than the more expensive types. The geared sets usually show considerable better economy than the direct connected sets with slower speed turbine rotors.

The steam consumption of small turbines is dependent on several factors. An increase in steam pressure results in a decrease in steam consumption. Superheat also has a beneficial effect but not to the same extent as in condensing turbines. An increase in back pressure in non-condensing units results in an increase in steam consumption.

Small turbines generally have a low ratio of blade speed to steam speed. An increase in the blade speed of the turbine increases this ratio and thus improves the efficiency of the unit and decreases the steam consumption. Some performance curves of various types and sizes of small steam turbines will be found in "Small Steam Turbines" by Geo. A. Orrok, Vol. 31, Trans. A.S.M.E.

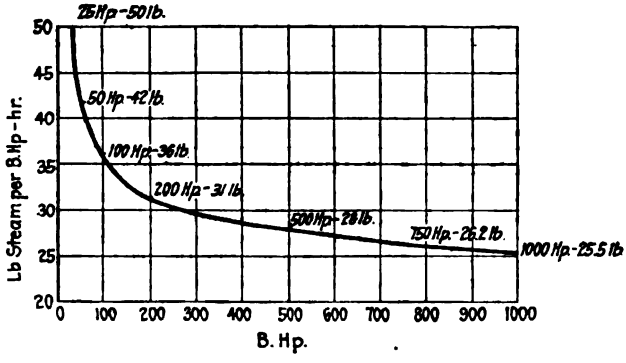


FIG. 189.—Approximate Full Load Steam Consumption, Non-condensing Turbines.

It would be impossible to include performance curves of all types of small turbines in this handbook. Fig. 189 represents the average water rates at normal load of average non-condensing steam turbines operating with 150 lbs. dry steam and no back pressure.

The following corrections may be applied to the figures in the above curves. Corrections for fractional load—increase normal load water rate as follows: ½ load—20 per cent; ¾ load—8 per cent; full load—0 per cent; 1¼ load—5 per cent.

Corrections for initial pressures—175 lbs. gage deduct 3 per cent; 200 lbs. gage deduct 5 per cent; 125 lbs. gage add 5 per cent; 100 lbs. gage add 10 per cent; 75 lbs. gage add 20 per cent.

Corrections for superheat—subtract 1 per cent for each 10 deg. of superheat up to 200 deg.

Corrections for increased back pressures—add for each 1 lb. back pressure the following per cent when the initial pressure is given, 200 lbs. gage—1 per

cent; 175 lbs. gage—1¼ per cent; 150 lbs. gage—1½ per cent; 125 lbs. gage—2 per cent; 100 lbs. gage—2½ per cent; 75 lbs. gage—3 per cent.

High-grade non-condensing turbines may better this performance by a considerable amount, but the above figures form a rational basis for estimates.

The following results were obtained from various tests of turbines of the Westinghouse return flow type. These were selected indiscriminately from manufacturers' tests and customers' witness tests. All tests were run with wet steam containing about 2 per cent moisture.

Rating	Turbine R.P.M	Gage inlet press.	Gage back press.	Water rate lbs. per B.H.P.-hr.	Rankine cycle eff.
10 kw	4000	122	Atm.	48.8	32.0
25 kw	7200	132	Atm.	32.2	47.2
50 kw	7200	139	Atm.	27.7	53.8
100 kw	7200	140	Atm.	27.4	54.2
300 kw	6000	132	3 lb.	28.3	59.5
750 kw	6000	120	Atm.	25.3	62.1
38 H.P	6000	178	10 lbs.	38.0	43.8
52 H.P	6000	167	10 lbs.	36.0	46.8
675 H.P	5600	91	5 lbs.	31.8	64.5

Comparison of Marine Steam Turbines and Other Prime Movers

In comparing steam turbines with other marine prime movers **two classes of service** with quite different requirements must be considered, viz.: (*a*) Merchant service and (*b*) Naval service. Furthermore, no complete comparison can be made without considering the weight of the complete power plant, the space occupied, the fuel or bunker requirements and the performance not only of the engine and its auxiliaries alone but also of the propeller. Space is not available for a complete discussion of all these factors nor is reliable and comprehensive data at hand covering all types of prime movers from which to draw final conclusions. Hence, as this section deals particularly with steam turbines, attention will be directed principally to the combined performance of prime mover and propeller.

The **requirements of the merchant service** are influenced by the necessary operating conditions. Such vessels steam at or near full speed for long voyages. Low speeds are required for comparatively short intervals when maneuvering in harbors. Hence these installations should produce very **high economy at and near full load** while at half speed and below economy is relatively of lesser importance.

The necessary **requirements to secure high propeller efficiency** at full load are low peripheral velocity of propeller blades, narrowness of blade widths, large propeller diameters and slow speed of revolution. This latter requirement is a determining factor in prime mover design for the prime mover must be slow speed to secure maximum propeller efficiency. If the prime mover is more efficient at higher speeds, sometimes the propeller efficiency is sacrificed to some extent by running at the desired higher speed. The **problem of the marine designer** is to secure the best performance of the prime mover at the desired low propeller speed.

The **triple expansion vertical reciprocating engine** was used for this service for many years and was installed in large numbers by the Emergency Fleet Corporation because such engines were available during the war. While well designed reciprocating engines expand steam very efficiently between the limits within which they usually operate, they generally cannot

use superheat nor can they efficiently employ high vacuum. The savings that can be secured by high superheat and high vacuum are such that another more suitable prime mover must be sought.

All the earlier steam turbines were **direct connected** to the propellers and in order to secure the necessary high blade speeds at comparative low propeller speeds, these units were of large diameter. These turbines were large, heavy and not particularly economical. Speeds were increased to improve economy although propeller efficiency was sacrificed by this change. Superheat was seldom employed on account of difficulties from heat expansion in the large turbines.

Turbines with mechanical gears and particularly those with a Curtis stage at the H.P. end can use high superheats and also high vacuums. They can operate at high speeds and hence at high economy. Well designed gears have small losses and hence the propeller efficiency at the slow speed is maintained high. Some difficulties were experienced with the earlier geared units due to improper tooth forms, lubrication methods, insufficient foundation, etc., but these troubles will be gradually overcome as experience accumulates.

Hydraulic gearing and the electric drive are discussed on pp. 692 to 709. These also employ high speed turbines which can use both superheat and high vacuum. Some authorities hold that the final type will be the electric drive. Some remarkably good results have been secured on the Swedish boat referred to on pp. 703 to 704. It is, therefore, apparent that as far as turbines are concerned, the prevailing **type of the future** will be a high speed unit rather than the slow speed direct connected type. The high speed type only has been fully discussed in other parts of this book.

For powers 1500 S.H.P. and above the geared turbine has a higher efficiency than the reciprocating engine particularly when superheat is used. Some performances of turbine installations given on pp. 715 to 723 are available for comparison with reciprocating engine tests.

In the case of **naval vessels**, it is not only necessary to secure good economy at full rated load but also at small fractions of this power which form the usual cruising speeds of these vessels. Probably the **most authorative data** available regarding the performance of various classes of prime movers on naval vessels is contained in a paper by *Rear Admiral C. W. Dyson*, on "*The Passing of the Direct-connected Turbine for the Propulsion of Ships*," *Jour. A.S.N.E.*, August, 1919, from which the following notes are abstracted. This paper outlines the steps which have been taken to date to increase the economy of propulsion of war vessels and also the step common to both naval vessels and merchantmen for the betterment of this economy.

The machinery of certain ships of the U. S. Navy were chosen for comparison, although these machines are not the best in every case of the class of prime mover used. The reciprocating engines chosen have a ratio of expansion of 11, a steam pressure at the throttle of 265 lbs. gage and a vacuum of 27 in. The piston speed at designed power is 1000 ft. per min. The turbines operate with a steam pressure of 250 lbs. gage and 28.5 in. vacuum.

Basic Point of Comparison

In order to compare the efficiencies of the various types of engines and the efficiencies of propulsion of the different combinations of these types as applied to the actual work of propulsion, the designed power of the engines has been taken and is indicated as "100 per cent full power."

On Fig. 190 are shown curves of **water consumption of main engines only** in pounds per hour per shaft horsepower:

A. Reciprocating engines, *Delaware* type, with dry steam at throttle.

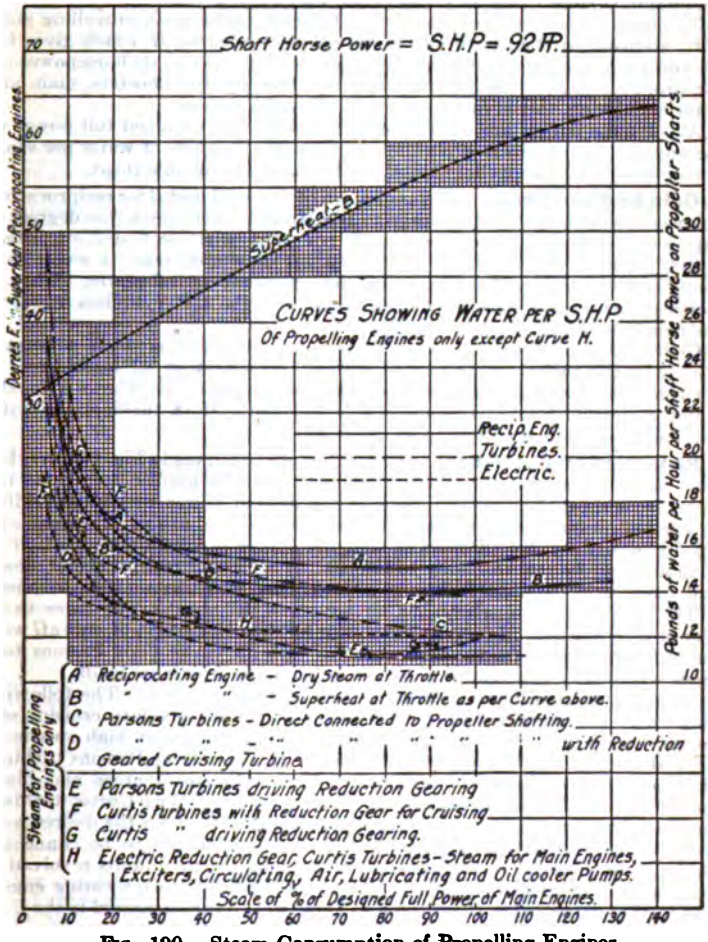

Shaft Horse Power = S.H.P = .92 I.P.
Degrees F. in Superheat-Reciprocating Engines
Pounds of water per Shaft Horse Power on Propeller Shafts
Superheat-B
CURVES SHOWING WATER PER S.H.P
Of Propelling Engines only except Curve H.
Recip. Eng.
Turbines.
Electric.
A Reciprocating Engine - Dry Steam at Throttle.
B " " - Superheat at Throttle as per Curve above.
C Parsons Turbines - Direct Connected to Propeller Shafting.
D " " " " " " " with Reduction Geared Cruising Turbine
E Parsons Turbines driving Reduction Gearing
F Curtis Turbines with Reduction Gear for Cruising
G Curtis " driving Reduction Gearing.
H Electric Reduction Gear, Curtis Turbines - Steam for Main Engines, Exciters, Circulating, Air, Lubricating and Oil cooler Pumps.
Steam for Propelling Engines only
Scale of % of Designed Full Power of Main Engines.

FIG. 190.—Steam Consumption of Propelling Engines.

B. Reciprocating engines, *Delaware* type, with superheat as per curve above marked superheat *B*.

C. Parsons turbines direct connected to the propeller shafting.

D. Parsons turbines direct connected to the propeller shafting with Reduction Geared Cruising Element for use at low powers.

E. Parsons turbines driving Reduction Gearing as the main propelling unit.

F and *F.* Curtis turbines direct connected to the propeller shafting with Reduction Geared Cruising Element for use at low powers.

G. Curtis turbines driving Reduction Gearing as the main propelling unit.

In addition to the above curves, is shown the curve *H* which gives for electric propulsion the water consumption per hour per shaft-horsepower on the propeller shaft, of main generators, exciters, electric boosters, main air, main circulating, forced lubrication and oil cooler pumps.

All of these curves are erected on percentages of "designed full power of the engines" as abscissas while the ordinates are "pounds of water per shaft horsepower on the propeller shafts" and degrees, F., of superheat.

Gain by Superheat. An examination of curves *A* and *B* for reciprocating engines shows the great advantage to be gained by even moderate degrees of superheat. This gain amounts to 8 per cent with 58.6 deg. superheat at 100 per cent of full power. These curves show a decrease in water consumption for reciprocating engines up to 80 per cent of full power. Then it increases due to the increase in back pressure in the L.P. cylinders owing to the restricted areas of the exhaust ports.

Curve *C* for the direct connected Parsons unit shows lower steam consumption than the reciprocating engine without superheat at all loads over 17 per cent and with superheat at all loads over 33 per cent. Curve *F* for the direct connected Curtis is practically as efficient. Both turbines show decreasing steam consumption up to full power.

Reduction of Water Rate. Inspection of the curves in Fig. 1 shows that with direct connected turbines two remedies can be applied to reduce the steam consumption at low fractions of the desired power without providing additional propelling elements. These are (1) the use of superheat and (2) by designing the turbines so that the curves of steam consumption as *C* and *F* will be moved bodily to the left. This would require higher steam and blade velocities at 100 per cent of power than were used on these turbines.

The gain due to superheat has already been discussed. The gains to be obtained by the second remedy are shown by the comparison of curve *G* with curve *F* for Curtis turbines and the curve *E* with the curve *C* for Parsons turbines where very appreciable gains are apparent at fractional loads.

In naval practice neither of these steps were first taken. The following plans were discussed and some tried out: (*a*) Special auxiliary cruising elements were added. These could be either disconnected at high speeds or were allowed to revolve idly in a vacuum. In the case of Parsons turbines H.P. and I.P. cruising turbines were added with steam areas and blade speeds designed for the necessary cruising speeds. With Curtis turbines special cruising nozzles in the casing of the main unit served the desired end.

(*b*) Another plan was to provide reciprocating engines to be connected to the main shaft at cruising speeds. Sometimes the main units revolved in a vacuum. In other installations the steam from the reciprocating engine was further expanded in the turbine. These were quite successful in the U. S. Navy.

(*c*) A third proposed plan was to install combined reciprocating engines and L.P. turbines. This combination seemed to promise satisfactory results but has not been used to date in the U. S. Navy. (*d*) The fourth plan is to install cruising turbines with reduction gears. These are now being used in ships when the high cruising speeds are above the desirable speeds of reciprocating engines.

The following data gives the performance of main units and auxiliaries at cruising speeds. These figures are taken from the trial reports of similar vessels and give a good idea of the relative economy of direct-connected turbines in combination with cruising elements as outlined in (*a*), (*b*) and (*d*).

Direct connected turbines with	Fraction of designed power	Pounds of water per S.H.P.-hr. for all purposes
(*a*) Direct connected cruising turbines	0.047	35.486
(*b*) Direct connected cruising reciprocating engine	0.04	29.828
(*d*) Cruising turbine connected through reduction gear.	0.04	34.543

The above table indicates an enormous gain in steam economy when the change from (*a*) to (*b*) was made but a decided loss occurred when the change from (*b*) to (*d*) took place. Performances of the (*d*) combination are indicated on Fig. 190 by the curves *FF* for Curtis turbines and the curve *D* for Parsons turbines.

The curve *H* for the electric drive in Fig. 190 includes auxiliaries not generally chargeable to the main propelling unit. When allowance is made for these, curve *H* drops more nearly to the position of curve *F*. These curves would then cross at about 30 per cent below which the electric drive appears most economical.

The steam economies of the different types as shown in Fig. 190 are given in the following table when the standing in the order of economy is stated at various percentages of rated power. It must be remembered that these values refer to the propelling engine alone and do not include the propulsion efficiencies of the propellers nor the combined efficiencies of propellers and engines.

Type	Percentage of designed power		
	5	30	Full
A-Reciprocating engine—dry steam	7	6	7
B-Reciprocating engine—superheat	6	4	6
C-Parsons turbines—D. C.	8	5	4
D-Parsons turbines—D. C. with G. C. E.	4		
E-Parsons turbines—reduction gear	5	2	1
F-Curtis turbines—D. C.	9	7	5
F-Curtis turbines—D. C. with G. C. E.	3		
G-Curtis turbines—reduction gear	2	3	3
H-Electric propulsion	1	1	2

For purposes of comparison the foregoing types of engines were applied to a Dreadnaught type of vessel and propeller performances from three typical U. S. Naval vessels were chosen as representative values for various desired speeds. These desired propeller speeds were 125 R.P.M. the low speed for reciprocating engines; 214 R.P.M. for the intermediate low speed and 310 the low speed for the turbines. The vessel is assumed to have an effective propeller horsepower of 15,250 at 21.3 knots speed. Its effective horsepower and speed curves are shown in Fig. 191.

The three curves of propulsive coefficients are laid down on Fig. 192 on percentage of designed power.

Comparison of Types. It is now possible to make **comparisons of the different types of propulsion by a true measure of economy.** This is not the measure and comparison of the steam economies of the types of

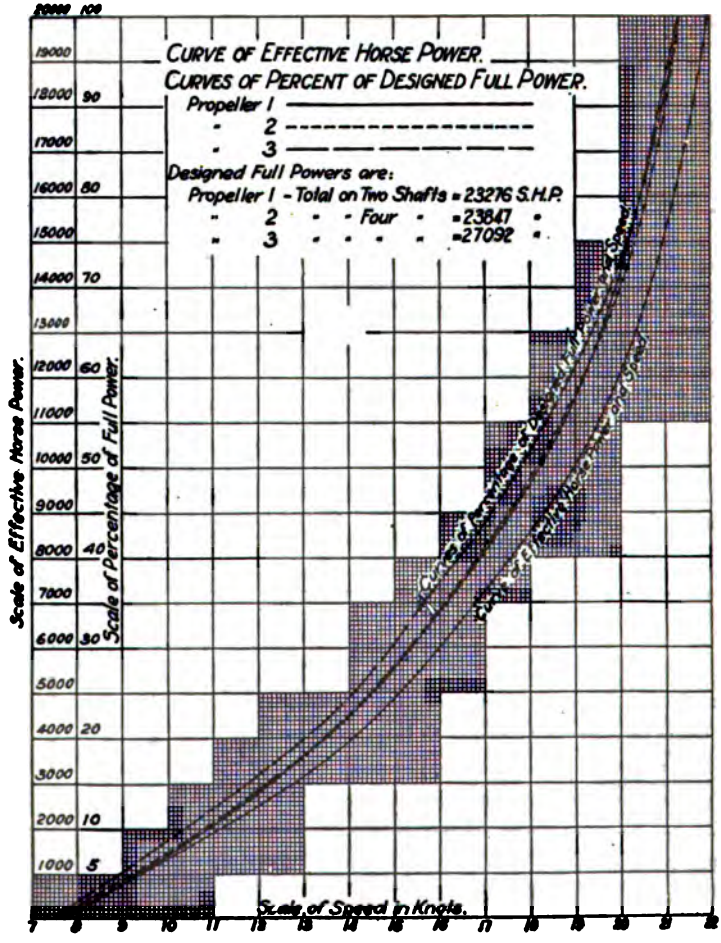


FIG. 191.—Relation of Effective Horse Power to Speed.

engines alone but rather of the combined steam economies of the engines and the propulsive efficiencies of the propellers when driving the vessels at the same speeds under similar conditions of resistance.

To make such a comparison Fig. 192 has been prepared as a preliminary step. The curves shown carrying the same letters of the alphabet as Fig. 190 are the values of "Pounds of water per shaft horsepower ÷ Propulsive Coeffi-

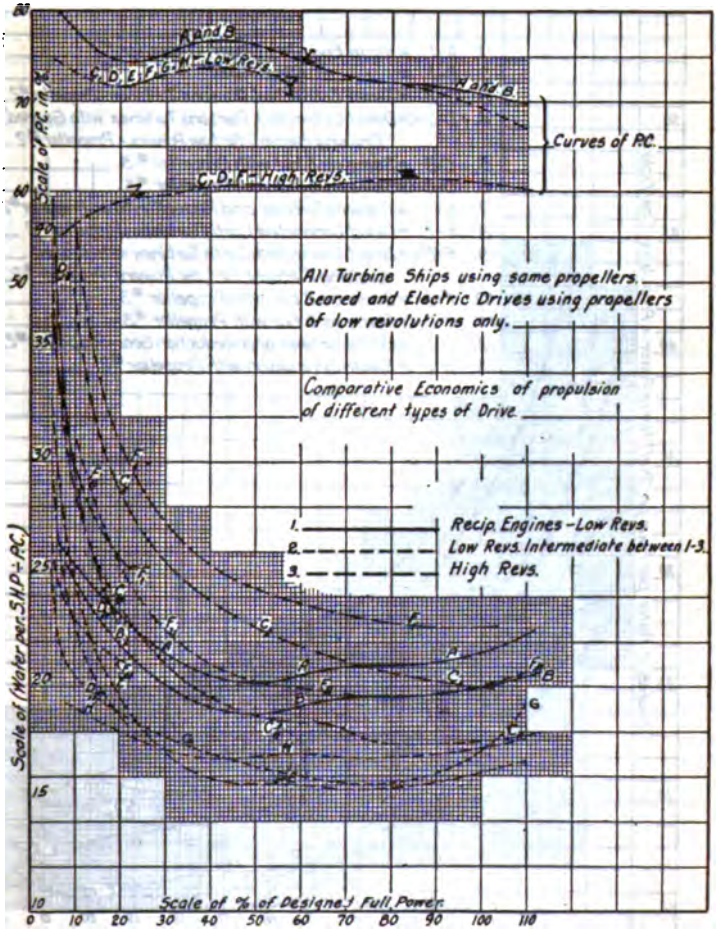

A and B
C D E F G H I J K L M
A and B
Curves of P.C.
C D K — High Revs.
All Turbine Ships using same propellers.
Geared and Electric Drives using propellers
of low revolutions only.
Comparative Economics of propulsion
of different types of Drive
1. Recip. Engines – Low Revs.
2. Low Revs. Intermediate between 1-3.
3. High Revs.
Scale of (Water per S.H.P. ÷ P.C.)
Scale of P.C. in %
Scale of % of Designed Full Power
0 10 20 30 40 50 60 70 80 90 100 110
10 15 20 25 30 35 40 50 70 80

FIG. 192.—Comparative Economies of Propulsion at Per cents of Designed Full Power.

cient at the same percentage of the total designed power." Curves C_1 and F_1 are curves corresponding to C and F divided by the Propulsive Coefficient (3), while curves F_{11} and C_{11} are for propulsive coefficient (2).

The final comparisons are shown on Fig. 193 which is a combination of Figs. 191 and 192. Fig. 193 shows the relative economies of engines and propellors when running at various speeds in knots rather than in per cent of rated power

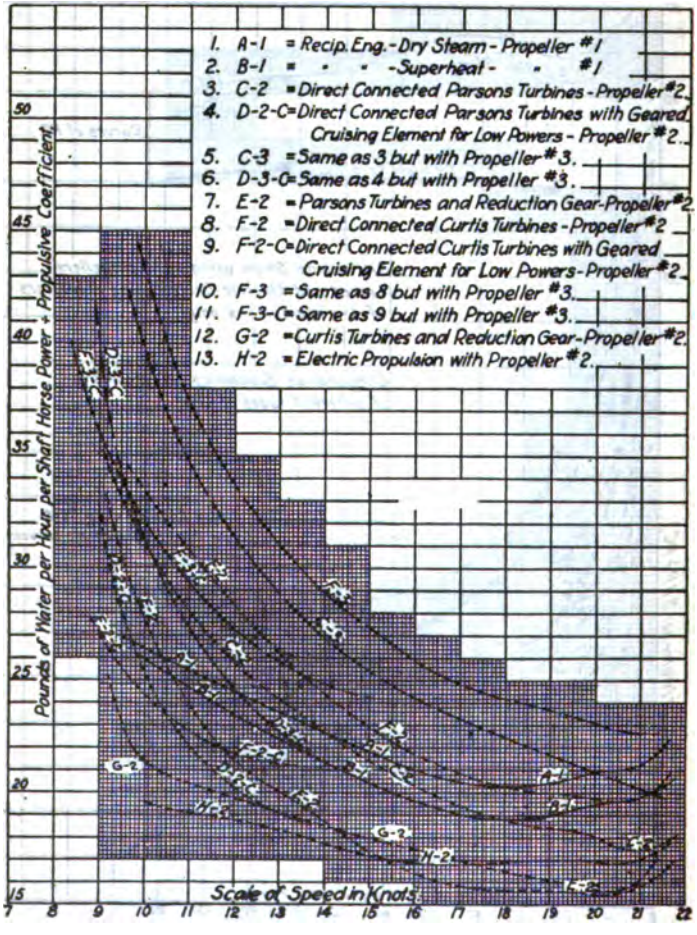

1. A-1 = Recip. Eng. - Dry Steam - Propeller #1
2. B-1 = " " -Superheat - " #1
3. C-2 = Direct Connected Parsons Turbines - Propeller #2.
4. D-2-C = Direct Connected Parsons Turbines with Geared Cruising Element for Low Powers - Propeller #2.
5. C-3 = Same as 3 but with Propeller #3.
6. D-3-C = Same as 4 but with Propeller #3.
7. E-2 = Parsons Turbines and Reduction Gear - Propeller #2.
8. F-2 = Direct Connected Curtis Turbines - Propeller #2
9. F-2-C = Direct Connected Curtis Turbines with Geared Cruising Element for Low Powers - Propeller #2.
10. F-3 = Same as 8 but with Propeller #3.
11. F-3-C = Same as 9 but with Propeller #3.
12. G-2 = Curtis Turbines and Reduction Gear - Propeller #2.
13. H-2 = Electric Propulsion with Propeller #2.
Pounds of Water per Hour per Shaft Horse Power ÷ Propulsive Coefficient.
50
45
40
35
30
25
20
15
Scale of Speed in Knots
7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22
G-2
H-2

FIG. 193.—Comparative Economies of Propulsion at Various Speeds

as Fig. 192. The ordinates again are pounds of steam per shaft horsepower divided by the propulsive coefficients which are usually obtained with the type of hull possessed by the assumed standard vessel and the different propellers.

Tabulating the results given by Fig. 193, for 21, 19, 15 and 10 knots, and giving 100 per cent as the efficiency of that combination which is the best at any given speed, also allowing an average correction of 5 per cent for electric propulsion in order to give an approximation to the actual economy of that system by allowing this percentage, which is small, for the demand of the main air, main circulating, forced lubrication and oil cooler pumps, we have:

No.	Letter	Type	Propeller No.	Measure of economy per cent at			
				Speed		Knots	
				10	15	19	21
1	F—3	D. C. Curtis	3	43.1	59.6	66.2	70.0
2	C—3	D. C. Parsons	3	46.5	64.3	72.1	78.2
3	F—3—C	1 with Geared Cr.	3	56.1	67.5	66.2	70.0
4	D—3—C	2 with Geared Cr.	3	59.4	80.0	72.1	78.2
5	F—2	D. C. Curtis	2	55.5	71.4	78.2	89.0
6	C—2	D. C. Parsons	2	56.1	77.1	85.0	90.8
7	F—2—C	5 with Geared Cr.	2	76.2	77.8	78.2	80.0
8	D—2—C	6 with Geared Cr.	2	73.2	90.0	85.0	90.8
9	E—2	Parsons Geared	2	66.9	93.9	100.0	100.0
10	G—2	Curtis Geared	2	91.6	94.5	97.5	96.9
11	H—2	Electric Propulsion	2	100.0	100.0	96.9	97.8
12	A—1	Recip. Dry Steam	1	70.2	75.0	76.7	74.3
13	B—1	Recip. Sup. Steam	1	71.2	80.0	82.4	80.0

On examination of the foregoing table and of Fig. 193, there is seen to exist a large difference between the economies of all the Parsons direct-connected turbine installations and those in which the Curtis direct-connected marine turbine are used, with the difference decidedly in favor of the Parsons installations except where geared cruising elements are used. When such elements are fitted the Curtis installation with the medium speed propeller No. 2 is seen to be slightly superior from 10½ knots down. Both Parsons and Curtis installations with geared cruising units are superior in economy to the Parsons Geared installations at speeds below 14½ to 15 knots, but are both inferior to the Curtis Geared turbines at all speeds.

Furthermore, it is seen that the earlier superiority of the reciprocating engine practically disappeared when the propeller conditions for the other types of drive were changed from those represented by Propeller No. 3 to those of Propeller No. 2, and geared cruising elements were adopted.

With the adoption of the Direct Geared drive another advance in economy of propulsion was realized, that with Parsons turbines at the highest speed and at the high cruising speed of 19 knots standing No. 1 on the list, being far superior to all the other types except electric propulsion and the Curtis geared turbines at these speeds, but being inferior at 10 knots to both Nos. 7 and 8 conditions of the table and being far inferior to conditions Nos. 10 and 11 of the same table.

It is seen that condition No. 11 is far superior in economy at all speeds up to and slightly above 15 knots, to all other types of machinery, yet at the higher speeds it is apparently very slightly inferior to both geared drives. This inferiority is, however, so small that it is completely outweighed by the other advantages possessed by the Electric Drive and can be disregarded in making choice between the two types.

As to the Curtis marine turbine which shows up so well in the above comparison, its resort for assistance in raising its economy of propulsion to a still

higher plane appears to be high superheat which it can use with a much greater degree of safety than can the Parsons turbine unless the latter be fitted with an impulse H.P. stage, a change which is bound to come in the next few years on this side of the water as it has already come on the other.

TYPES OF AMERICAN STEAM TURBINES

This section is devoted to descriptions of various types of American built Steam Turbines that may be adopted to marine services.

Some of these units are designed for direct-connection to the propeller shaft and therefore run at slow speed to conform with propeller requirements. Such turbines hence must be long and of large diameter. They are generally less efficient than the high speed types.

Methods of securing the advantages of high turbine speed and slow propeller speed by means of mechanical gearing, hydraulic gearing and the electric drive are fully discussed on pp. 692 to 709. These sets operate at higher efficiency than the direct-connected slow speed turbines and use much less steam. They are therefore being widely adopted both for naval use and in the Merchant Marine.

The different large geared turbines will be fully described in the following paragraphs. Descriptions of several standard types of turbines suitable for electric drive are also included.

Various types of small turbines have been developed for driving such auxiliaries as pumps, blowers, fans, electric generators, etc. These may be either direct-connected or geared. Many of these will be described. These turbines have certain advantages over reciprocating engines. They are lighter and more compact, which fit them especially for marine service. They require a minimum of attention and are simple in construction, which tends toward longer life with fewer repairs and shut-downs. These sets are usually operated non-condensing and the exhaust is used to heat the feed water. The absence of oil in the exhaust steam is a decided advantage and reduces the work of keeping the filters clean. Even in the matter of steam consumption these turbine units compare favorably with engine sets, especially when geared so that both turbine and auxiliary may run at their economical speeds. Their economy is maintained through a long life, as there is no increase in steam consumption due to leakage past valves and pistons. The probabilities are that in the near future the use of the turbo-driven auxiliary will become as common on shipboard as in standard land practice.

WESTINGHOUSE MARINE TURBINES

Marine Propelling Turbines

Marine propelling turbines are built by the Westinghouse Electric & Manufacturing Co. in a variety of sizes to suit the requirements of all classes of merchant marine and naval vessels. The accompanying illustrations show one of the types which has come into considerable use. This is a cross-compound unit of about 3000 shaft horse power consisting of a high and a low pressure turbine, through which the steam passes in series. Each turbine drives one of the high speed pinions in a train of double reduction gears, which reduces the 3600 P.R.M. of the turbine down to 90 R.P.M. on the main shaft. For a description of the flexible frame gears employed see marine gear section, p. 818.

The high-pressure turbine, Fig. 194, consists of a two-row impulse stage followed by a reaction drum, from which the steam exhausts through a cross-connecting pipe into the **low-pressure turbine,** Fig. 195, where a second reac-

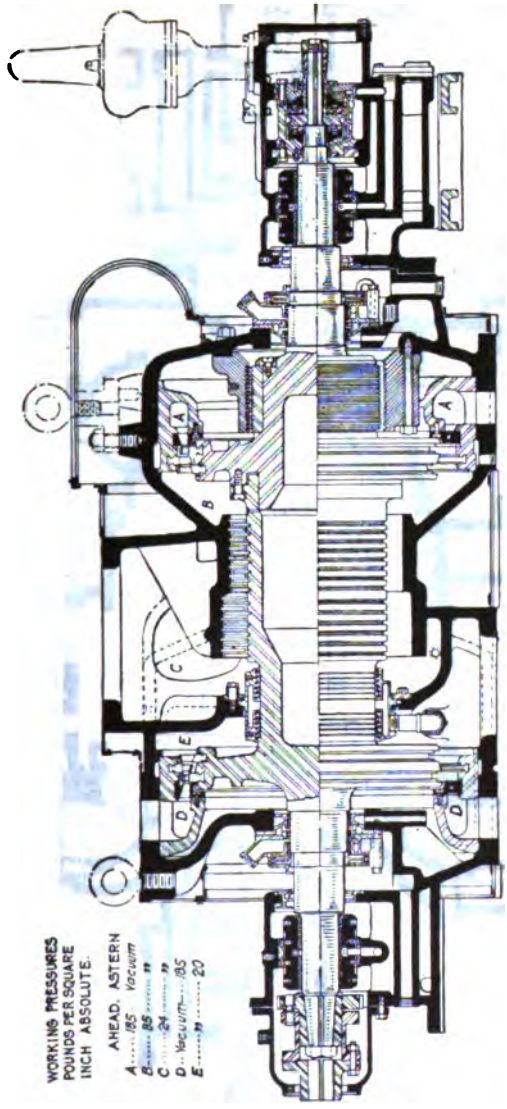


FIG. 194.—Westinghouse Marine Turbine—High Pressure Cylinder.

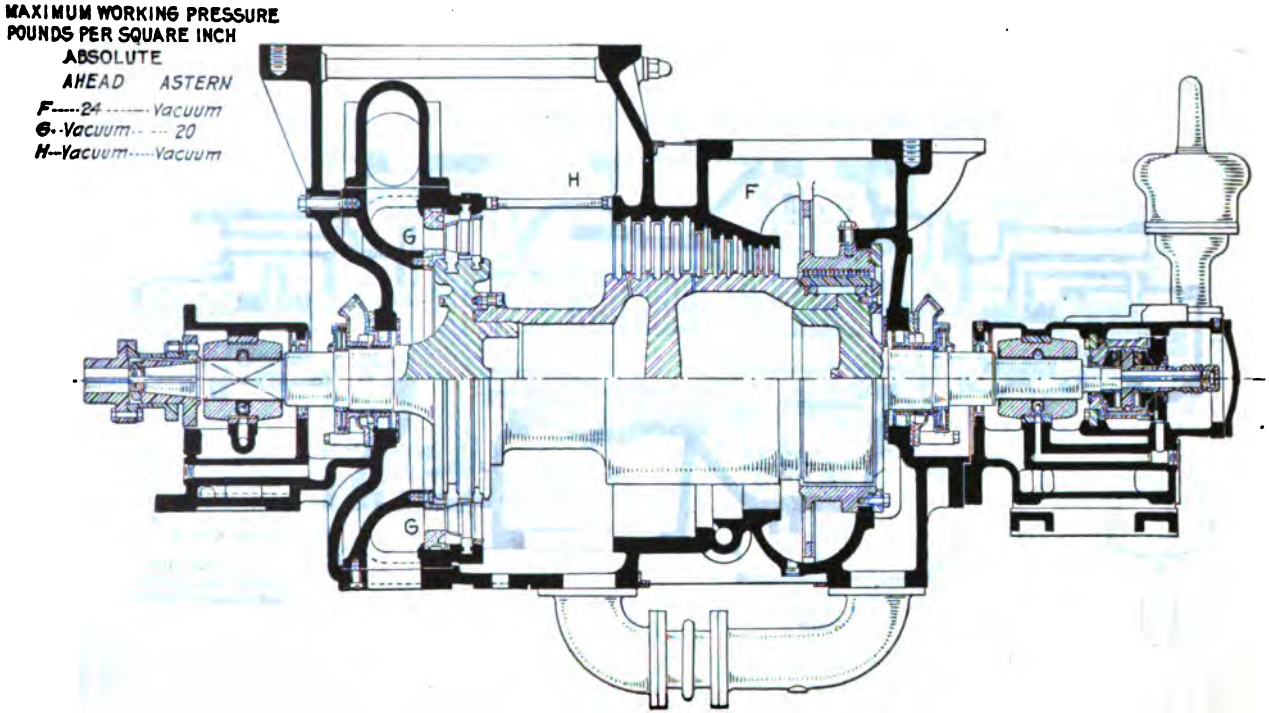


FIG. 195.—Westinghouse Marine Turbine—Low Pressure Cylinder.

tion drum completes the expansion. By an arrangement of emergency pipes and valves it is possible to operate either turbine alone under high pressure steam, exhausting directly into the condenser. **The casings** are split horizontally. They are of cast iron except at the high pressure end where a steel casting is used

The reversing requirements are taken care of by placing a two-row impulse stage in the exhaust end of each turbine. These are compounded in a similar manner to the ahead elements. High-pressure steam is admitted to the H.P. turbine only, from which it exhausts through a cross-connecting pipe to the L.P. astern steam chest. The astern element of the H.P. turbine is separated from the ahead stages by means of a diaphragm which is secured as an integral part of the turbine casing. Reversing power to the extent of about 60 per cent of the full power ahead is obtained with the same steam flow as for full power ahead.

An arrangement of emergency pipes and valves similar to that for ahead operation makes it possible to operate either the H.P. or the L.P. astern element alone, using high-pressure steam and exhausting directly into the condenser.

As in all other turbines of this type, when in operation there is a steam thrust, tending to move the spindle endwise. This is partly counterbalanced by means of a balance piston or dummy placed at the ahead end of each turbine and the remainder of the end thrust is absorbed by a double-faced, single collar thrust bearing of the "Kingsbury" type. By this arrangement a short dummy is obtained which adds considerably to the reliability and prolonged economy of the machine. An equilibrium pipe leading from the back of the dummy to the exhaust end of the turbine serves to carry away whatever steam leakage there is through the dummy and to prevent the building up of a back pressure which would destroy the effect of the balance piston.

The exhaust steam from the auxiliary apparatus, having a pressure somewhat above atmospheric, can be used to advantage in the low-pressure turbine and a suitable connection is provided for this purpose. A constant pressure valve is interposed which automatically admits to the turbine any excess of exhaust steam not used in the feed heater, and thus keeps the pressure in the auxiliary header approximately constant.

The spindle is of the drum type built up of two or more steel parts. A very stiff construction is thereby obtained in which the deflections are so small that the critical speed is far above the normal R.P.M. The impulse wheels are an integral part of the spindle body. **The dummy piston** is formed by a steel ring, shrunk and keyed onto the turbine spindle. On this ring are turned square ridges alternately high and low. In the cylinder dummy ring, which is a casting bolted to the cylinder, are fitted narrow aluminum strips, which are turned to correspond with the ridges on the dummy piston. The inner edges of the aluminum strips are approximately $\frac{1}{32}$ inch wide and clear the ridges on the dummy piston by only a few thousandths of an inch, thereby reducing the steam leakage to a very small amount. However, since the close clearance is in a radial direction only, the dummy imposes no limits on the end movement of the spindle. The diaphragm packing between the ahead and astern elements of the H.P. turbine consists of two sets of labyrinth packing strips with an annular space between, from which a pipe leads to a suitable stage in the L.P. turbine. Any leakage past the first set of labyrinth rings is thus taken care of and utilized to advantage.

The glands for preventing leakage where the spindle passes through the casing at each end are of the combined steam and water sealed type, see p. 633. **The water gland** is designed to effectively seal at all speeds from above half speed, and below that the **steam seal** is used.

The water gland consists of a runner similar to the runner of a centrifugal pump, which revolves in the gland casing with comparatively small clearances. Fresh water is supplied from a gravity tank and a solid annulus of water is maintained at the outer edge of the runner, thereby producing an hermetic seal, which precludes leakage entirely.

The steam seal on the L.P. turbine consists of two sets of aluminum packing rings with an opening between, to which steam is admitted at a reduced pressure. The steam seal on the H.P. turbine is somewhat different since it has to pack against a pressure greater than atmosphere. In this case there are three sets of packing rings. Between the two inner sets is a leak-off chamber, piped to a suitable connection on the L.P. turbine, which serves to dispose of the leakage steam and at the same time utilises it to good advantage. Reduced pressure sealing steam is supplied to the opening between the two outer sets of rings the same as in the L.P. glands.

These spindle glands can be removed for inspection or repair without disturbing the turbine casing.

The main inlet nozzles which direct the steam against the impulse blading in both the ahead and astern elements are made of phosphor bronze, cast in multiple and carefully filed to a smooth finish on the inside.

The impulse blades are of very rugged construction, being milled out of cold drawn nickel steel stock and fitted into tee slots in the spindle. The **shrouding** is formed as an integral part of the blades.

The reaction blading is made from cold drawn bronze stock. The lower end of each blade is upset to form a foot which extends down into a supplementary groove cut in the bottom of the main blading groove. Between the blades are inserted dovetail packing pieces which do not extend into the lower groove, and thereby provide an interlocking construction which requires no calking. The larger blades are further secured in the slots by means of a series of sister wedges on one side.

A fly ball governor is installed on the forward end of each turbine shaft which operates a governor valve in the main steam line through the medium of a steam relay valve.

Steam to the turbine is controlled by means of a combined ahead and astern maneuvering valve. The arrangement consists of twin balance valves cast in a single body, the valve stems being so connected through a crank and gear as to be operated by a single handwheel. Turning the wheel in one direction admits steam to the ahead nozzles, and in the other direction to the astern nozzles. It is impossible to open both valves at the same time. Some nozzles on the ahead end may be shut off by hand-controlled valves when operating at slow speeds.

The power from the turbine is transmitted to the pinion through a **sliding pin coupling** and flexible shaft, care being taken to thoroughly lubricate the sliding parts by means of a centrifugal lubricator which catches the oil spilled from the main bearing.

All the main details of construction are described in other parts of this section.

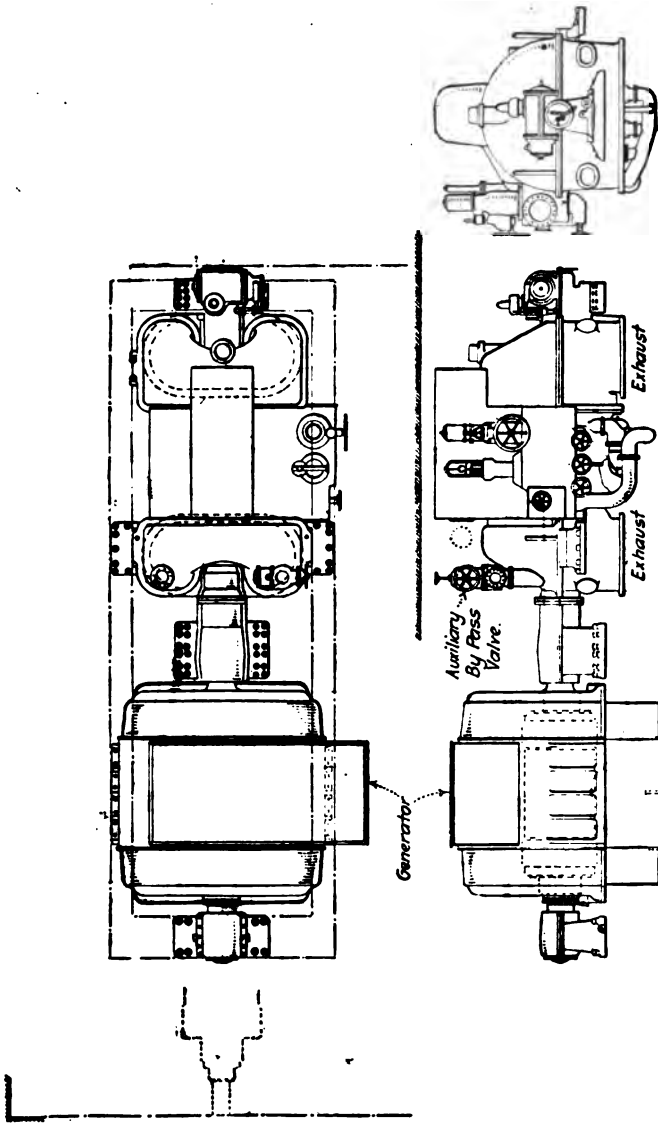


Fig. 196.—Turbo-generator, U.S.S. Tennessee.

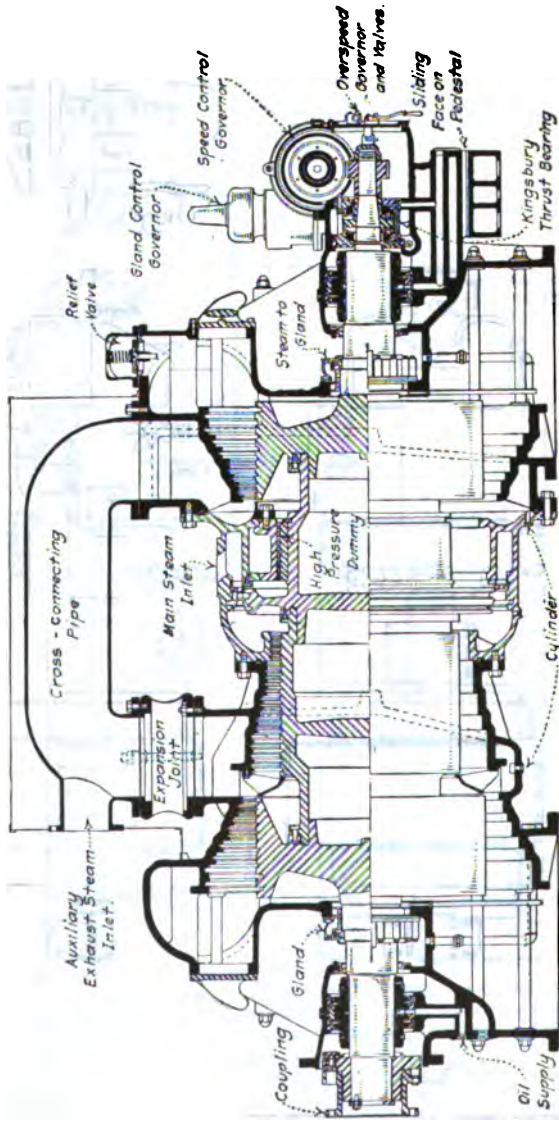


FIG. 197.—Main Turbine, U.S.S. Tennessee.

Large Turbines for Electric Drive

Fig. 196 shows the general outlines of a turbo-generator set furnished by Westinghouse for electric drive on shipboard. Fig. 197 shows a cross-section of the steam turbine. This unit is of the **semi-double flow type.** Steam enters through **governor control valves** and after passing through expanding nozzles is directed onto a two row Curtis stage. A barrel of Parsons blading is placed on the other side of this Curtis stage. A portion of the steam continues through the succeeding L.P. stages. The other portion is carried through the overhead connecting pipe with an expansion joint to the other L.P. barrel placed on the opposite end of the spindle. The L.P. section is thus double-flow, and there are two exhausts to the condenser. The condenser is placed directly below the turbine. The shaft is built up in drum sections as shown with solid web hubs at certain points. The L.P. sections run at the highest blade speeds and are built as discs with the spindle ends as an integral part of the disc. The Curtis blades are standard Westinghouse construction with dovetail base and welded shrouds. The Parsons blades are all of standard Westinghouse construction with the exception of the last two rows on each L.P. section where blades with dovetailed bases are used to withstand the higher peripheral speeds.

The central portion of the **casing** is made of cast steel so that high pressure and high superheat may be used with the unit. The other two sections of the casing which include the two exhaust openings are of cast iron. The exhaust sections are firmly braced and reinforced by pipe spacers and heavy steel bolts. A small atmospheric relief valve is placed as a sentinel valve on one exhaust end.

The pedestals are cast integral with the casing at both ends. The **bearings and coupling** are of standard Westinghouse design, both described elsewhere.

The **glands** consist of a combined steam seal and water gland. The steam seal is automatically regulated by a secondary governor.

A Kingsbury thrust bearing is used to care for any end thrust and to adjust the spindle axially in the casing.

The **governor** is of unusual design and is shown in section in Fig. 198. It is driven from the main spindle by helical gearing. The clutch and lever shown at the left hand side of the section are drawn to the right when the governor weights move outward under the action of centrifugal force due to increasing speed. This movement of the clutch and the central spindle is opposed by a variable oil pressure behind the piston on the opposite end of the spindle. This oil pressure causes the governor to act exactly like a weighted governor. In this case, however, the governor loading may be made variable by varying the oil pressure behind the piston and the turbine speed can thus be varied accordingly. The oil pressure on this piston is supplied from the main lubricating system but its pressure can be varied at the speed control centre by a special relief valve which regulates the pressure on the line supplying oil to this piston. Hence the necessary variations in speed are secured.

Auxiliary exhaust steam is ordinarily furnished to the nipple on the connecting pipe between the two L.P. turbine sections, as shown in Fig. 197. The auxiliary **exhaust** is also connected through an auxiliary exhaust by-pass valve, as shown in Fig. 196, to the exhaust end of the turbine and thus directly to the condenser. The construction of this by-pass valve is shown in Fig. 199.

The by-pass valve itself is controlled through a bell-crank attached to its valve stem. Provision is made for hand operation or for automatic operation

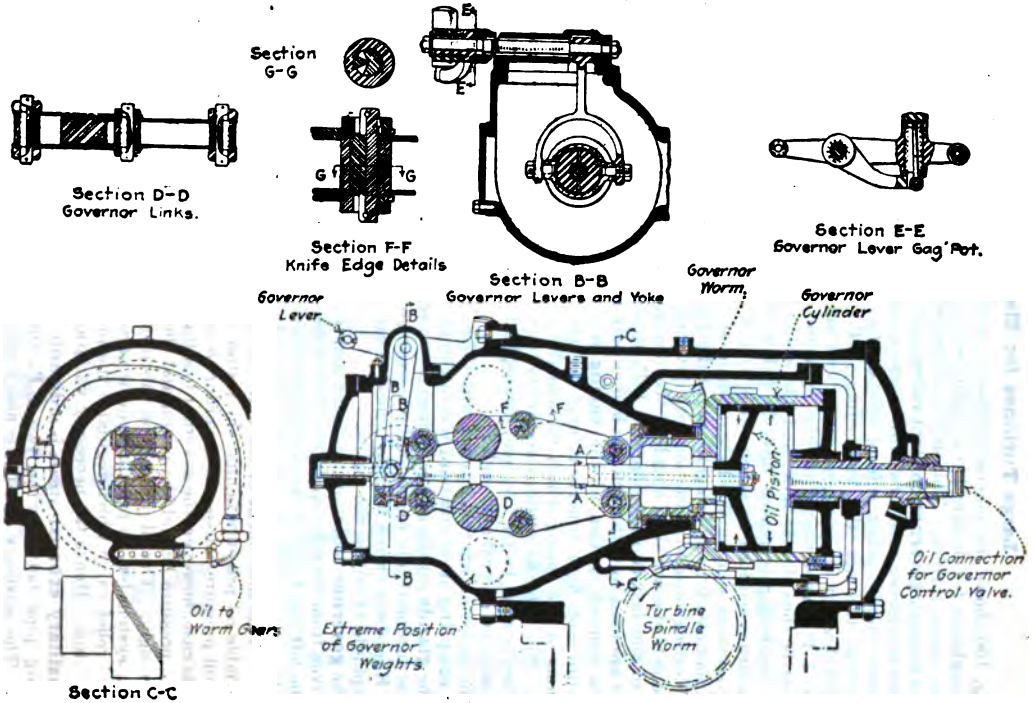


FIG. 198.—Speed Control Governor, Main Turbine, U.S.S. Tennessee.

from the oil relay cylinder of the main governor valve. This is done by means of the oil piston on the right hand side of the horizontal rod.

The valve shown in Fig. 199 is in the wide open position being held so by the hand valve. Normally the hand wheel would be at the outer limit and the valve would be resting tightly on its seat. As long as the governor valve on

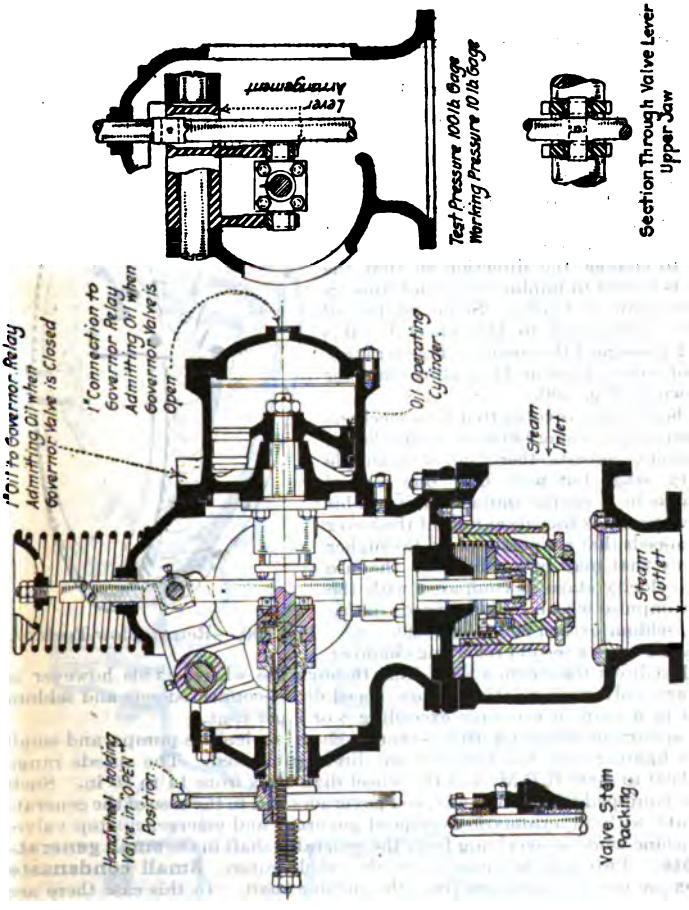


FIG. 199.—Auxiliary By-pass Valve, Maine Turbine, U.S.S Tennessee.

the main turbine is open an oil relay valve furnishes oil under pressure to the right hand side of the oil piston and forces it to the left. By means of the bell crank the valve is held tightly on its seat. However, when the governor cuts steam off the main turbine, the relay valve reverses the oil connection and furnishes oil under pressure to the opposite side of the oil piston which causes

it to move to the right and thus through the bell crank to open the auxiliary exhaust direct to the condenser.

Another special feature of design is the provision of a **secondary governor** to control the supply of steam to the steam seals on the glands. At full speed of the turbine the water glands are sufficient to effectively seal against vacuum. But at low speeds steam must be added to the steam seal. This is done automatically by this secondary governor.

Automatic overspeed governors and valves of standard design are used on this unit.

Small Turbines

The construction of these turbines is that known as the **re-entry type.** The turbine consists of a single wheel carrying one row of blades. A nozzle expands the steam from the inlet to the exhaust pressure and delivers it to the blades at an angle of 18 to 20 deg. A reversing chamber receives the steam after its first passage through the blades. It forms a guide to change the direction so that the steam is caused to impinge a second time on the same row of blades. Some additional energy is imparted to the wheel by this second passage of the steam. This arrangement of nozzle, rotor and reversing chamber is shown in Fig. 200.

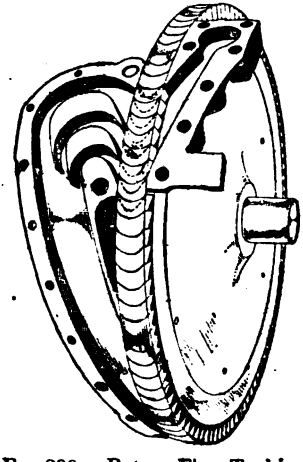


FIG. 200.—Return Flow Turbine.

As the velocity is converted into mechanical energy by passing twice over the blades, the turbine consists therefore of a simple velocity stage but with only one row of blades, as in a simple impulse unit. This construction has the advantage of the lower wheel speeds that are possible and the higher economy that may be secured by the use of the velocity stage as compared with the simple impulse wheel. Blade speeds in this design seldom exceed 600 ft. per sec.

In a few cases a second reversing chamber is used to direct the steam a third time through the wheel. This however is necessary only with relatively slow speed direct-connected sets and seldom results in a gain in economy exceeding 8 or 9 per cent.

For apparatus requiring little power, such as condensate pumps, and small electric lighting sets, the turbines are direct connected. The speeds range from 2500 to 4000 R.P.M. and the wheel diameters from 14 to 20 in. Such sets are equipped with constant speed governors and in the case of the generating units with an automatic overspeed governor and emergency stop valve. The turbine wheel is overhung from the generator shaft in the **small generating sets.** Two ball bearings carry the whole rotor. **Small condensate pumps** are usually overhung from the turbine shaft. In this case there are also but two bearings, which are babbitted and ring lubricated.

For circulating pumps which are inherently slow speed and **for larger electric generating sets,** the turbine is made small and operated at speeds ranging from 6000 to 7200 R.P.M. The speed is reduced by mechanical gearing. The turbine casing in such sets under 100 H.P. capacity is supported from the gear housing.

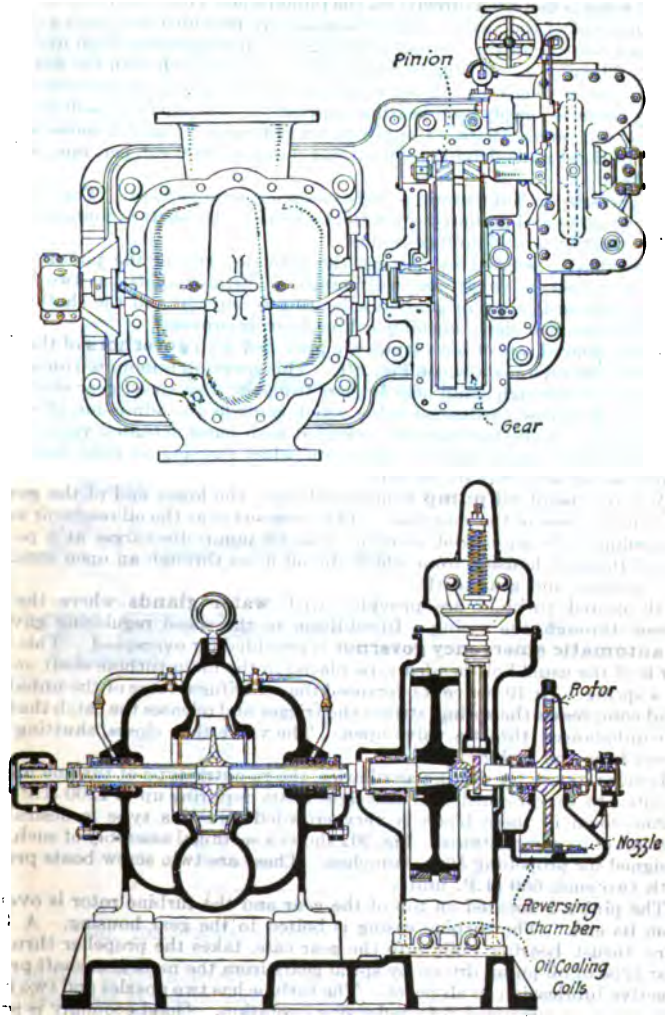


FIG. 201.—Westinghouse Geared Centrifugal Pump Drive.

The glands for preventing leakage where the spindle passes through the casing at each end are of the combined steam and water sealed type, see p. 633. **The water gland** is designed to effectively seal at all speeds from above half speed, and below that the **steam seal** is used.

The water gland consists of a runner similar to the runner of a centrifugal pump, which revolves in the gland casing with comparatively small clearances. Fresh water is supplied from a gravity tank and a solid annulus of water is maintained at the outer edge of the runner, thereby producing an hermetic seal, which precludes leakage entirely.

The steam seal on the L.P. turbine consists of two sets of aluminum packing rings with an opening between, to which steam is admitted at a reduced pressure. The steam seal on the H.P. turbine is somewhat different since it has to pack against a pressure greater than atmosphere. In this case there are three sets of packing rings. Between the two inner sets is a leak-off chamber, piped to a suitable connection on the L.P. turbine, which serves to dispose of the leakage steam and at the same time utilises it to good advantage. Reduced pressure sealing steam is supplied to the opening between the two outer sets of rings the same as in the L.P. glands.

These spindle glands can be removed for inspection or repair without disturbing the turbine casing.

The main inlet nozzles which direct the steam against the impulse blading in both the ahead and astern elements are made of phosphor bronze, cast in multiple and carefully filed to a smooth finish on the inside.

The impulse blades are of very rugged construction, being milled out of cold drawn nickel steel stock and fitted into tee slots in the spindle. The **shrouding** is formed as an integral part of the blades.

The reaction blading is made from cold drawn bronze stock. The lower end of each blade is upset to form a foot which extends down into a supplementary groove cut in the bottom of the main blading groove. Between the blades are inserted dovetail packing pieces which do not extend into the lower groove, and thereby provide an interlocking construction which requires no calking. The larger blades are further secured in the slots by means of a series of sister wedges on one side.

A fly ball governor is installed on the forward end of each turbine shaft which operates a governor valve in the main steam line through the medium of a steam relay valve.

Steam to the turbine is controlled by means of a combined ahead and astern maneuvering valve. The arrangement consists of twin balance valves cast in a single body, the valve stems being so connected through a crank and gear as to be operated by a single handwheel. Turning the wheel in one direction admits steam to the ahead nozzles, and in the other direction to the astern nozzles. It is impossible to open both valves at the same time. Some nozzles on the ahead end may be shut off by hand-controlled valves when operating at slow speeds.

The power from the turbine is transmitted to the pinion through a **sliding pin coupling** and flexible shaft, care being taken to thoroughly lubricate the sliding parts by means of a centrifugal lubricator which catches the oil spilled from the main bearing.

All the main details of construction are described in other parts of this section.

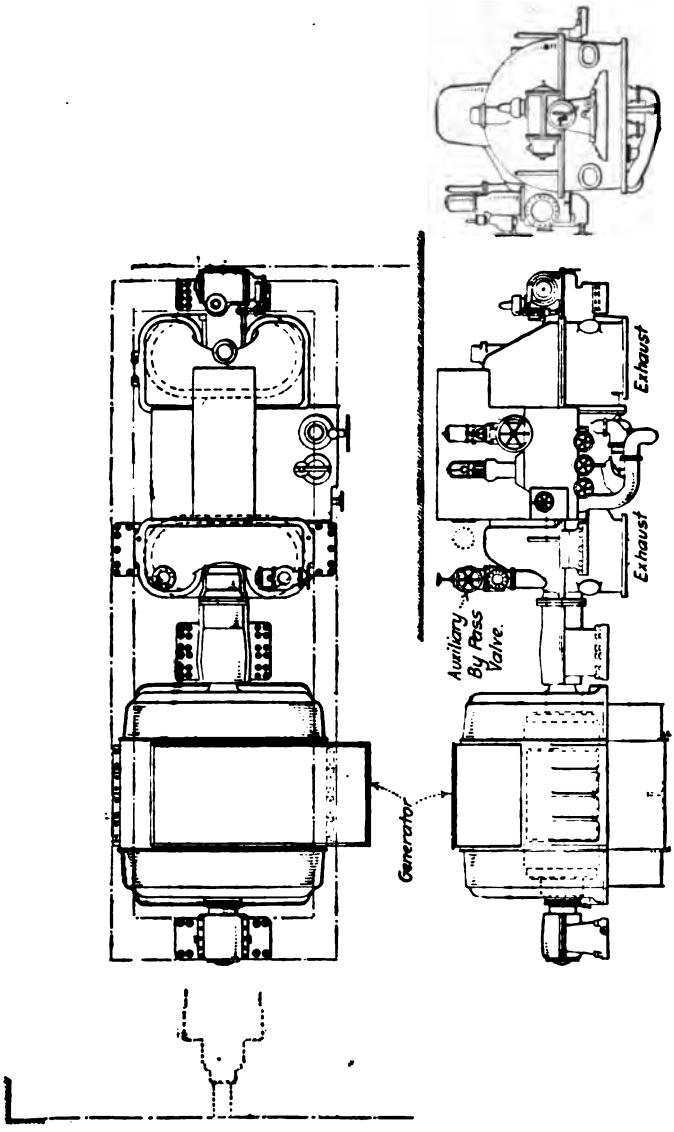


FIG. 196.—Turbo-generator, U.S.S. Tennessee.

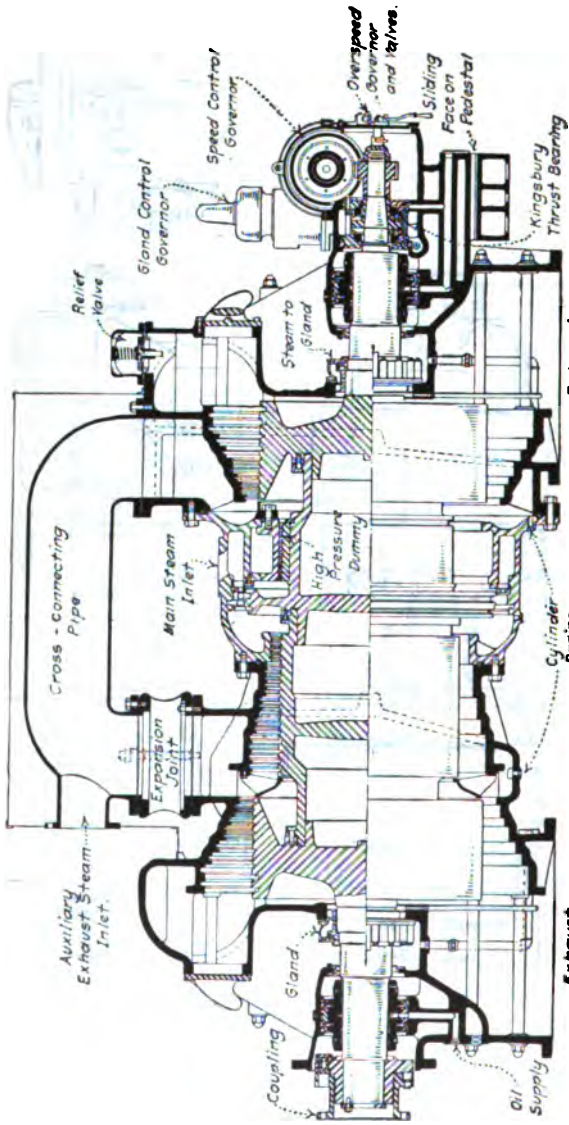


Fig. 197.—Main Turbine, U.S.S. Tennessee.

Large Turbines for Electric Drive

Fig. 196 shows the general outlines of a turbo-generator set furnished by Westinghouse for electric drive on shipboard. Fig. 197 shows a cross-section of the steam turbine. This unit is of the **semi-double flow type**. Steam enters through **governor control valves** and after passing through expanding nozzles is directed onto a two row Curtis stage. A barrel of Parsons blading is placed on the other side of this Curtis stage. A portion of the steam continues through the succeeding L.P. stages. The other portion is carried through the overhead connecting pipe with an expansion joint to the other L.P. barrel placed on the opposite end of the spindle. The L.P. section is thus double-flow, and there are two exhausts to the condenser. The condenser is placed directly below the turbine. The shaft is built up in drum sections as shown with solid web hubs at certain points. The L.P. sections run at the highest blade speeds and are built as discs with the spindle ends as an integral part of the disc. The Curtis blades are standard Westinghouse construction with dovetail base and welded shrouds. The Parsons blades are all of standard Westinghouse construction with the exception of the last two rows on each L.P. section where blades with dovetailed bases are used to withstand the higher peripheral speeds.

The central portion of the **casing** is made of cast steel so that high pressure and high superheat may be used with the unit. The other two sections of the casing which include the two exhaust openings are of cast iron. The exhaust sections are firmly braced and reinforced by pipe spacers and heavy steel bolts. A small atmospheric relief valve is placed as a sentinel valve on one exhaust end.

The pedestals are cast integral with the casing at both ends. The **bearings and coupling** are of standard Westinghouse design, both described elsewhere.

The **glands** consist of a combined steam seal and water gland. The steam seal is automatically regulated by a secondary governor.

A Kingsbury thrust bearing is used to care for any end thrust and to adjust the spindle axially in the casing.

The **governor** is of unusual design and is shown in section in Fig. 198. It is driven from the main spindle by helical gearing. The clutch and lever shown at the left hand side of the section are drawn to the right when the governor weights move outward under the action of centrifugal force due to increasing speed. This movement of the clutch and the central spindle is opposed by a variable oil pressure behind the piston on the opposite end of the spindle. This oil pressure causes the governor to act exactly like a weighted governor. In this case, however, the governor loading may be made variable by varying the oil pressure behind the piston and the turbine speed can thus be varied accordingly. The oil pressure on this piston is supplied from the main lubricating system but its pressure can be varied at the speed control centre by a special relief valve which regulates the pressure on the line supplying oil to this piston. Hence the necessary variations in speed are secured.

Auxiliary exhaust steam is ordinarily furnished to the nipple on the connecting pipe between the two L.P. turbine sections, as shown in Fig. 197. The auxiliary exhaust is also connected through an auxiliary exhaust by-pass valve, as shown in Fig. 196, to the exhaust end of the turbine and thus directly to the condenser. The construction of this by-pass valve is shown in Fig. 199.

The by-pass valve itself is controlled through a bell-crank attached to its valve stem. Provision is made for hand operation or for automatic operation

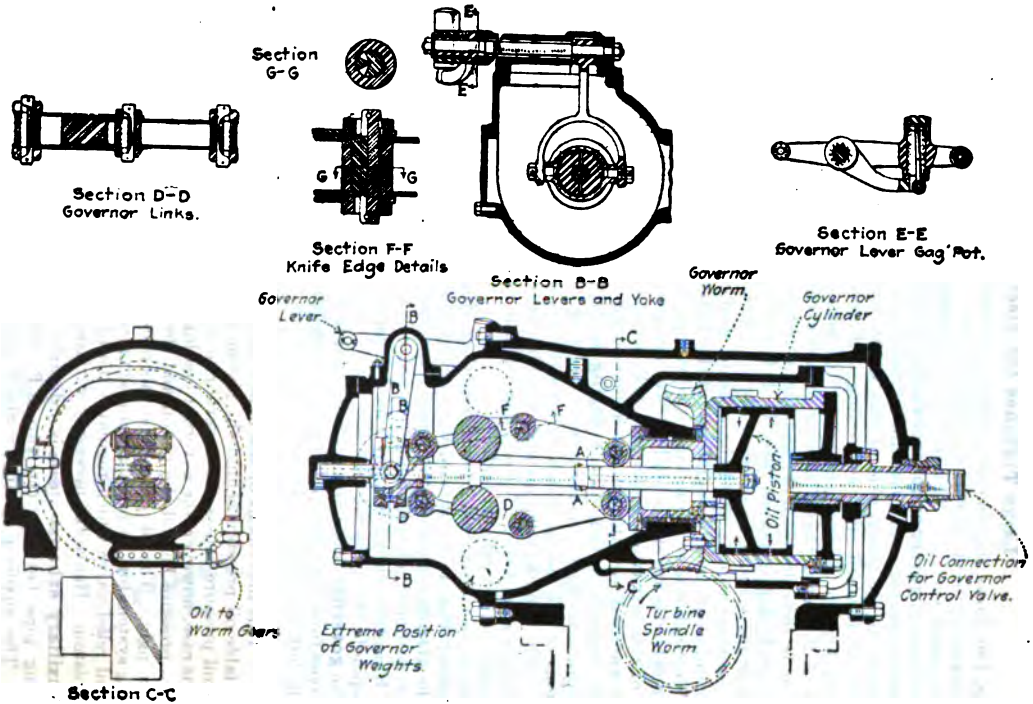


FIG. 198.—Speed Control Governor, Main Turbine, U.S.S. Tennessee.

from the oil relay cylinder of the main governor valve. This is done by means of the oil piston on the right hand side of the horizontal rod.

The valve shown in Fig. 199 is in the wide open position being held so by the hand valve. Normally the hand wheel would be at the outer limit and the valve would be resting tightly on its seat. As long as the governor valve on

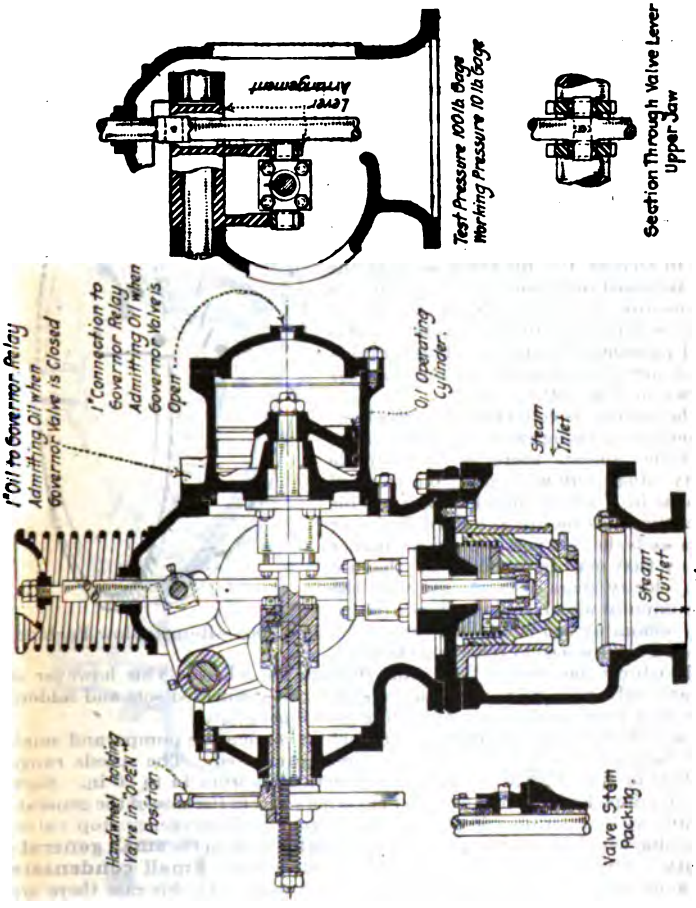

Lever Arrangement
Test Pressure 100 lb Gage
Working Pressure 10 lb Gage
Section Through Valve Lever Upper Jaw
1"Oil to Governor Relay Admitting Oil when Governor Valve is Closed
1"Connection to Governor Relay Admitting Oil when Governor Valve is Open
Oil Operating Cylinder
Steam Inlet
Steam Outlet
To Turbine
Handwheel holding Valve in "OPEN" Position
Valve Stem Packing

FIG. 199.—Auxiliary By-pass Valve, Maine Turbine, U.S.S Tennessee.

the main turbine is open an oil relay valve furnishes oil under pressure to the right hand side of the oil piston and forces it to the left. By means of the bell crank the valve is held tightly on its seat. However, when the governor cuts steam off the main turbine, the relay valve reverses the oil connection and furnishes oil under pressure to the opposite side of the oil piston which causes

it to move to the right and thus through the bell crank to open the auxiliary exhaust direct to the condenser.

Another special feature of design is the provision of a **secondary governor** to control the supply of steam to the steam seals on the glands. At full speed of the turbine the water glands are sufficient to effectively seal against vacuum. But at low speeds steam must be added to the steam seal. This is done automatically by this secondary governor.

Automatic overspeed governors and valves of standard design are used on this unit.

Small Turbines

The construction of these turbines is that known as the **re-entry type.** The turbine consists of a single wheel carrying one row of blades. A nozzle expands the steam from the inlet to the exhaust pressure and delivers it to the blades at an angle of 18 to 20 deg. A reversing chamber receives the steam after its first passage through the blades. It forms a guide to change the direction so that the steam is caused to impinge a second time on the same row of blades. Some additional energy is imparted to the wheel by this second passage of the steam. This arrangement of nozzle, rotor and reversing chamber is shown in Fig. 200.

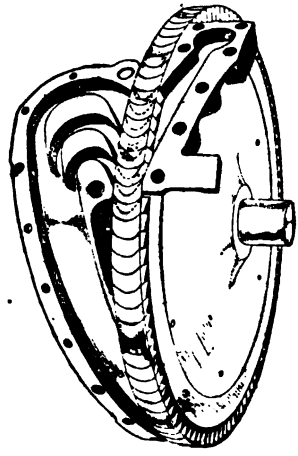


FIG. 200.—Return Flow Turbine.

As the velocity is converted into mechanical energy by passing twice over the blades, the turbine consists therefore of a simple velocity stage but with only one row of blades, as in a simple impulse unit. This construction has the advantage of the lower wheel speeds that are possible and the higher economy that may be secured by the use of the velocity stage as compared with the simple impulse wheel. Blade speeds in this design seldom exceed 600 ft. per sec.

In a few cases a second reversing chamber is used to direct the steam a third time through the wheel. This however is necessary only with relatively slow speed direct-connected sets and seldom results in a gain in economy exceeding 8 or 9 per cent.

For apparatus requiring little power, such as condensate pumps, and small electric lighting sets, the turbines are direct connected. The speeds range from 2500 to 4000 R.P.M. and the wheel diameters from 14 to 20 in. Such sets are equipped with constant speed governors and in the case of the generating units with an automatic overspeed governor and emergency stop valve. The turbine wheel is overhung from the generator shaft in the **small generating sets.** Two ball bearings carry the whole rotor. **Small condensate pumps** are usually overhung from the turbine shaft. In this case there are also but two bearings, which are babbitted and ring lubricated.

For circulating pumps which are inherently slow speed and **for larger electric generating sets,** the turbine is made small and operated at speeds ranging from 6000 to 7200 R.P.M. The speed is reduced by mechanical gearing. The turbine casing in such sets under 100 H.P. capacity is supported from the gear housing.

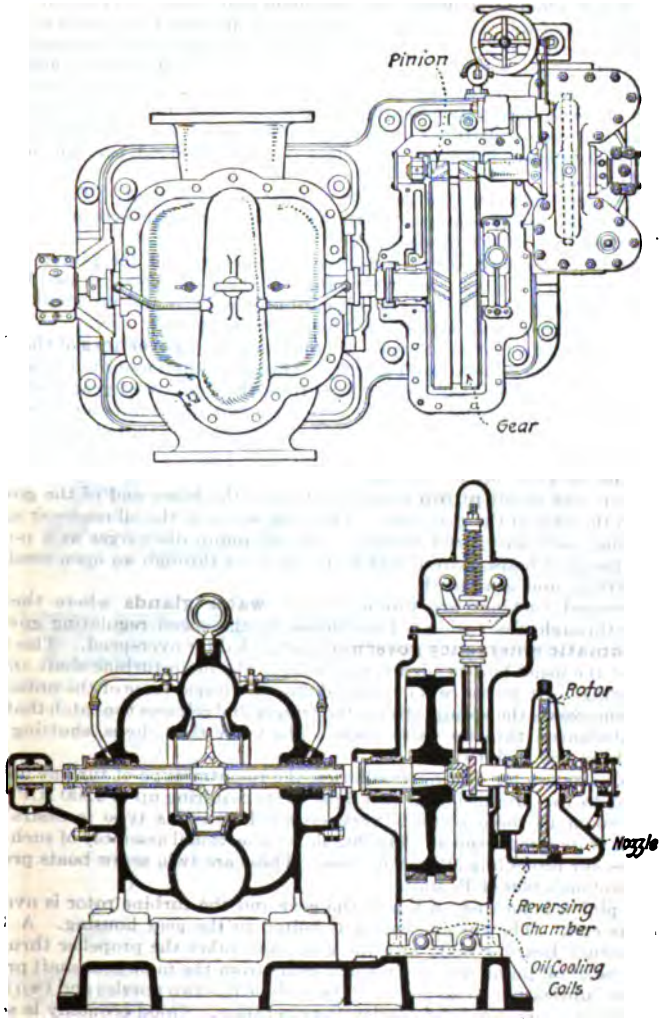


FIG. 201.—Westinghouse Geared Centrifugal Pump Drive.

The rotor is mounted directly on the pinion shaft with but two bearings, one on either end of the shaft. These bearings are provided with pads and can be adjusted relative to the casing by adding or removing shims from under these pads. The pinion can thus be brought into proper mesh with the gear. The gear wheel is overhung on an extension of the pump or generator shaft. Hence there is simply two-bearing support for this shaft as well as for the turbine and pinion shaft. Couplings are not used. Fig. 201 shows a section of one of these units driving a circulating pump of 4000 gal. per. min. against a 30 ft. head.

Larger sets do not permit of this compact arrangement. The turbine is made separate and mounted upon the bedplate. Its shaft is connected to the pinion shaft by a solid muff coupling.

Flexibility is secured by turning the extended end of the pinion shaft as small as conservative stresses will permit. The gear shaft has two bearings but the inboard pump or generator bearing is omitted and the shaft coupled solidly to the gear shaft, forming a three-bearing support.

All the geared sets of both styles are provided with **governors** of the centrifugal fly-ball type as shown on Fig. 201. This governor is mounted on a vertical shaft driven through small helical gears from the slow speed gear shaft. The governor actuates a balanced valve which controls the admission of steam to the nozzles. Some turbines are provided with hand actuated valves so that one group of nozzles may be closed off when running at light load. This improves the economy of the set.

A gear type of **oil pump** is mounted upon the lower end of the governor shaft in the base of the gear case. This case serves as the oil reservoir and the oil cooling coils are placed therein. The oil pump discharges at a point on top of the gear housing from which the oil flows through an open strainer to the bearings and gear teeth.

All geared turbines are provided with **water glands** where the shaft passes through the casing. In addition to the speed regulating governor, an **automatic emergency governor** is provided for overspeed. This governor is of the usual bolt-headed type placed in the main turbine shaft and held by a spring. At 10 per cent overspeed the centrifugal force of the unbalanced head compresses the spring, strikes the trigger and releases the catch that holds the unbalanced throttle valve open. The valve then closes, shutting off all steam from the turbine.

Besides its use for driving auxiliaries, the re-entry type of turbine has been adapted to the propulsion of fast light boats requiring up to 1200 H.P. The engine room in these boats is very crowded and this type is desirable on account of its compactness. Fig. 202 shows a sectional assembly of such a unit designed for propelling 50 ft. launches. These are twin screw boats provided with two such 600 H.P. units.

The pinion is located on top of the gear and the turbine rotor is overhung from its end. The turbine casing is bolted to the gear housing. A Kingsbury thrust bearing built into the gear case, takes the propeller thrust. A gear type of oil pump driven by spiral gears from the main gear shaft provides effective lubrication to all parts. The turbine has two nozzles and two reversing chambers arranged for condensing operation. Good economy is secured as the blade speeds are around 600 ft. per sec.

The turbine is very simply arranged for reverse operation. Astern buckets are milled in the shrouding of the blades in such a manner that the nozzles are in the plane of the wheel and discharge almost tangentially against the wheel as in several other designs described in this section. Full peripheral

admission is used with no reversing chambers, since economy when running astern is unimportant. This arrangement admits of astern operation without the complication of a separate turbine or wheel.

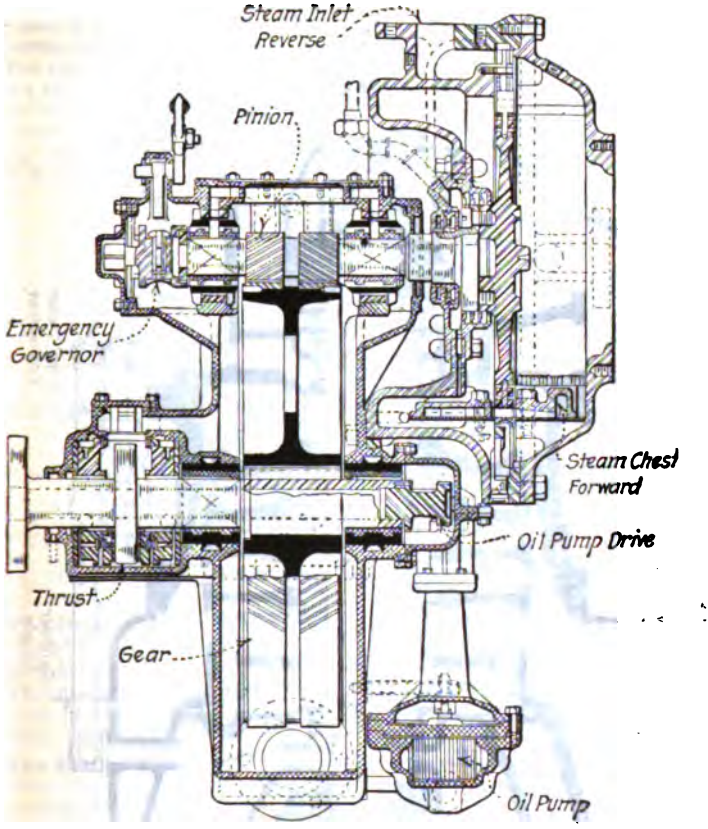


FIG. 202.—Westinghouse Marine Impulse Turbine.

In addition to the inherently light weight of the re-entry type of unit, a further reduction of weight may be obtained, if necessary, by making the gear housing and possibly some other parts of an aluminum alloy instead of cast iron or steel.

GENERAL ELECTRIC TURBINES

The General Electric Co., Schenectady, N. Y., build marine turbines for gear drive, turbo-generator equipment for electric drive and small steam turbine units for auxiliary drive.

Marine Propelling Turbines

The marine units **for gear drive** are made single cylinder up to 3000 H.P. Larger sizes have H.P. and L.P. cylinders. Fig. 203 shows a section of a single cylinder turbine known as Type "S."

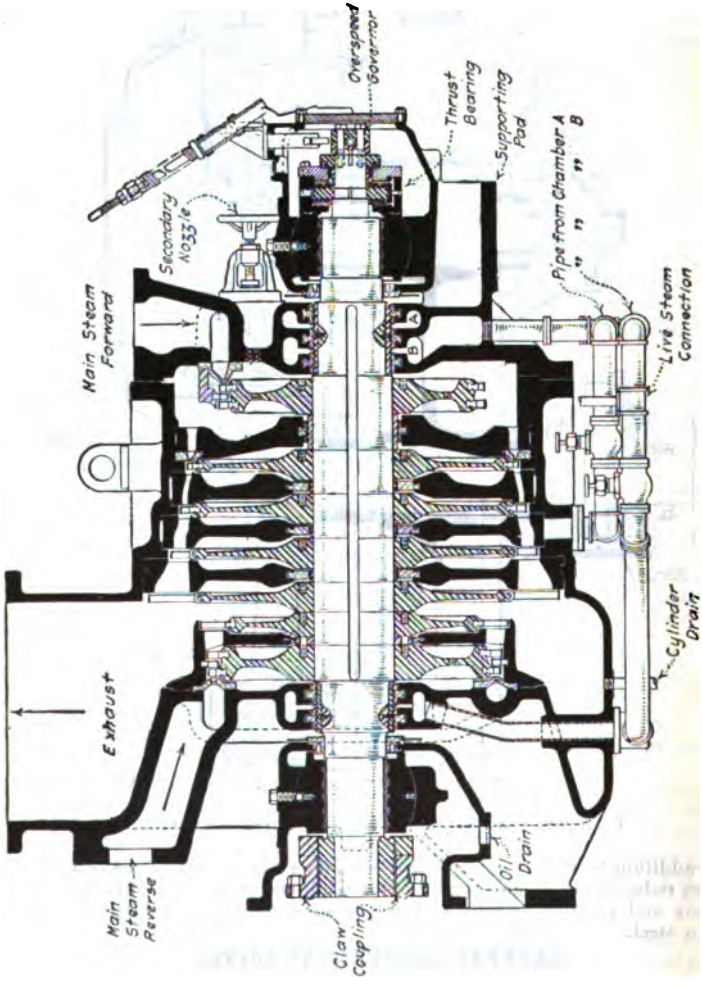


FIG. 203.—General Electric Marine Turbine.

This turbine is arranged with ahead and astern elements in the same casing. The forward section consists of one velocity stage followed by four simple

impulse stages. The astern section has one velocity stage and one simple impulse stage.

The maneuvering is done by hand controlled valves which are interlocked in such a manner that it is impossible to open the valve for the astern turbine if the valve for the ahead turbine is not closed and *vice versa*. This maneuvering gear is shown on p. 660. **Double reduction Alquist gearing** is used to reduce the turbine speed to a suitable propeller speed.

The turbine and high speed pinion are designed to operate at a full power speed of 3234 R.P.M. The high speed gears and low speed pinions have a speed of 425 R.P.M. The propeller shaft turns at 90 R.P.M. The total speed reduction ratio is 35.9 to 1.

Fig. 204 shows one of the several forms of double reduction gearing in use with this turbine.

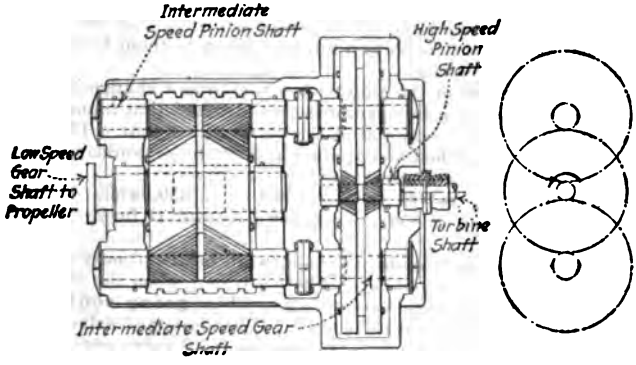

Intermediate Speed Pinion Shaft
High Speed Pinion Shaft
Low Speed Gear Shaft to Propeller
Turbine Shaft
Intermediate Speed Gear Shaft

FIG. 204.—General Electric Co.'s Double Reduction Gearing.

The turbine casing is made of cast iron with the exception of the forward head end which is cast steel when high superheat is used. The whole cylinder is split horizontally so that it can be readily inspected.

The discs are all of forged steel and are keyed onto the shaft with spacing sleeves, between the wheels where diaphragm glands are used. These sleeves are also keyed to the shaft.

The shaft is a nickel steel forging. Two keyways are cut in it, to balance strains. One of these is plugged with a feather key flush with the shaft; the other carries the keys which hold the disks.

The first stage ahead **nozzles** are made of bronze in one ring. This nozzle ring is bolted to the upper half of the turbine head. The corresponding nozzle ring astern is in halves and is bolted to the turbine shell. Steam is furnished to the astern nozzles by a cored passage in the upper half casing connecting at one end to the maneuvering valves and at the other to the nozzle ring.

The second stage ahead nozzles are made up in four segments, each bolted to the diaphragm.

Other nozzles are made by casting sheet metal partitions in the diaphragm. All **diaphragms** are split in halves and are made very heavy to withstand distortion from the high differences of pressure on their two sides.

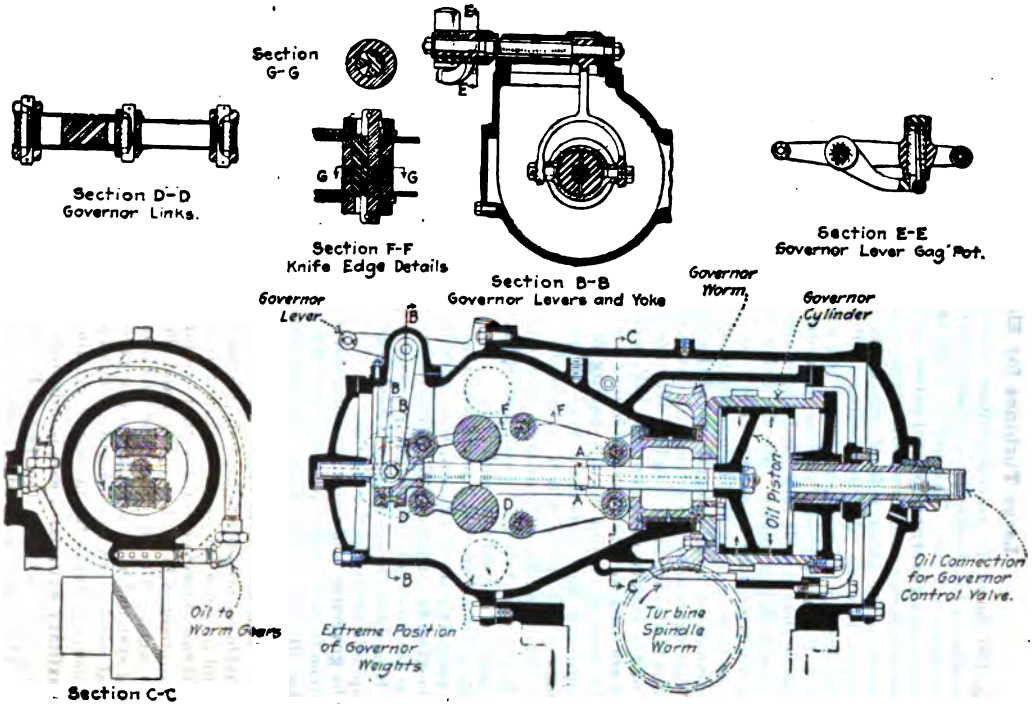


FIG. 198.—Speed Control Governor, Main Turbine, U.S.S. Tennessee.

from the oil relay cylinder of the main governor valve. This is done by means of the oil piston on the right hand side of the horizontal rod.

The valve shown in Fig. 199 is in the wide open position being held so by the hand valve. Normally the hand wheel would be at the outer limit and the valve would be resting tightly on its seat. As long as the governor valve on

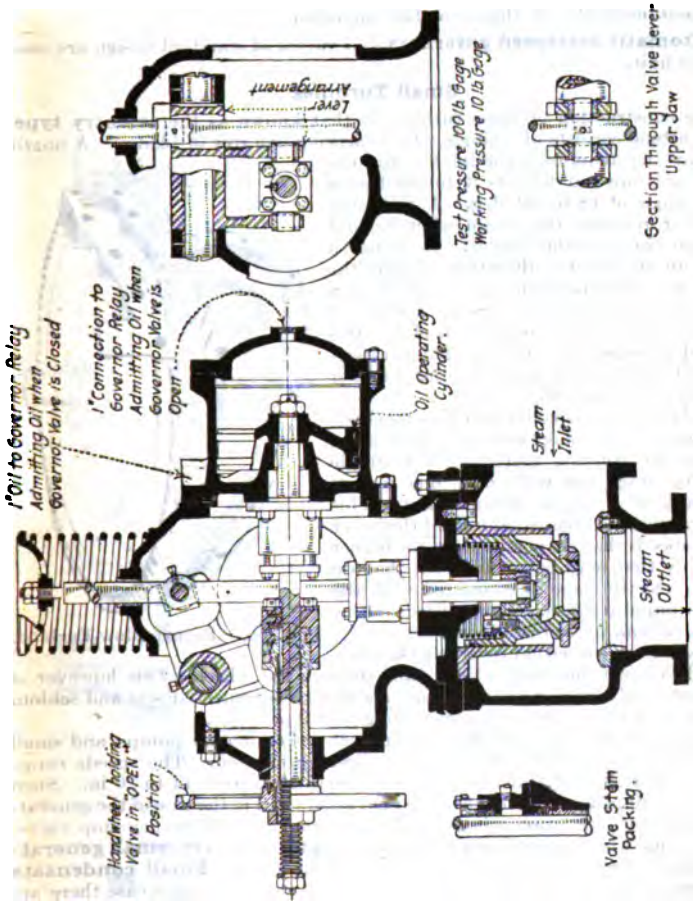

Lever Arrangement
Test Pressure 100 lb Gage
Working Pressure 10 lb Gage
Section Through Valve Lever Upper Jaw
"Oil to Governor Relay Admitting Oil when Governor Valve is Closed
"Connection to Governor Relay Admitting Oil when Governor Valve is Open
Oil Operating Cylinder
Steam Inlet
Steam Outlet
To Turbine
Handwheel holding Valve in "OPEN" Position
Valve Stem Packing

FIG. 199.—Auxiliary By-pass Valve, Maine Turbine, U.S.S Tennessee.

the main turbine is open an oil relay valve furnishes oil under pressure to the right hand side of the oil piston and forces it to the left. By means of the bell crank the valve is held tightly on its seat. However, when the governor cuts steam off the main turbine, the relay valve reverses the oil connection and furnishes oil under pressure to the opposite side of the oil piston which causes

it to move to the right and thus through the bell crank to open the auxiliary exhaust direct to the condenser.

Another special feature of design is the provision of a **secondary governor** to control the supply of steam to the steam seals on the glands. At full speed of the turbine the water glands are sufficient to effectively seal against vacuum. But at low speeds steam must be added to the steam seal. This is done automatically by this secondary governor.

Automatic overspeed governors and valves of standard design are used on this unit.

Small Turbines

The construction of these turbines is that known as the **re-entry type.** The turbine consists of a single wheel carrying one row of blades. A nozzle expands the steam from the inlet to the exhaust pressure and delivers it to the blades at an angle of 18 to 20 deg. A reversing chamber receives the steam after its first passage through the blades. It forms a guide to change the direction so that the steam is caused to impinge a second time on the same row of blades. Some additional energy is imparted to the wheel by this second passage of the steam. This arrangement of nozzle, rotor and reversing chamber is shown in Fig. 200.

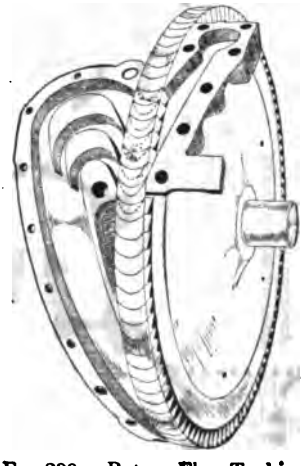


FIG. 200.—Return Flow Turbine.

As the velocity is converted into mechanical energy by passing twice over the blades, the turbine consists therefore of a simple velocity stage but with only one row of blades, as in a simple impulse unit. This construction has the advantage of the lower wheel speeds that are possible and the higher economy that may be secured by the use of the velocity stage as compared with the simple impulse wheel. Blade speeds in this design seldom exceed 600 ft. per sec.

In a few cases a second reversing chamber is used to direct the steam a third time through the wheel. This however is necessary only with relatively slow speed direct-connected sets and seldom results in a gain in economy exceeding 8 or 9 per cent.

For apparatus requiring little power, such as condensate pumps, and small electric lighting sets, the turbines are direct connected. The speeds range from 2500 to 4000 R.P.M. and the wheel diameters from 14 to 20 in. Such sets are equipped with constant speed governors and in the case of the generating units with an automatic overspeed governor and emergency stop valve. The turbine wheel is overhung from the generator shaft in the **small generating sets.** Two ball bearings carry the whole rotor. **Small condensate pumps** are usually overhung from the turbine shaft. In this case there are also but two bearings, which are babbitted and ring lubricated.

For circulating pumps which are inherently slow speed and **for larger electric generating sets,** the turbine is made small and operated at speeds ranging from 6000 to 7200 R.P.M. The speed is reduced by mechanical gearing. The turbine casing in such sets under 100 H.P. capacity is supported from the gear housing.

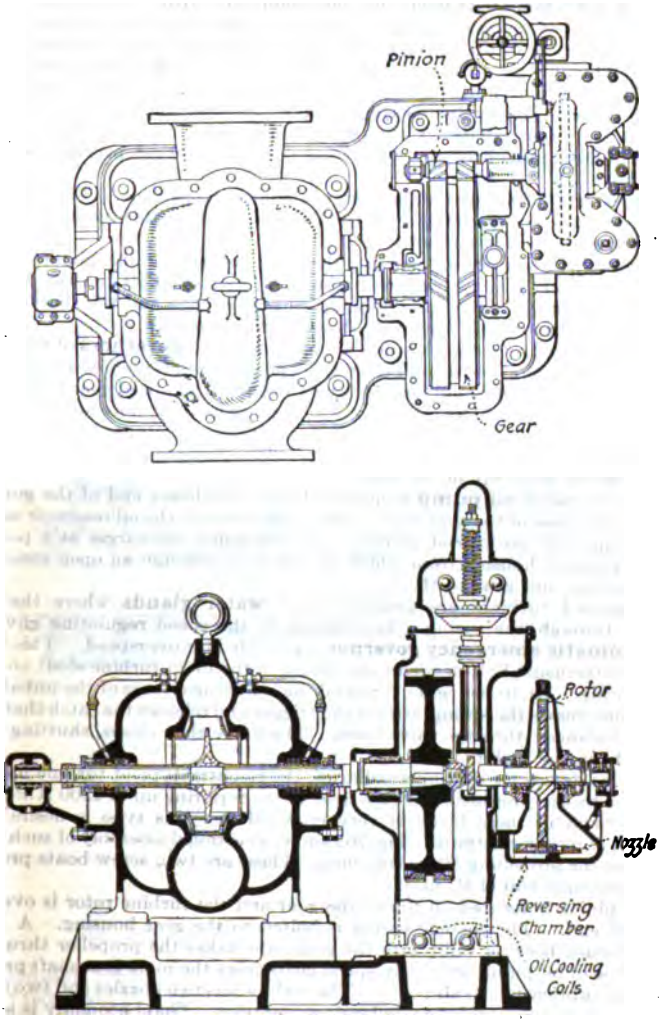


FIG. 201.—Westinghouse Geared Centrifugal Pump Drive.

The rotor is mounted directly on the pinion shaft with but two bearings, one on either end of the shaft. These bearings are provided with pads and can be adjusted relative to the casing by adding or removing shims from under these pads. The pinion can thus be brought into proper mesh with the gear. The gear wheel is overhung on an extension of the pump or generator shaft. Hence there is simply two-bearing support for this shaft as well as for the turbine and pinion shaft. Couplings are not used. Fig. 201 shows a section of one of these units driving a circulating pump of 4000 gal. per. min. against a 30 ft. head.

Larger sets do not permit of this compact arrangement. The turbine is made separate and mounted upon the bedplate. Its shaft is connected to the pinion shaft by a solid muff coupling.

Flexibility is secured by turning the extended end of the pinion shaft as small as conservative stresses will permit. The gear shaft has two bearings but the inboard pump or generator bearing is omitted and the shaft coupled solidly to the gear shaft, forming a three-bearing support.

All the geared sets of both styles are provided with **governors** of the centrifugal fly-ball type as shown on Fig. 201. This governor is mounted on a vertical shaft driven through small helical gears from the slow speed gear shaft. The governor actuates a balanced valve which controls the admission of steam to the nozzles. Some turbines are provided with hand actuated valves so that one group of nozzles may be closed off when running at light load. This improves the economy of the set.

A gear type of **oil pump** is mounted upon the lower end of the governor shaft in the base of the gear case. This case serves as the oil reservoir and the oil cooling coils are placed therein. The oil pump discharges at a point on top of the gear housing from which the oil flows through an open strainer to the bearings and gear teeth.

All geared turbines are provided with **water glands** where the shaft passes through the casing. In addition to the speed regulating governor, an **automatic emergency governor** is provided for overspeed. This governor is of the usual bolt-headed type placed in the main turbine shaft and held by a spring. At 10 per cent overspeed the centrifugal force of the unbalanced head compresses the spring, strikes the trigger and releases the catch that holds the unbalanced throttle valve open. The valve then closes, shutting off all steam from the turbine.

Besides its use for driving auxiliaries, the re-entry type of turbine has been adapted to the propulsion of fast light boats requiring up to 1200 H.P. The engine room in these boats is very crowded and this type is desirable on account of its compactness. Fig. 202 shows a sectional assembly of such a unit designed for propelling 50 ft. launches. These are twin screw boats provided with two such 600 H.P. units.

The pinion is located on top of the gear and the turbine rotor is overhung from its end. The turbine casing is bolted to the gear housing. A Kingsbury thrust bearing built into the gear case, takes the propeller thrust. A gear type of oil pump driven by spiral gears from the main gear shaft provides effective lubrication to all parts. The turbine has two nozzles and two reversing chambers arranged for condensing operation. Good economy is secured as the blade speeds are around 600 ft. per sec.

The turbine is very simply arranged for reverse operation. Astern buckets are milled in the shrouding of the blades in such a manner that the nozzles are in the plane of the wheel and discharge almost tangentially against the wheel as in several other designs described in this section. Full peripheral

admission is used with no reversing chambers, since economy when running astern is unimportant. This arrangement admits of astern operation without the complication of a separate turbine or wheel.

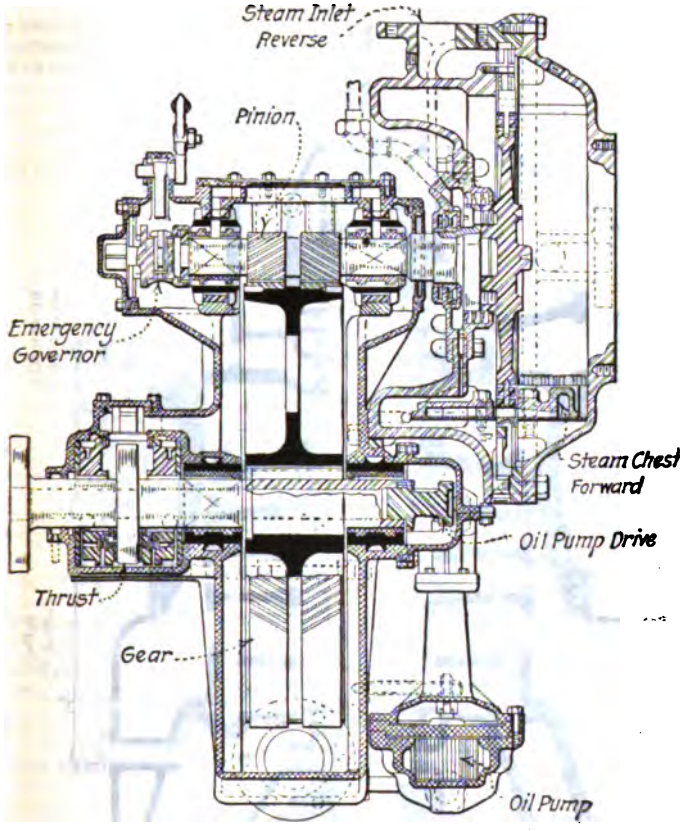


FIG. 202.—Westinghouse Marine Impulse Turbine.

In addition to the inherently light weight of the re-entry type of unit, a further reduction of weight may be obtained, if necessary, by making the gear housing and possibly some other parts of an aluminum alloy instead of cast iron or steel.

GENERAL ELECTRIC TURBINES

The General Electric Co., Schenectady, N. Y., build marine turbines for gear drive, turbo-generator equipment for electric drive and small steam turbine units for auxiliary drive.

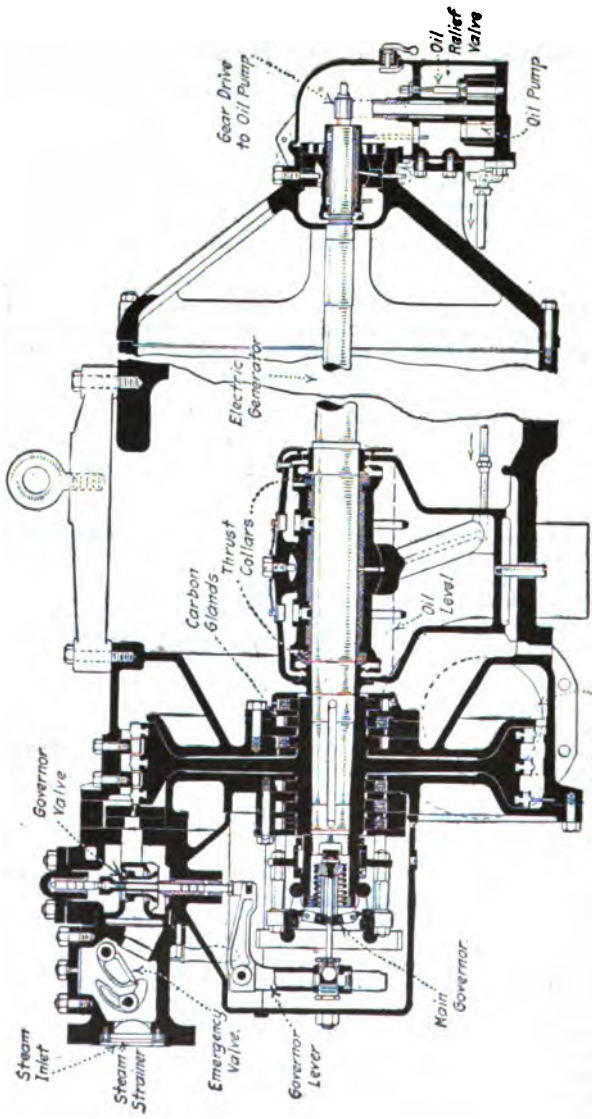


FIG. 208.—General Electric Co. Small Turbo-generator Set.

turn operates the valve stem, admitting steam through the poppet valve to the steam chest.

Two **emergency governors** of the clock spring type are mounted on studs on the outer ring of the main governor. These springs rest under tension against stops, and, when the turbine reaches about 11 per cent overspeed, the free end of the clock spring emergency governor flies outward, tripping the trigger and allowing the spring to trip the mechanism which in turn closes the **"Flap" valve.** The valve mechanism is assembled and mounted in a separate chest, the valve being suspended loosely on the resetting shaft. The valve is normally held open by a latch which is keyed to the tripping shaft. This tripping lever, shaft and latch are normally held in position by a square rod when engaged by the trigger.

The **oil pump** for supplying oil to both bearings is located at the outboard end of the generator shaft and is operated by the shaft through a worm and gear. The casing which houses the pump also acts as the oil reservoir.

The **packing** on the condensing unit consists of carbon rings which operate without lubrication.

Each ring is made in three sections to insure a good fit and to allow the wear to be taken up automatically. Three rings are used on each side of the wheel casing, each ring being carried in a circular cell of close clearance. The segments are held together by three alloy holders fitted with flat springs. Each carbon segment is provided with a keyway engaging with which is a key riveted to the alloy holder. The outer lug on the alloy holder engages a lug which is tapped rigidly into the cell, thus preventing the carbon rings from revolving with the shaft. Live steam, throttled to about 5 to 10 lbs. gage by means of a globe valve, is supplied to the packing and forms a steam seal.

Soft metallic packing rings, carried in adjustable glands, which are fastened to the wheel casing, are used on non-condensing units. These glands may be repacked in place without disassembling any part of the unit.

DELAVAL STEAM TURBINES

The DeLaval Steam Turbine Co. of Trenton, N. J., manufacture two general classes of turbines for marine use, (*a*) small single stage units either geared or direct connected and (*b*) marine propelling geared sets.

Small Single Stage Units

The geared turbines usually have a single Curtis wheel but in some small sizes a single impulse wheel is used in which case the turbine runs at high speed. These latter machines are the original well-known DeLaval type and as the marine turbine will illustrate the use of gears in connection with turbines, the following description applies to the construction of direct-connected turbines.

The Curtis stage may have two or three rows of blades. Fig. 209 shows a cross-section of the smallest size where the nozzle, revolving blades, reversing blades, etc., are readily seen. These turbines are usually operated non-condensing.

The casing is ordinarily of cast iron, but cast steel is used when the steam chest is integral with the cover and high pressure and superheat are used. If the steam chest is separate from the cover, it alone is of steel. The casing is split horizontally.

Hand operated valves are usually located in the steam chest to allow the power to be varied and for use with changed operating conditions. These valves close off certain nozzles. Steam enters the lower half of the casing

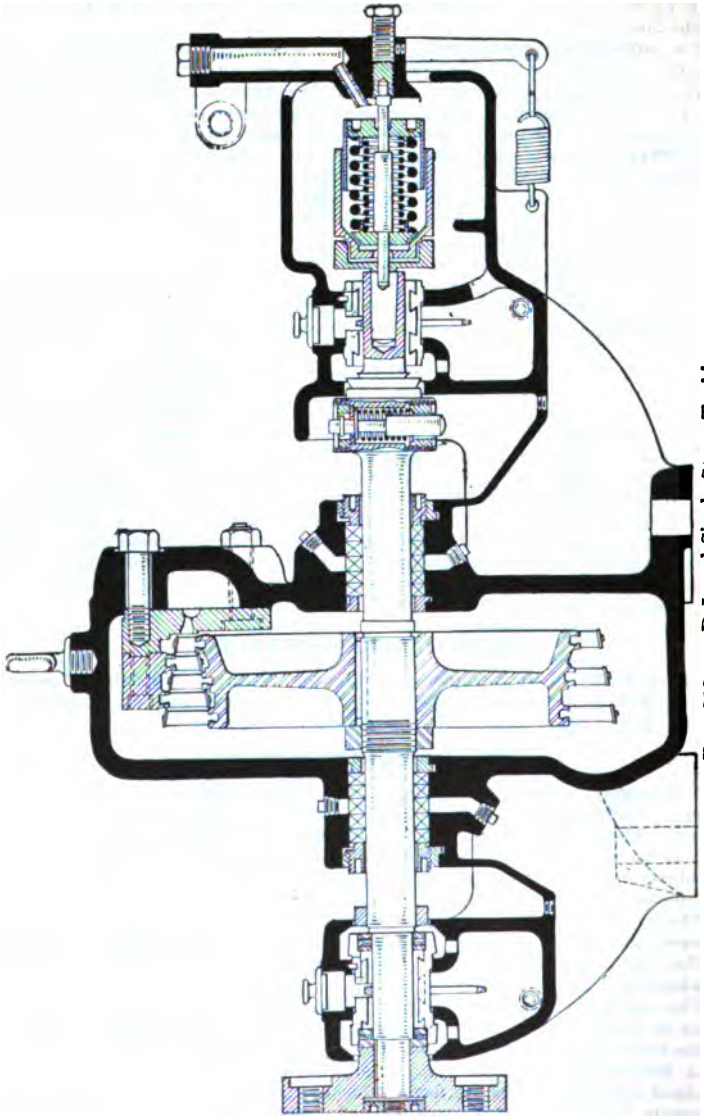


FIG. 209.—DeLaval Single Stage Turbine.

unless high superheat makes it advisable to locate such parts above the bearings. The exhaust is in the lower half and does not have to be disturbed in making examinations of the machine.

The casing rests on the bed plate and carries the bearings on brackets. The bearing covers can be readily removed for examination of the bearings. These are babbitt lined, reamed and scraped to a perfect fit on both shaft and pedestal. They are lubricated by oil rings. Two bearings are employed on all machines.

The nozzles are of the diverging type reamed in a removable bronze ring or plate. The rotor consists of a large diameter shaft of open hearth steel

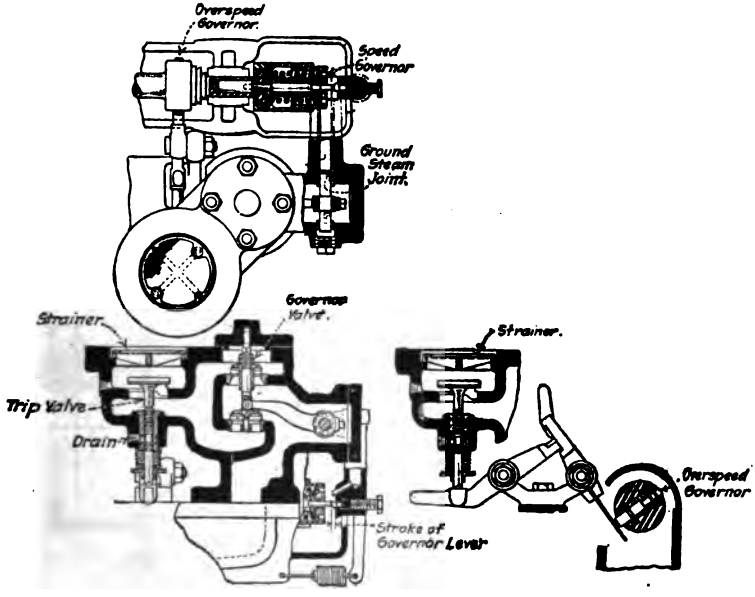


FIG. 210.—DeLaval Governor Valve and Overspeed Governor for Small Turbines.

upon which is keyed a single forged steel disc carrying the bucket rows upon the rim. The buckets are inserted in dovetailed slots and are held apart by distance pieces. The blades are milled from extruded bars or are drop forgings.

The governing arrangement is shown in Fig. 210. The regulating governor is of the centrifugal type and is mounted on the end of the shaft. It acts through levers upon a double seated balanced valve.

The emergency governor, entirely independent of the main governor, consists of a bolt headed weight in the shaft which at 10 per cent over normal speed compresses the resisting spring, moves out and trips the emergency valve, thus cutting off the supply of steam.

The glands are only subjected to exhaust pressure. For back pressures of 10 lbs. per sq. in. or more, or for vacuum a series of carbon rings is furnished

in cellular stuffing boxes which are split in a horizontal plane along the centre line. For less exacting service, metallic packing is furnished, as shown in Fig. 209.

These turbines are suitable for driving small alternating and direct-current generators, pumps for various services and high speed blowers. They are extensively used with boiler feed pumps where exhaust steam is used to heat boiler feed water.

Marine Propeling Sets

Turbines for this service are of the multistage impulse type and are designed to drive the propellers through speed reducing gears. Both single cylinder and compound turbines are built, depending upon the service, speed of turbine and propeller, power, etc. Such points also determine whether single or double reduction gears are used.

The following illustrations cover a compound turbine of 2500 S.H.P. at 2400 R.P.M. having a single reduction gear that drives the propeller at 110 R.P.M. Fig. 211 shows the arrangement of the turbines on a twin screw vessel. Cross-sections of the high and low pressure cylinders are shown in Figs. 212 and 213.

The first stage in the high pressure cylinder and both stages of the astern element are of the Curtis type. The first stage nozzles both ahead and astern are reamed in steel nozzle rings bolted to the steam chest. Hand controlled valves are furnished on these steam chests so that a number of

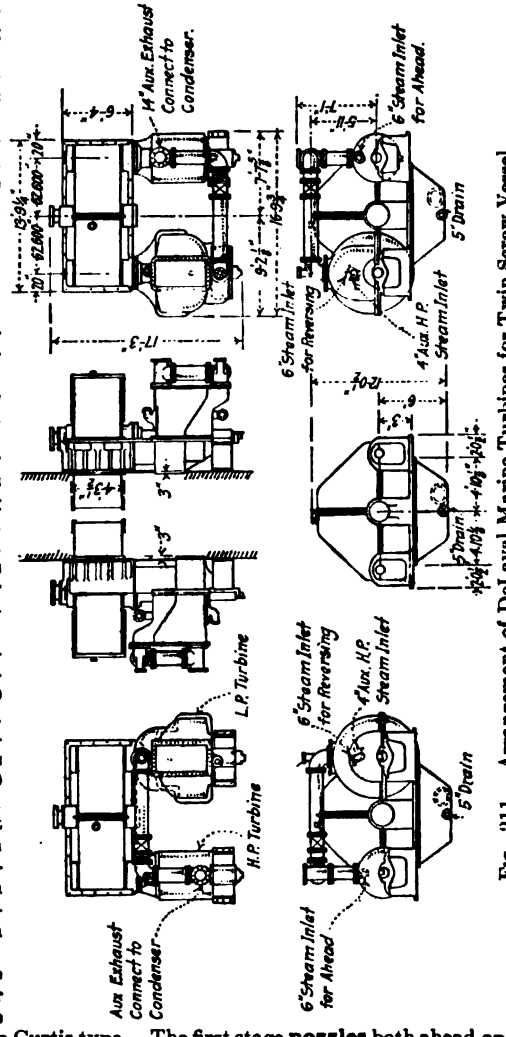

14" Aux. Exhaust Connect to Condenser.
6" Steam Inlet for Ahead.
5" Drain
6" Steam Inlet for Reversing
4" Aux. H.P. Steam Inlet
L.P. Turbine
H.P. Turbine
Aux. Exhaust Connect to Condenser
6" Steam Inlet for Ahead

FIG. 211.—Arrangement of DeLaval Marine Turbines for Twin Screw Vessel.

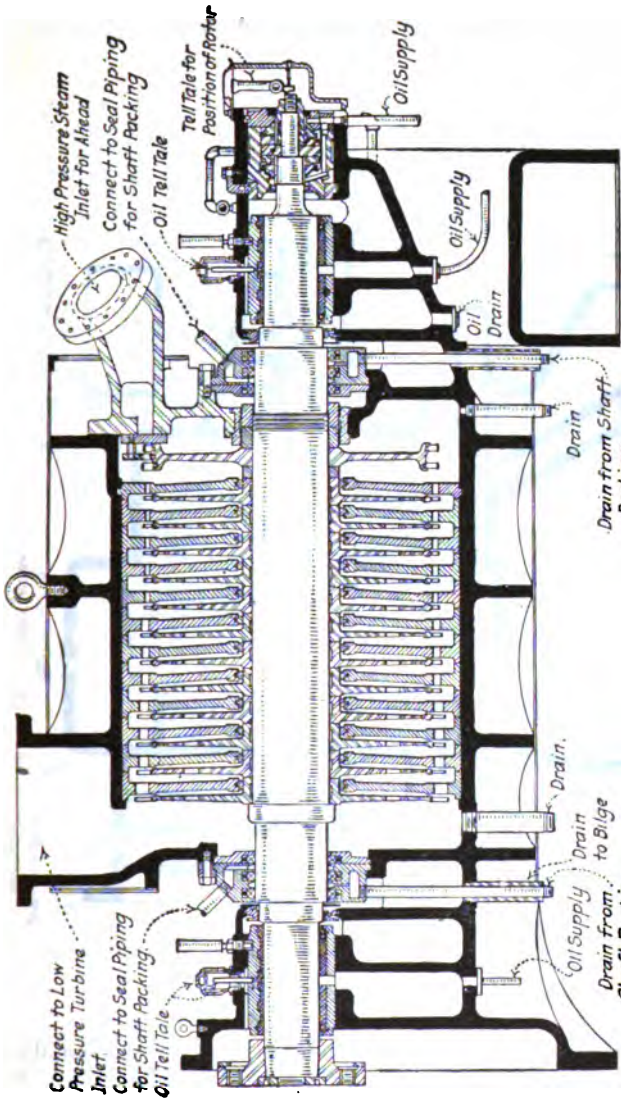


FIG. 212.—DeLaval Marine High Pressure Turbine.

nozzles may be closed if desired. The steam chests are of cast steel bolted to the upper half of the casing. Nozzles of both circular and square cross-sections are used.

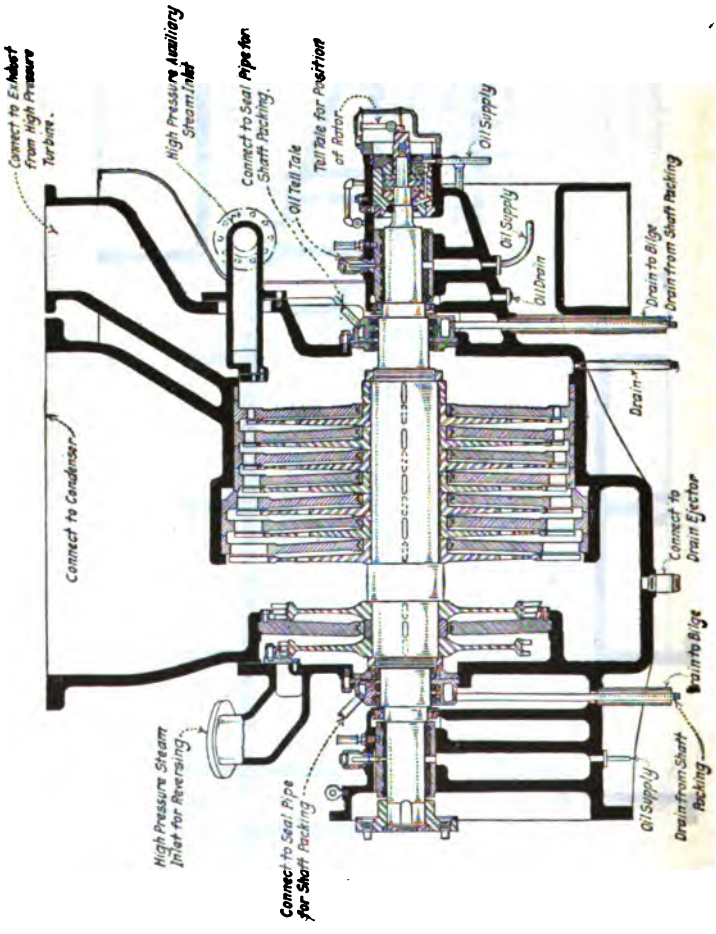


FIG. 213.—DeLaval Marine Low Pressure Turbine.

The diaphragms are of cast iron with sheet metal vanes to form the orifices cast in place. These diaphragms are split on the center line and are not firmly fastened to either half of the casing. Split labyrinth packings of a bronze or aluminum alloy are firmly fastened to the inner bore of these diaphragms.

The blades are of the DeLaval, drop forged, bulb-shank type, made of steel or monel metal. Forged lips on the outer end of each blade form an outer boundary for the steam jet in place of a shroud. These blades can be replaced individually if damaged.

The minimum axial clearance between rotating and stationary parts is 0.10 in. The mean blade speed is about 600 ft. per sec.

The casing is of cast iron or semi-steel split on the horizontal centre line and with the exhaust opening in the cover. One end of the turbine is supported by the gear case to which it is bolted, while the other is supported by a pedestal. Brackets integral with the casing support the bearings. The casings are provided with suitable drains to keep them free of water.

The shafts are solid steel forgings machined and ground on all surfaces. They are so designed that the critical speed is well above the operating speed.

The wheels are heat treated mild steel forgings. They are mounted directly upon the shaft secured to it by means of keys, and held in position by shoulders and lock nuts.

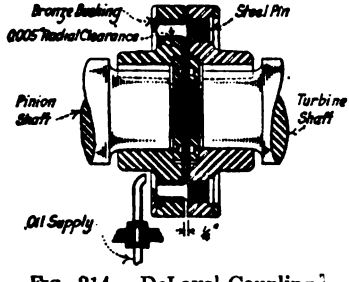


FIG. 214.—DeLaval Coupling.

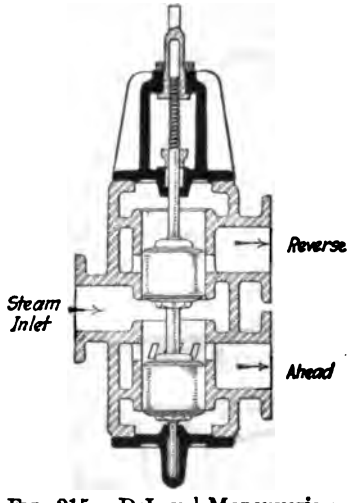


FIG. 215.—DeLaval Maneuvering Valve.

The bearing housings are supported by the casing brackets. They are split and contain split cast iron shells, babbitt lined. The bore of the bearing is 0.015 in. larger than the shaft diameter. The shells are ground upon the outer surface and the straight seats are scraped to secure an accurate fit. Axial alignment is maintained by a thrust bearing of the Kingsbury type. All bearings and thrust bearing have forced feed lubrication from the main oiling system which supplies the gears and all also have tell-tale flow indicators.

Carbon stuffing boxes as shown in Fig. 97 are provided where the turbine shafts pass through the casing. The upper half of the box can be removed for examination and as the carbon is in segments, all parts are accessible. At the high pressure packing there is, in addition to the carbon, **a labyrinth packing** shown between the wheel and stuffing box in Fig. 212. The leakage from the high pressure gland is used to seal the low pressure stuffing box against air leakage into vacuum.

The couplings between turbines and gears are of the steel pin type which fit in bronze bushings carried by the driven half. There is a clearance of

0.005 in. between the reamed hole in the bronze bushing and the steel pins. The coupling must be well oiled. The general arrangement and provision for lubrication is shown in Fig. 214. A rigid coupling is used between the gear shaft and the propeller shaft.

The maneuvering valve for controlling the steam supply to the ahead or reversing turbine is shown in Fig. 215. It consists of a steel valve body having one inlet and two outlets, one to the ahead turbine and the other to the astern element. It is impossible with this arrangement of sliding pistons to feed steam to the ahead and astern elements of the turbine at the same time.

An emergency governor is provided to trip a quick closing valve on the steam supply when the speed becomes 10 per cent above normal, and thus shut the turbine down. **A speed regulating governor** driven from the slow speed gear shaft is furnished on many turbines.

The oiling system is made up of galvanized pipe and fittings. Great care is taken to thoroughly clean the system before starting up.

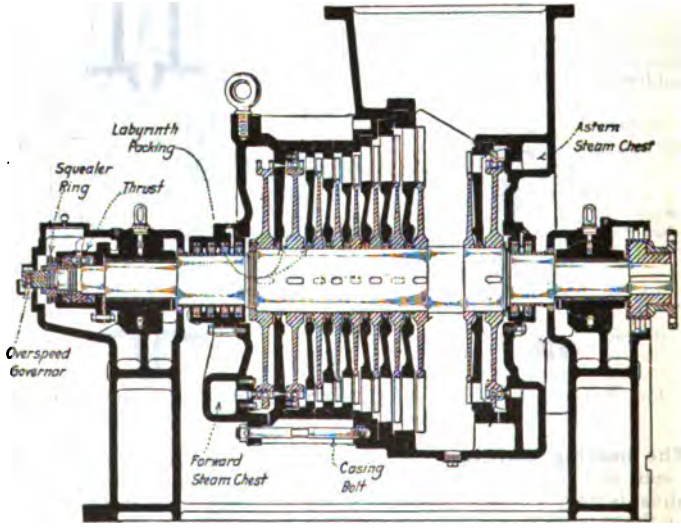


FIG. 216.—Kerr Single Cylinder Marine Turbine.

KERR TURBINES

The Kerr Turbine Co., Wellsville, N. Y., build turbines for marine geared drive, for turbo-generator sets and in small units for auxiliary service.

Fig. 216 shows a **single cylinder geared** marine turbine design for powers up to 3000 S.H.P. In this design a large number of stages is used at a low peripheral speed. The normal blade speed does not exceed 500 ft. per sec. The designers claim that the large number of stages eliminates erosion of blading and makes the machine safer to operate.

The first two H.P. **stages** of the forward section are of the Curtis type; the remaining stages are of the Rateau type. The astern section consists of one Curtis and one Rateau stage.

The cylinder consists of a steam end casting to which is bolted the ahead nozzle segment, exhaust end casting to which is bolted astern nozzle ring and a number of diaphragms between these two castings.

Diaphragms. Astern diaphragms and the last three ahead diaphragms are set in grooves in the exhaust end casting. The remaining diaphragms are set between the steam and exhaust ends, accurately grooved with them and with each other by turned and bored tongue and groove joints as shown in Fig. 216. Diaphragms that are between steam and exhaust ends thus form a portion of the cylinder proper. Outside the tongue and groove, the diaphragms are made tight by an accurately faced metal-to-metal joint. Asbestos wick is laid in the diaphragm grooves before drawing up tight. The assembled unit

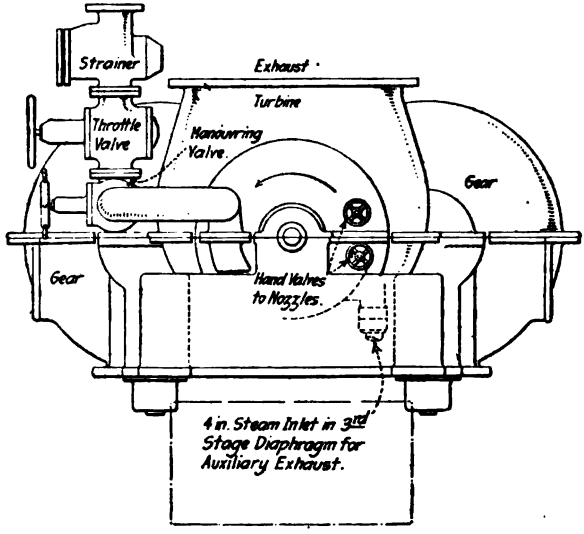


FIG. 217.—End View Kerr Geared Marine Turbine.

is held tight together by a series of tie bolts extending from steam end to exhaust end castings.

All diaphragms, steam and exhaust castings are split so that the turbine can be inspected by lifting the upper half. Lifting of the upper casing is facilitated by four threaded guide pins. Feet are cast on the bottom half to support the casing. These feet are placed close to the central plane of the shaft to minimise heat expansion. The feet rest on a base containing pedestal bearings, thus preserving accurate alignment, as shown in Fig. 217. These pedestal bearings do not form a part of the turbine casing. It is claimed that they are thus kept cooler than the usual bearings cast integral with the cylinder.

The shaft is a single forging, turned and tapered where discs are pressed on. The **discs** which are turned from solid forgings of high grade chrome nickel steel and are heat treated, are held from turning by keys placed on opposite sides of the shaft for every alternate disc to improve the balance of the rotor,

They are held apart by steel spacers. The discs on the forward end are held in place by a locked nut. The astern discs are held by a split and a shrink ring.

The blading of the Curtis stages is of extruded metal inserted in a circular groove turned on the rim of the disc. Spacers are provided of the same material. The outer ends of the blades are stiffened by a one-piece **shroud ring** fastened to the blades by curling the thin end of their outer ends.

The L.P. blades are drop forged of high grade nickel steel and heat treated. Each blade is forged in one piece with its bulb shaped base and an inner and an outer projection. The bulb shaped base fits tight in a groove of similar form milled in rim of discs, and is riveted over to prevent end movement of bucket. Inner projections that are forged on serve as a boundary for the steam passage and are spaced to be $\frac{1}{64}$ in. apart to prevent binding of buckets at this point. Outer projections butt against each other tightly and are tongued and grooved to prevent side slipping, and form thus a rigid outer shroud. Long buckets on the L.P. end are reinforced near their bases to enable them to withstand vibration and shock from sudden reversal.

Nozzles. The H.P. nozzles in the first stage both forward and astern are made of special cast iron. The sheet metal partitions in the diaphragms are made of $3\frac{1}{2}$ per cent nickel steel.

The glands consist of carbon packing rings. These rings are made in three segments and are tightly held by a coiled garter spring around their circumference. The joints of the segments are fitted so that the shaft clears the ring by about 0.006 in. The leakage from the gland on the forward end is carried to the gland on the astern end and seals it against vacuum.

The thrust bearing consists of a single thrust collar held to the shaft by a nut and with bearings on each side of the general type of the Kingsbury thrust bearing described on p. 638.

The Kerr **turning gear** is described on p. 646.

The bearings are made with spherically seated cast iron shells, babbitt lined. The lower halves of the two bearings carry an emergency water cooling coil. An oil sight flow device is placed above each bearing to indicate the flow of oil and carry away air that may accumulate in the oil.

A coupling of claw type is used to connect the turbine and pinion shafts. It allows $\frac{1}{8}$ in. axial movement as well as a slight radial movement.

The emergency overspeed governor is shown in Fig. 127. It is of the steam relay type and is set to operate at 15 per cent overspeed. When the governor acts and the lever opens the small steam relay valve, steam is exhausted from below the piston of the emergency valve shown in Fig. 128. The valve is now unbalanced and is closed by the steam pressure above the piston (see p. 658).

Interlocking duplex maneuvering valves are furnished with these turbines. A guarding valve is placed between the maneuvering valve and the astern nozzles. This is closed during the voyage to absolutely prevent leakage through the astern nozzles.

The turbine is provided on the forward end of the casing with two hand valves. These open additional nozzles when the steam pressure is low or the vacuum poor, but should be closed under normal conditions.

Normal speeds of single cylinder Kerr marine geared turbines that are usually provided with double reduction gearing are as follows:

700 H.P. and below	6000 R.P.M.
800–1500 H.P.	4500 R.P.M.
1600–3000 H.P.	3600 R.P.M.

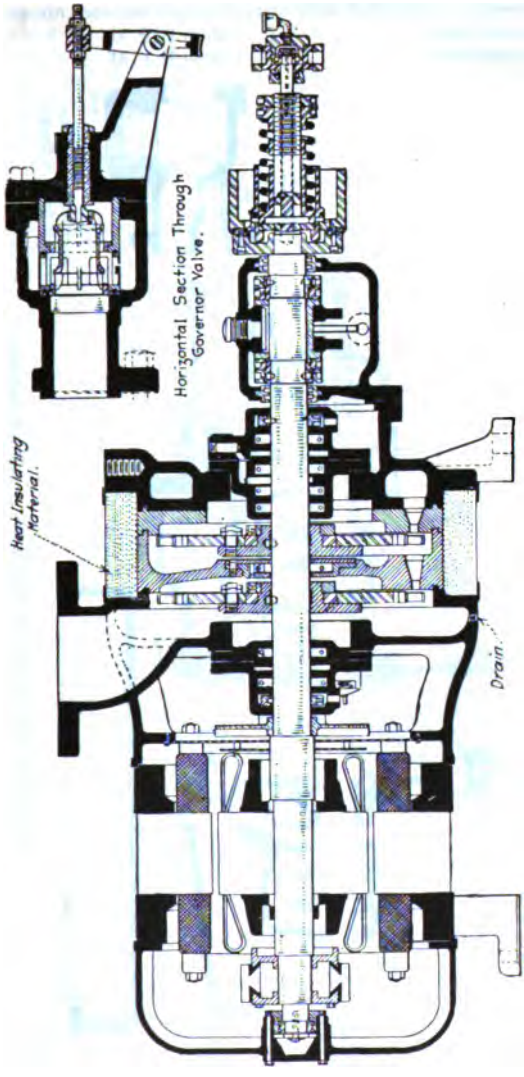


Fig. 218.—Kerr Marine Electric Lighting Set.

Normal speeds of compound Kerr marine geared turbines are as follows:

2500 to 3500 H.P.	3600 R.P.M.
4000 to 6000 H.P.	3000 R.P.M.

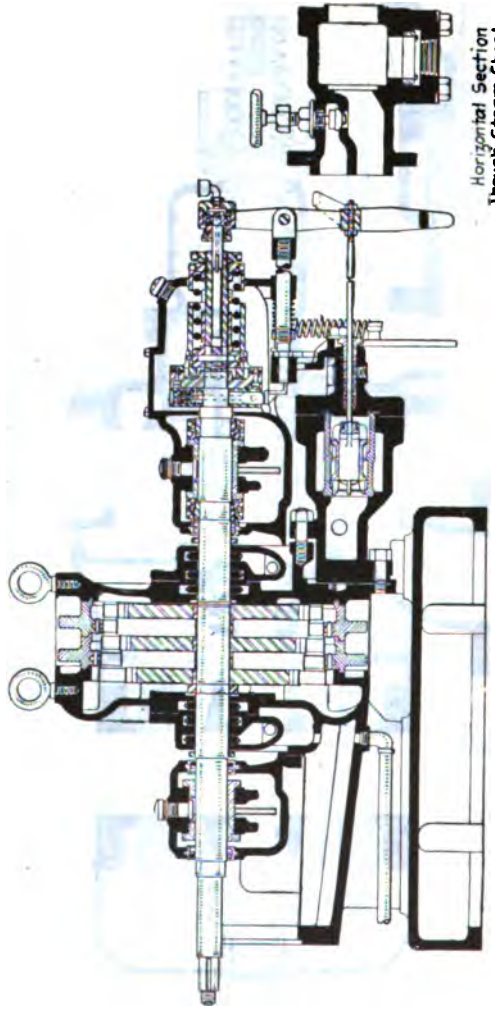


FIG. 219.—Small Kerr Turbine for Auxiliary Service.

Compound machines are usually connected through single reduction gearing of an approximate ratio 1 : 20 to propeller, although in exceptional cases double reduction gearing is applied.

Kerr turbines are also built **for electric drive** both in small and medium sizes. The discs are bolted to three piece hubs fitted to the shaft and kept from endwise or rotary displacement by pin keys. A split mitered ring is forced by a locknut into the bore of the disc and against turned tapers. This provides a true and rigid construction and allows the quick removal of discs from shaft.

The bearings are furnished with oil rings for emergency service, although oil is circulated by an oil pump at the bottom of the governor shaft.

The turbine is governed by an oil relay type of governor. An emergency governor is also provided.

The glands are of the same type as in the marine set.

Fig. 218 shows a **small 10–15 kw lighting set** that has been used largely on shipboard. A horizontal section of the governing valve is also shown. The unit is of very simple construction designed to run non-condensing. It consists of two simple impulse stages. The casing is split vertically as in the regular electric types.

Fig. 219 shows a **small single three row** velocity stage turbine built for small pump services such as condensate pumps, etc. The unit is extremely simple in construction. The bearings are furnished with ring oiling only. The Kerr form of vertical split casing is used on this unit also.

POOLE MARINE TURBINES

Marine Propelling Turbines

The Poole Engineering and Machine Co., of Baltimore, have designed and built marine turbines of the impulse type in units of 1500 and 2500 S.H.P. A considerable number of these turbines were used with Poole reduction gears on the "Eagle" boats. The turbine conforms with good standard design and has no radical departures from recognised practice. Its builders claim high steam economy and reliability for these units.

Fig. 220 shows one of these units with an 8.4 to 1 single reduction gear. The turbine consists of a 2-row velocity stage followed by five simple impulse stages on the forward section with a 2-row velocity stage for astern service. The normal speed ahead is 4300 R.P.M. at 2500 H.P. The **casing** is of cast iron split horizontally with the exhaust opening on top. A drain is provided in the lower half of the exhaust section.

The shaft is a forging of medium carbon steel. The central portion is turned straight and keyways are cut alternatively on opposite sides of the shaft to hold the discs in place, one key to a disc. The discs themselves fit the shaft loosely, but are made tight by being pressed hydraulically over slightly tapered split brass rings placed at either end of the hub. The brass, being in compression, takes care of any increase in hub diameter due to centrifugal forces.

The discs are of forged steel and carry the blades in tee-shaped slots on the outer periphery. They have holes provided to equalise the pressure on the two sides of the disc. The hubs of these discs form the inner surface of the diaphragm packing.

The blades are all machined from rolled bars of chrome-vanadium steel which has an elastic limit of 75,000 lbs. and an ultimate strength of 125,000 lbs. Metal is left in machining the base to form the distance piece between blades. A shoulder is also left at the tip which projects over to and touches the next blade, thus forming an outer boundary to the steam channel in place of a shroud ring. As the back curve of this shoulder conforms with the

front curve of the blade, these ends interlock and, to a considerable measure, prevent vibration. The **normal blade speeds** on these units have not exceeded 600 ft. per sec., which is a conservative limit.

One unusual feature is the enclosure of the **idle blade segments** of the velocity stages, both forward and astern, by a casing which tends to greatly reduce the windage loss of these blades and also decreases the steam consumption of the unit. This windage loss is ordinarily of considerable magnitude in the first stage.

The steam nozzles are of semi-steel. Both forward and astern nozzles are placed in the bottom half of the casing. Inspection of the turbine can be made by lifting the cover without disturbing the main steam connections.

The diaphragms are of cast iron with chrome-vanadium partitions cast in place to form the orifices. The labyrinth gland at the shaft consists of a bronze sleeve surrounding the shaft. It is halved, one half being fastened firmly to each half of the diaphragm. A fine V-shaped thread is cut in this sleeve and forms the necessary baffling to prevent the passage of the steam. Tests indicate that this design is very effective in preventing leakage.

The glands are made with similar V-thread labyrinth packing. The leakage from the high pressure gland is carried by ports and piping to the low pressure gland to seal it against vacuum.

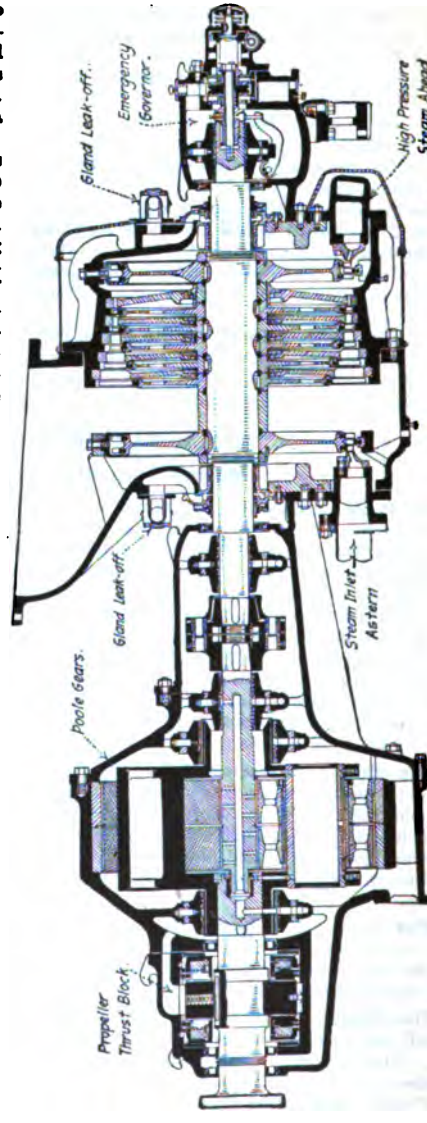


FIG. 220.—Poole Marine Geared Turbine 2500 S.H.P.

The bearings consist of spherically seated cast steel shells lined with babbitt and amply lubricated with oil which enters through the supporting base. Two open ended grooves are cut clear across the bottom half of the bearing about 30 deg. below the horizontal split on each side. These receive oil from a channel running all around the bearing and being open-ended rapidly flush out any hot oil carried around by the shaft. The upper halves are held from turning by a dowel pin.

Any end thrust is taken up by a thrust collar fastened to an extension of the shaft. This collar runs between two babbitted self-adjusting spherically seated stationary thrust rings which may also be adjustable axially by means of liners. An overspeed governor is fitted in the shaft extension. It consists of an unbalanced circular weight slipped over the shaft and opposed up to 10 per cent above normal speed by the compression of a spring. When the weight acts it strikes a trigger which releases the stop valve and closes it instantly, thereby cutting off the steam supply.

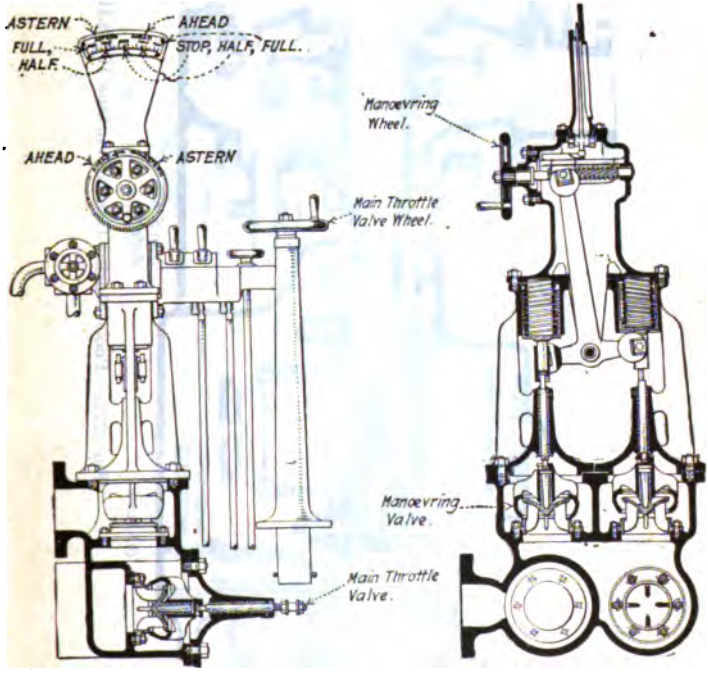


FIG. 221.—Poole Maneuvering Valves and Throttle Valve.

The stop valve and the maneuvering valves are shown in Fig. 221. It will be noted that these maneuvering valves are both operated from a single handwheel through an oscillating beam gear so that only one valve can open at a time. The valves themselves are of unusual construction, being of the double seated "night-cap" type.

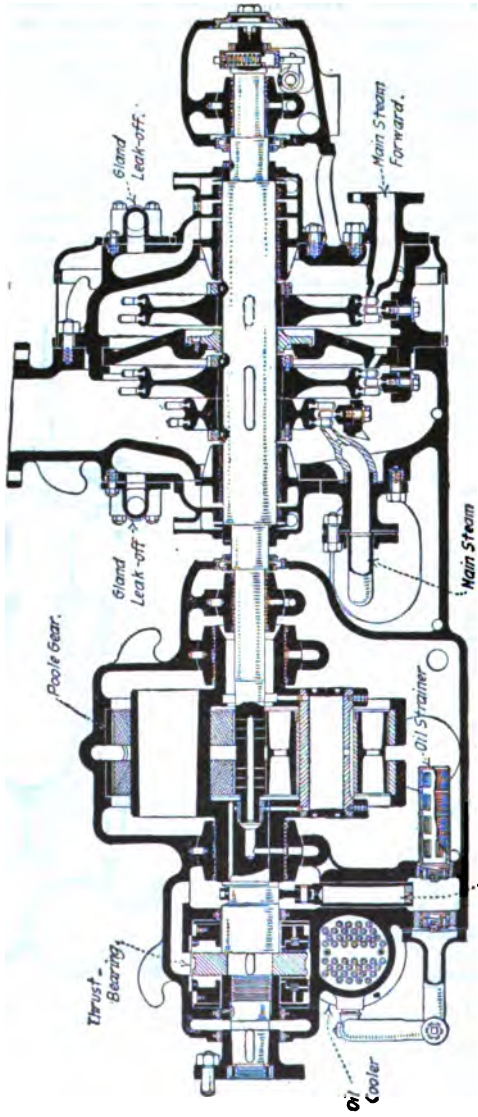


FIG. 222.—Poole Marine Geared Turbine 100 S.H.P.

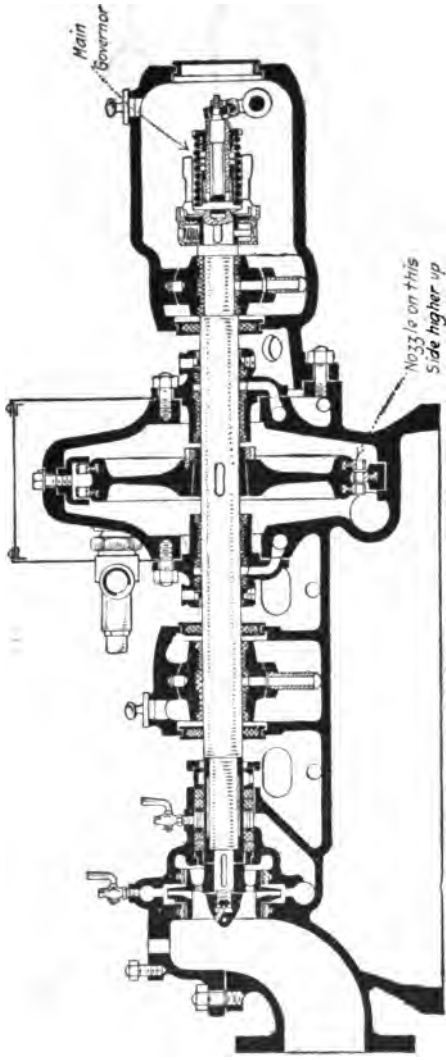


FIG. 223.—Poole Impulse Turbine Direct Connected to Centrifugal Pump.

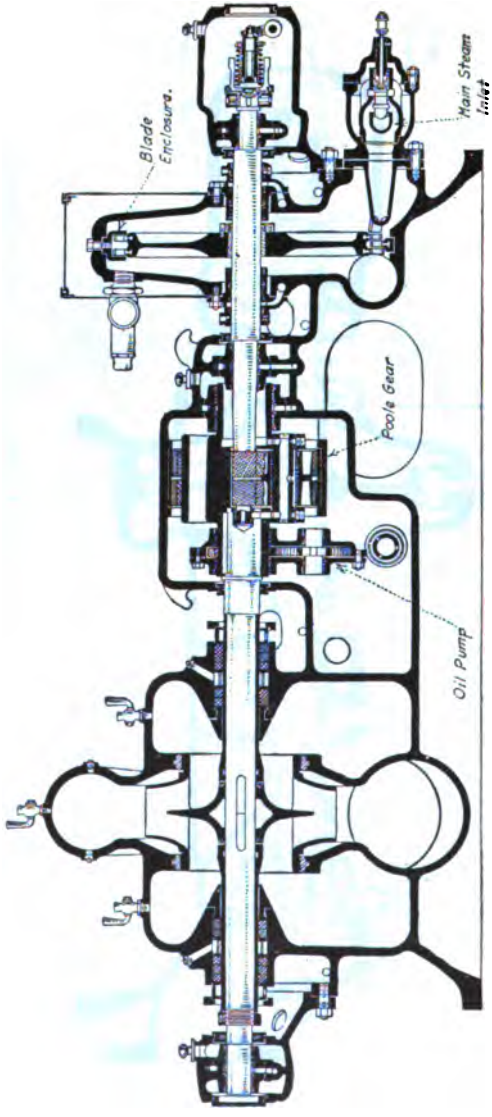


FIG. 224.—Poole Single Stage Turbine Geared to Centrifugal Circulating Pump.

The coupling is designed to give freedom in an axial direction only. One half of the coupling carries hardened pins pressed in place and secured by a set screw. The other end of the pin projects into loosely fitted brass bushings which enter holes in the other half coupling. These bushings are well lubricated by oil from the adjacent bearing, as can be seen from Fig. 220.

Lubricating oil for turbines and gears is supplied by either of two vertical turbine-driven oil pumps. These consist of a simple impulse wheel mounted on the upper end of a vertical shaft, while the lower end carriers a centrifugal pump impeller placed in a casing near the bottom of the oil drain tank. The oil flows through strainers on its way to the pump and is delivered at 20 lbs. pressure through an oil cooler in the drain tank to the oiling system. There is practically no oil pressure at the bearings.

The turning gear is placed on the forward end of the thrust bearing. A quill carrying a worm wheel is normally held from engaging a clutch on the main shaft by a spring placed under a false cover on the end of the outer pedestal. This cover is also bolted to the end of the quill. When the cover is removed, the quill can be forced into the shaft clutch and at the same time the worm wheel engages a worm in the pedestal body. The shaft can then be turned by a crank on the worm shaft.

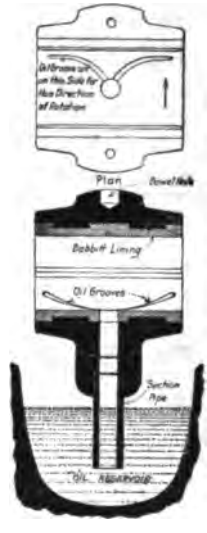


FIG. 225.—Poole Bearing for Small Turbines.

Small Turbines

Fig. 222 shows a 100 S.H.P. marine unit also built by this company. It consists of two velocity stages on the forward element and one on the astern. It operates at 8000 R.P.M. This speed is reduced to 800 R.P.M. on the propeller shaft. The oil pump is of particular interest. A reciprocating plunger is driven from an eccentric on the slow speed shaft. This pump delivers oil through a cooler on its way to the bearings and gears. The thrust bearing is placed on the slow speed shaft. The pinion of the reduction gears is integral with the turbine shaft and serves to hold it central.

A number of small turbines with single Curtis wheels have been built for auxiliary pump service. Fig. 223 shows one **direct connected** to a condensate pump. This unit is very compact. The casing is split in halves horizontally and contains both pump and turbine. A governor on the turbine shaft throttles the entering steam.

Fig. 224 shows one of the small units **geared** to a centrifugal pump for circulating condenser cooling water.

An interesting detail on these small units is the bearing oiling device. This is shown in detail in Fig. 225. A short piece of pipe dipping into the oil reservoir is screwed into the bottom of the oil supply hole in the spherical seat of the bearing. A hole is drilled through the bearing to register with this hole. Two shallow grooves are cut in the surface of the bearing from this central groove sloping in the general direction of rotation toward the ends of the bearing. The rotating shaft acts as a centrifugal pump and sucks the oil up this pipe, keeping the bearing flooded.

DIRECT DRIVE MARINE PROPELLING TURBINES

Newport News Shipbuilding and Dry Dock Co. Installation

A large twin screw direct driven turbine installation is shown in Fig. 226. Each main turbine is a complete unit of about 12,500 S.H.P. and at full power drives the shaft at 625 R.P.M.

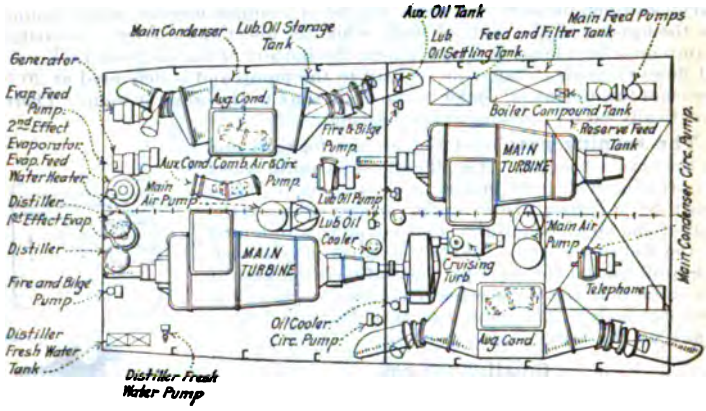


FIG. 226.—Plan of Newport News Direct Drive.

Main Turbine

The main ahead turbine, Figs. 227 and 228, consists of two three-row, velocity-compounded impulse stages followed by forty-three simple impulse stages mounted on a drum, which also carries the second stage. The first and second stages are separated by a cast iron diaphragm fitted with labyrinth packing. The forward drum disc is open and an extension carries labyrinth packing, which prevents steam from the second stage passing into the interior of the drum. Holes through the drum after the seventeenth stage allow the steam pressure at that point to act against the middle disk, which is solid, and overbalance the propeller thrust. The **casing** is of cast iron in four sections, split horizontally. **The rotor shaft** is hollow and carries the first stage ahead wheel and the ahead and the astern drums. The **wheels and drums** are cast steel. The light stiff construction gives a small deflection and high critical speed. A double faced single collar Kingsbury **thrust bearing** absorbs the difference between the propeller and the steam thrusts. **The glands** which prevent leakage where the shaft passes through the casing are of the labyrinth type, consisting of thirty rows of packing strips with four packing rings at the outer end. The packing strips for the ahead end are in three groups of ten rows each and for the astern end in two groups, one of ten and the other of twenty rows. The spaces between the groups are connected to receive steam from the gland steam system or to relieve steam to a lower pressure, as conditions may require. **The brass inlet nozzles** which direct the steam against the blades of the first stage in both ahead and astern turbines are cast in multiple and carefully machined on the inside. The ahead nozzles are divided into groups,

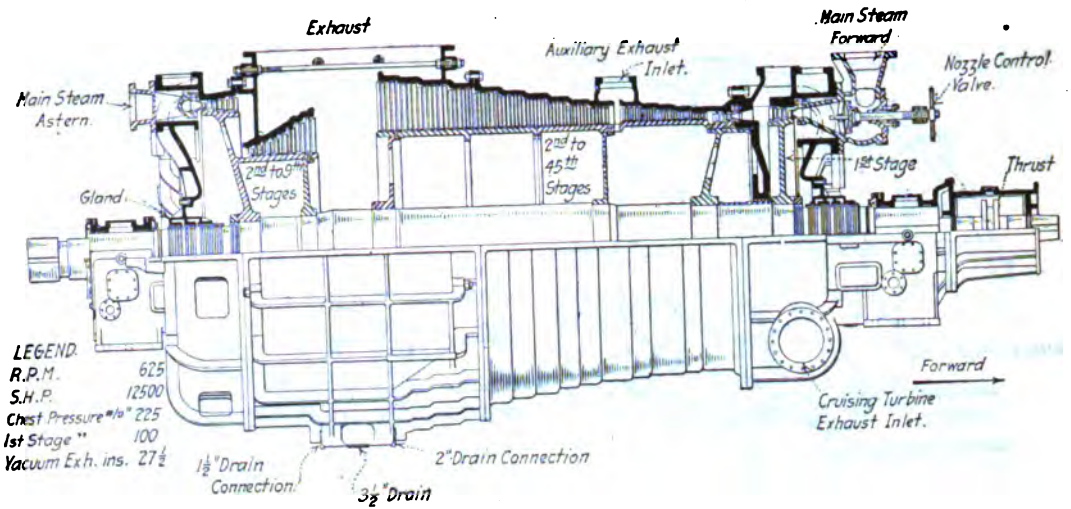


FIG. 227.—Newport News Main Turbine, Direct Drive.

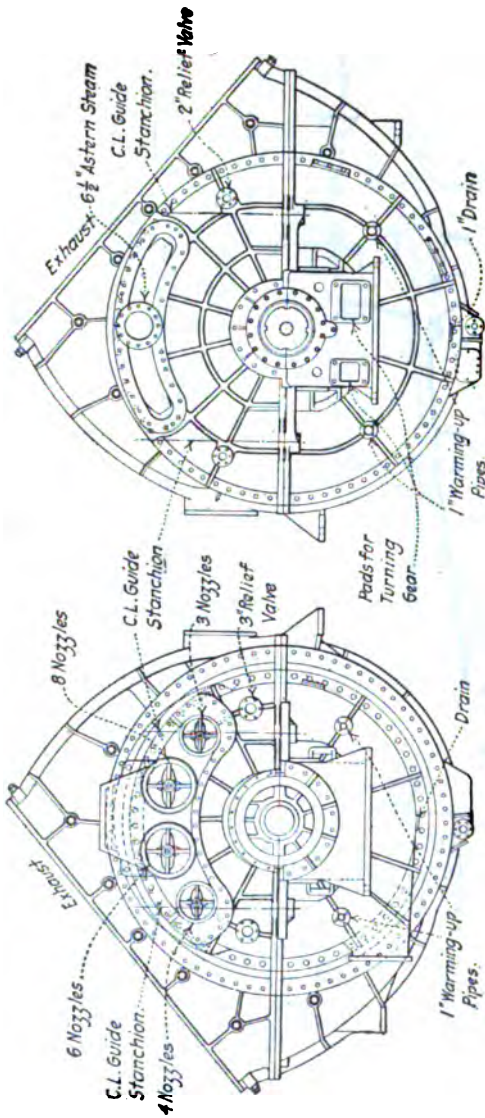


FIG. 228.—End Views Newport News Main Turbine.

each of which is controlled by a hand-operated valve fitted in the **steam chest**, which is of cast steel bolted to the turbine casing. The number of nozzles in use is varied to suit the speed required, finer adjustments being made by the main throttle valve. **The blading** is of cold drawn brass stock and the lower end of each blade is held in a dovetail groove. Packing pieces are inserted between the blades. The groove is closed by a calking piece. Shroud bands of channel section are used on the drum stages and flat bands on all other stages The bands are in short sections and are secured to the blades by rolling over a tenon on each blade projecting through a hole in the band.

Exhaust steam from the auxiliaries not required in the feed heaters is admitted to either the second or the seventeenth stage of the main turbines, as the pressure may permit.

The reversing turbines consist of one four-row, velocity-compounded impulse stage followed by eight simple impulse drum stages in the after end of each main turbine casing. Each unit is independent of the other and exhausts directly to the condenser. About 25 per cent of the full power ahead is obtained with two-thirds of the ahead steam flow.

Cruising Turbine

The cruising turbine, Figs. 229 and 230, for use to half speed, consists of one two-row and five single-row impulse stages and is connected to the forward

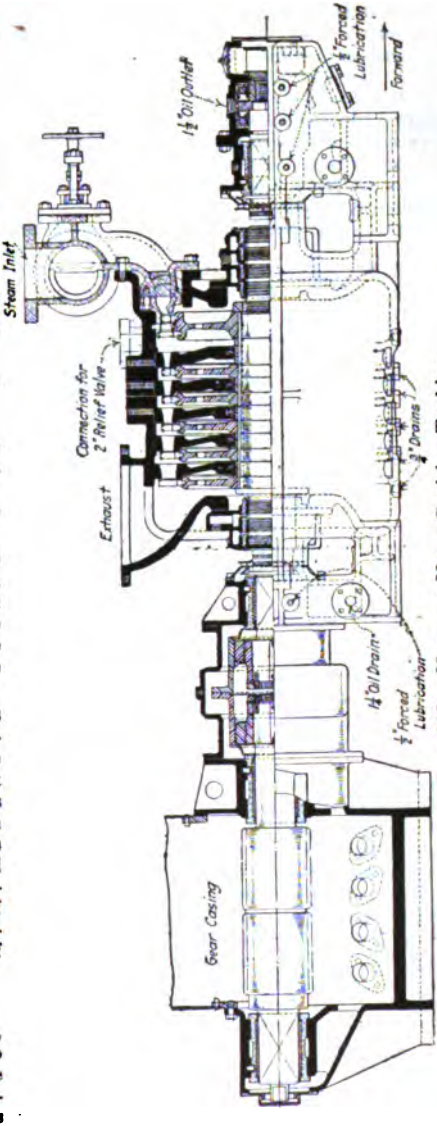


FIG. 229.—Newport News Cruising Turbine.

end of the starboard main turbine shaft through a single reduction gear and a flexible coupling of the jaw type, which is lubricated by oil escaping from the after turbine bearing. The exhaust from the cruising turbine passes to the second stage of the port main turbine, the starboard main turbine running in a vacuum. The cruising unit develops 1800 S.H.P. at 3200 R.P.M., which is reduced to 320 R.P.M. on the main shaft. The **casing** is of cast iron, split horizontally, the top being in two parts. The **wheels** are of forged steel and the **diaphragms** of cast iron with serrated composition bushings at the shaft. A smaller **thrust bearing** of the same type as on the main turbines, is fitted to absorb the steam thrust. The **glands** are similar to those on the main turbines, except for the omission of the packing rings at the outer end on account of the high speed. The

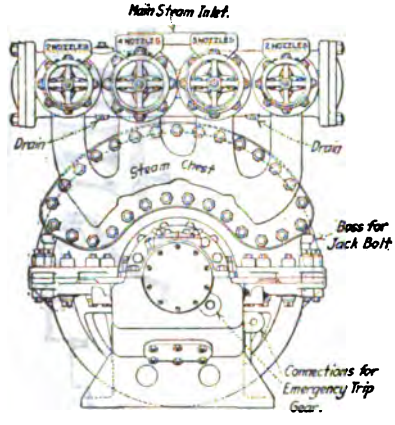


FIG. 230.—End View Newport News Cruising Turbine

first stage nozzles are also similar to those on the main turbines. For stages after the first, the nozzles are formed of nickel steel plates cast in the diaphragms. The **blades** are drop-forged manganese bronze with integral packing pieces, and are held in dovetail grooves. An emergency overspeed **governor** on the forward end of the shaft trips and closes the throttle valve when the speed exceeds about 3600 R.P.M.

The cruising turbine reduction gear is of the rigid frame double helical type. The wheel is of cast steel, divided vertically into halves, which are bolted together. The nickel steel pinion is also in two parts, the teeth of one helix being cut in the shaft, and of the other helix in a separate sleeve shrunk on the shaft after the teeth are cut. The housing is of light construction, the lower half being of cast iron and the upper half of sheet steel.

THE LJUNGSTROM TURBINE

The Ljungstrom Turbine is of the **radial flow reaction type.** It has received considerable attention in Europe and has recently been taken up in America. The following description of this turbine is largely abstracted from articles in *Engineering*.

The Ljungstrom turbine differs materially from the usual type of axial flow reaction turbine. **The aim of its design** is to increase the relative speed of the reaction blading and thus increase the ratio of blade speed to steam speed. The efficiency of reaction blading increases until this ratio of blade speed to steam speed reaches about 0.9. This increase in speed is secured by radial construction and by revolving each disc with its blades in the opposite direction to the other disc. The relative velocity of the blades to one another is thus twice the relative velocity were one disc to be held still.

The Ljungstrom turbine hence **consists fundamentally of two series of concentric blade rings,** each series of which is connected to its respective supporting disc and the two series rotating in opposite directions. Each blade ring is connected to its disc through a bulb-headed expansion ring. The other side of the blade ring is free. Steam enters the turbine at the centre and flows radially outward between the two series of blade rings which consequently rotate the supporting discs in opposite directions. Each disc is connected to an alternator rotor. The two generators are tied together electrically and operate as one unit. Thus each disc in a 7000 kw set carries 19 blade rings, the largest of which is 33½ in. in diameter and the smallest 5¾ in.

The turbine is very light in weight and is compact. As the ratio of blade speed to steam speed is high, **its efficiency** is unusually good. The design also lends itself to the use of high superheat without danger of distortion. The use of high temperature and moderate steam speeds results in little difficulty with erosion of blading.

A cross-section of the upper half of one of the 625 kw turbines for the S. S. Wulsty Castle, described on p. 704, is shown in Fig. 231. This turbine was built to operate at 220 lbs. gage pressure, 625 deg. F. steam temperature at the throttle valve, and runs at 3600 R.P.M. The moving parts of the turbine are 28¾ in. in diameter and 17½ in. long. Its weight is about 450 lbs. complete.

There are 39 rings in all, which are divided into a H.P., an intermediate and a L.P. section. The innermost ring has an outside diameter of 3 1/16 in. while the outer diameter of the last ring is 28⅜ in.

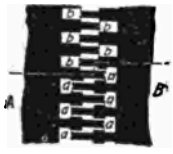
Each disc is built up of three sections fastened together by bulb-headed rings. This construction permits each section to expand freely without affecting the other sections and at the same time maintains alignment. This freedom of movement is necessary on account of the wide range of temperature from inside to outside circumference of the disc. The distortion in a solid disc would be so serious that large clearances would be necessary.

Since the steam that leaks past the dummies is necessarily highly superheated, each of the moving **dummy plates** (c) is connected to the turbine disc by two bulb-headed rings only. If this dummy plate were made a part of the turbine disc, the latter might be seriously distorted by the high temperature leakage steam on one side and the low temperature working steam on the other.

The blades are machined out of nickel-steel bars and have projections machined on each end. These projections fit holes in the assembly rings and maintain the blade at the proper angle. After assembly is complete,

the ends are welded over by an oxy-acetylene flame. The assembly rings are then turned to form and dovetailed at the back. This dovetail is secured by rolling into a corresponding slot formed in a stout steel blade carrier ring. This arrangement can be seen in Fig. 231. The **carrier rings** are held to the discs by bulb-shaped rings.

Fine **clearances** obtain throughout. Thin **fin dummy strips** of nickel steel are calked in grooves on each carrier ring to keep this clearance at a minimum. If a rub does occur, the thin fin wears away without enough heating to cause distortion. Leakage by axial flow of the steam from between the blade rings has in all cases to get past two of these dummy strips. Such leakage loss is said to be small.



The L.P. rings are built up in three sections in order to secure strength and stiffness, together with the proper steam flow area.

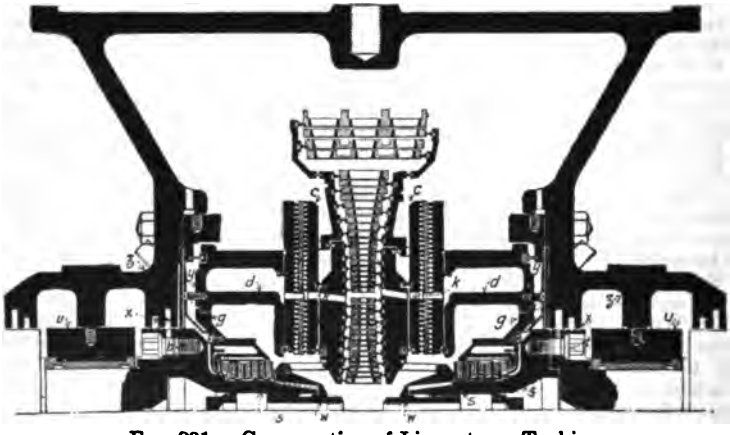


FIG. 231.—Cross-section of Ljungstrom Turbine.

The pressure tending to force the discs apart is balanced by the **dummies** (*c*)(*c*). Each consists of a pair of grooved discs, one of which is mounted on the turbine disc and the other on the casing. A section shows the detail construction. Each projecting tooth has a thin **baffle plate** of nickel steel fastened to its outer end, forming more fin packing. Two forms of baffling are used as shown, to automatically balance the thrust separating the discs. There is a tendency for the blade discs to force themselves apart or to draw together and this may cause axial hunting motion on the spindle. Such hunting is prevented by an **oil dash pot** on the outer end of each shaft.

Steam enters the turbine through a forked steam pipe to both sides of the blade discs. It enters the forged steel steam chest (*d*) (*d*). These support the gland bushing at one end and the stationary half of the dummy at the other. Steam usually passes into the inner ring of the steam chest. In case of overload, steam is admitted to the outer ring.

The discs carrying the blade rings are mounted on conical seats on the end of the short stub end of the hollow **shafting**. The stub shaft is bolted

firmly to the generator shaft and centered by a tongue and groove. When it is desired to dismantle the turbine, the top of the casing is removed and the top halves of the bearings are lifted off. The bolts holding the stub shaft to the generator shaft are then removed. The two turbine elements are firmly clamped together in a special fixture which holds the blades and the dummies in the true alignment. The generator shafts are then withdrawn until the end clears the male tongue of the stub shaft when the whole rotor can be lifted out.

Fig. 232 shows a cross-section of the turbine and its two generators.

The **throttle valve** (*A*) is combined with the **governing valve** *B*, as shown in Fig. 233. The valve (*B*) is so designed that it does not open appreciably until (*A*) has lifted some distance from its seat. Governing is done by varying the oil pressure under the piston (*C*). With increasing speed this oil pressure decreases and the throttle valve is closed by the heavy spring above (*C*). The **emergency governor** acts on lever (*F*) and allows valve (*E*) to equalize the oil pressure over the two sides of (*C*). The spring then closes the valve. The valve stem and hand wheel (*D*) is not fastened to the regulating valve and is merely used to close the throttle valve firmly when the turbine is shut down. On starting up steam can only be admitted to the turbine by establishing the oil pressure either by a hand pump or by the auxiliary pump. This ensures ample oil on the bearings before starting up. A **strainer** (*G*) is fitted below the throttle

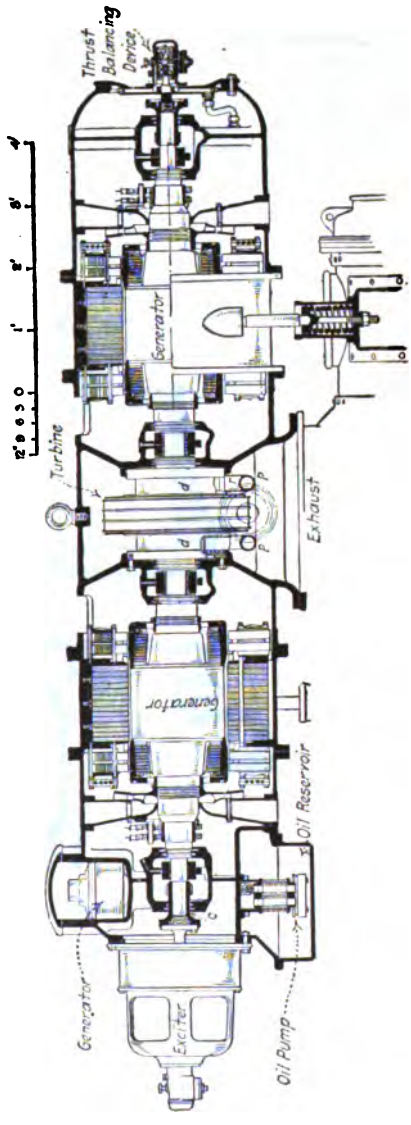


FIG. 232.—Arrangement of Ljungstrom Turbine with Generators.

valve (*A*). Fig. 234 shows the steam connection from the governing valve gears to the center of the turbine.

The **governor gear** is shown in Fig. 235. The governor is driven from the main shaft by spiral gears. A gear type oil pump is connected to the bottom of the governor spindle. The governor actuates through the rod *J*, the valve controlling the supply of oil to the governor valve. The compensating lever *I* is on the same rod as the lever *H*, Fig. 233, and through the rod *L* and levers control the compensating sleeve on the governor oil

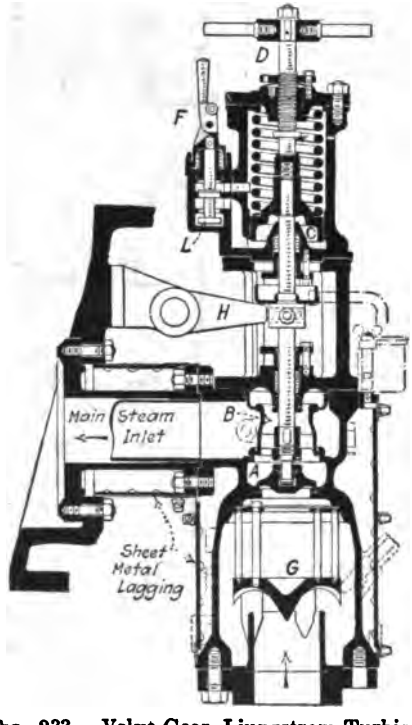


FIG. 233.—Valve Gear, Ljungstrom Turbine.

valve control. The governor regulates the amount and pressure of oil going to the governor valve and thus the lift of this control valve.

The Auxiliary oil pump is fitted, as shown in Fig. 235. This consists of a vertical centrifugal pump driven by a simple impulse turbine.

The Ljungstrom turbine is said to give very high economy. It is particularly adapted to high superheat and also to high vacuum. Special designs have been built for radial flow where one element is held stationary. In this case only one generator is used.

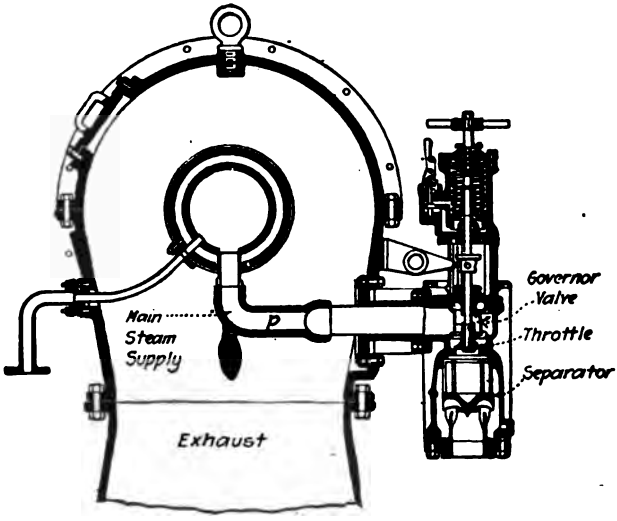


FIG. 234.—Governing Valve Gear and Steam Connection to Ljungstrom Turbine.

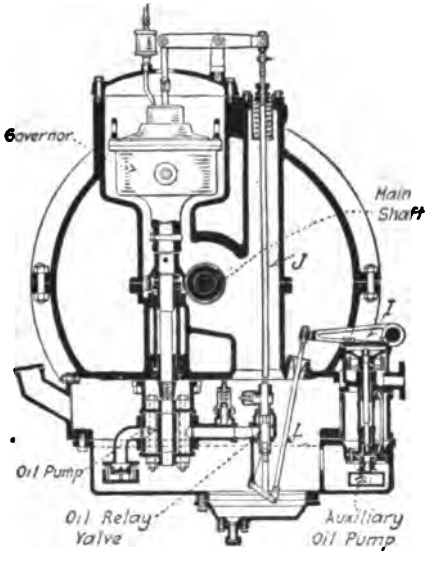


FIG. 235.—Ljungstrom Governor Set.

STURTEVANT TURBINES

B. F. Sturtevant Co., Boston, Mass. supply a large number of small turbines for auxiliary drive on shipboard. Fig. 239 shows their standard design of turbine.

FIG. 236.—Sturtevant Rotor.

The rotor is a solid forging of best open hearth steel keyed on the shaft. The buckets are semicircular in form and are milled into the rim at an angle with a tangent to the circumference, as shown in Fig. 236.

The steam is expanded in a Tobin bronze **nozzle** and discharged into one side of the **bucket**. In crossing a bucket the direction of the steam jet is reversed and on leaving thr bucket it enters a semicircular reversing chambee which is cast in one piece with the nozzle. This

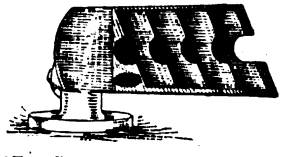


FIG. 237.—Sturtevant Nozzle and Reversing Buckets.

chamber redirects the steam back onto the buckets again. This reversing action is repeated several times. The **nozzle block** with the four reversing

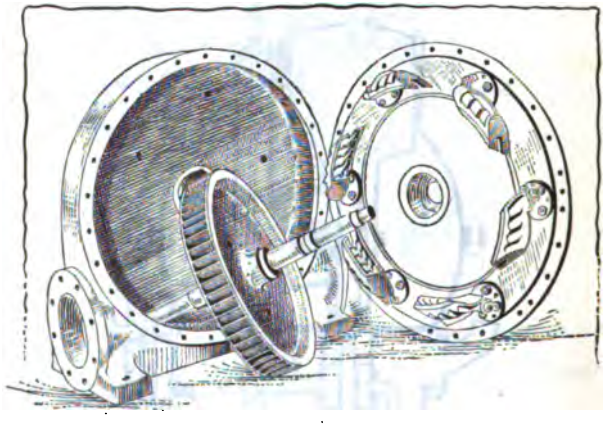


FIG. 238.—Sturtevant Turbine Parts.

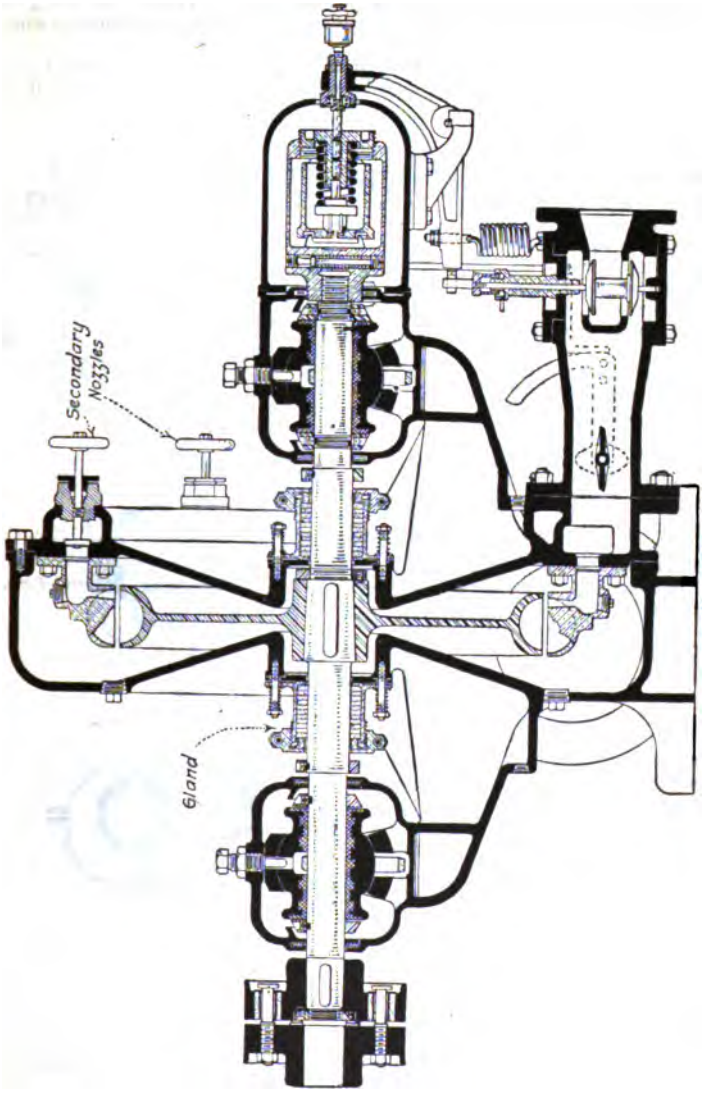


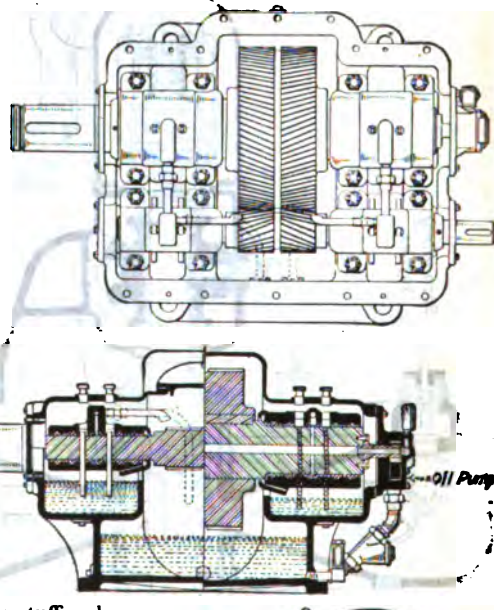
FIG. 239.—Horizontal Sturtevant Turbine.

shown in Fig. 238. Each nozzle can be shut off from outside the casing by a hand valve. The load can thus be adjusted to meet steam conditions when the demand for power varies widely.

Spherically seated, ring oiled **bearings** are furnished. These are of cast iron, babbitt lined. Special oil guards with felt pads prevent leakage of oil from the casings and admission of dust into them.

The shaft is turned and ground from a 3½ per cent nickel steel forging. It can be adjusted axially by threaded steel collars on either side of the bearing at the governor end.

A floating type of **stuffing box** filled with special packing is employed on these units. The faces of casing and stuffing-box are ground to make a steam-tight joint. The stuffing box is held against the cylinder by springs placed behind the nuts on special studs. This permits some lateral play of these stuffing boxes.



The turbine is provided with a **main governor** of centrifugal type mounted on one end of the shaft. An **emergency overspeed governor** of the usual unbalanced weighted spring type is fitted in the shaft itself and controls an emergency butterfly valve that shuts off the steam supply. The main governor controls a balanced regulating valve.

The casing is made of two castings split vertically. The bearing brackets are cast integral with the casing ends.

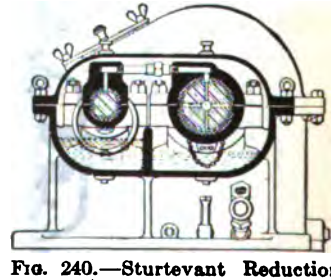


FIG. 240.—Sturtevant Reduction Gears

These turbines are used for many services such as for driving centrifugal pumps, fans, blowers, etc., through reduction gears. One of these gears is shown in Fig. 240. The gears are oiled by means of a small gear oil pump on the end of the slow speed shaft.

An application of the Sturtevant turbine for driving a vertical blower for marine service is shown in Fig. 241. A ball bearing is provided on the vertical

shaft to carry the weight of the revolving parts. A small gear type oil pump at the lower end of the spindle furnishes oil to the various bearings. The oil returns to the oil reservoir at the bottom of the set after passing through the

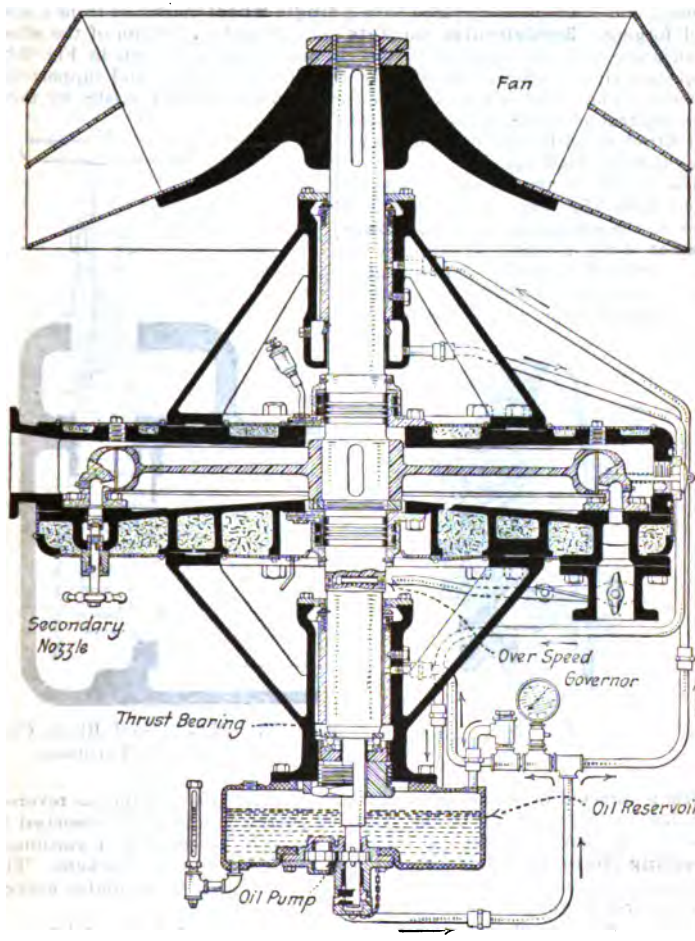


FIG. 241.—Vertical Sturtevant Blower Set.

bearings. The glands on this unit consist of a form of labyrinth packing with piston rings. If the machine stands idle at times, provision has to be made to supply grease to these packing glands. A very complete report of a series of tests on this fan equipment is found in the paper "Test of Sturtevant Forced Draft Blower," by M. C. Stuart, *Jour., A.S.N.E.*, Aug., 1918.

TERRY STEAM TURBINES

The Terry Steam Turbine Co., Hartford, Conn., build small turbines for auxiliary service and larger turbines, principally for turbo-generator sets.

Small non-condensing turbines have a **single wheel** machined from a solid steel forging. **Semicircular buckets** are cut in the solid rim of the wheel at an angle with the tangent to the circumference, as shown in Fig. 242. Projecting rims are left at the sides of the wheel to protect and support the buckets. The wheel is keyed to the forged shaft and held axially by nuts. This construction of wheel and buckets is well fitted to withstand high temperatures and high blade speeds.

The action of the steam can also be noted from Fig. 242. It is expanded down to the exhaust pressure in **bronze nozzles** of the divergent type which are so constructed that they can be easily replaced if steam operating conditions are changed. The jet issuing from the nozzle is directed into one side of the bucket and its direction is reversed 180 deg. Only a portion of the available energy in the jet is absorbed by this passage across the bucket. The jet is again reversed in a stationary **reversing chamber** and is directed a second time into the buckets. This action is repeated a number of times until the jet loses its available energy. This reversing process can be easily seen in Fig. 242.

FIG. 242

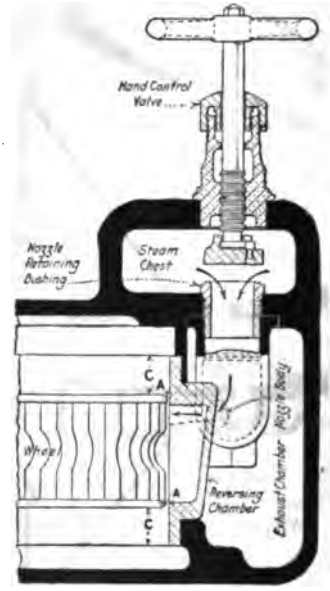


FIG. 243.—Nozzle and Blade Construction Terry Turbines.

Each nozzle has its own accompanying reversing chamber and may be closed off from the steam chest by a hand valve. The steam chest forms a part of the casing and is the only portion subject to high pressure. The details of this construction of nozzle, closing-off valve and reversing chamber are shown in Fig. 243. The radial clearance (A) between reversing chamber and wheel varies from $\frac{1}{32}$ to $\frac{3}{64}$ in. according to the wheel size. Side clearance of the wheel (C) is about 1 in.

Since each nozzle is proportioned to carry a portion of the total load, partial loads may be obtained at high economy by using only the required number of nozzles to carry the load.

The **casing** is split horizontally. The **bearing housings** are cast integral with the lower half of the cylinder. All steam and exhaust connections are placed on the lower half to facilitate dismantling. Fig. 244 shows a cross-section of a standard small turbine.

The **bearings** in the larger sizes are of cast iron babbitt lined. In the smaller sizes they are of bronze. All are ring oiled.

A **direct acting governor** of the fly-ball type is mounted directly on the end of the turbine shaft. It actuates a balanced double-seated governor valve through a simple lever, as shown in Fig. 244. An emergency governor is also provided which closes a butterfly type of valve on the main steam supply in case of overspeed.

The **glands** are of the regular carbon ring type though occasionally labyrinth packing is used. The carbon rings are made in three segments held together by a circumferential garter spring on the outside. They are placed in a separate gland housing made either of bronze or cast iron, which can be easily removed for inspection or repairs.

Fig. 245 shows a vertical type of unit frequently used on shipboard for blower drive. The casing, bearings and bearing housings are split vertically and the casing is hung on hinges, so that all moving

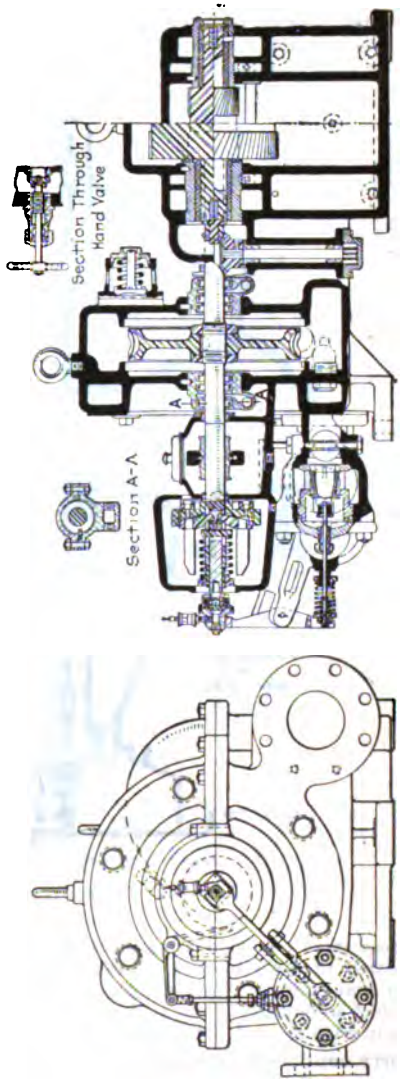


FIG. 244.—Terry Geared Turbine.

parts can be inspected without removal of the piping or of the turbine from its seating. Lubrication in this type is furnished under a slight pressure by a **gear type oil pump** driven from the lower end of the main shaft. A ball step bearing supports the weight of the rotating parts and operates in a continuous flood of oil. Down drive vertical turbines are also built.

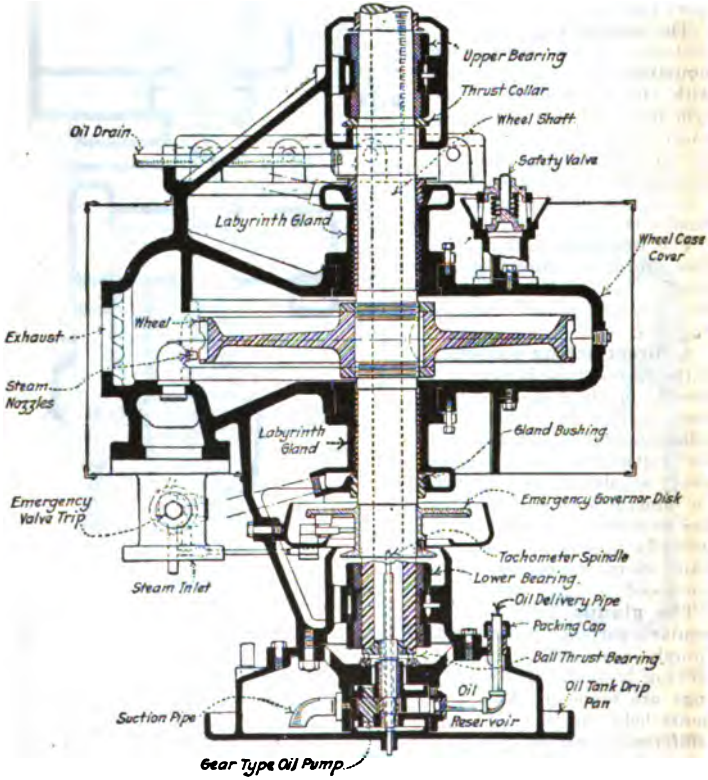


FIG. 245.—Terry Vertical Turbine.

These small non-condensing units have been used either direct-connected or geared to drive blowers, exciters, condenser auxiliaries and centrifugal boiler feed pumps and other pumps. They have also been geared to rotary type fuel oil pumps. Some such units have been geared to pulleys and used for belt drive on land. Fig. 246 shows a centrifugal pump set.

Terry condensing turbines for marine service are furnished in 100 H.P. size and up. They are built on the "return-flow" principle shown in Fig. 247. They consist of a single Curtis stage with two rows of revolving blades. The

steam then is carried to the opposite end of the casing through a port cast in the upper half of this cylinder. It flows in the opposite direction through a series of Rateau stages which expand the steam to the exhaust pressure. This construction produces a very compact unit and removes all chance of air leakage into the condenser with heavy loads. It also permits a very simple form of gland to be used. Correct alignment is secured by supporting the casing near the centre line and by having both ends of the casing at the same temperature.

The nozzles are fastened to the casing by means of bronze bushings which form the seats of the hand control valves. The orifices in the Rateau section are formed by steel plates cast in the diaphragms.

The Curtis element consists of two discs mounted on a forged steel hub and screw riveted together. The Rateau wheels are made in a similar manner, as the H.P. discs. The shaft is stepped to receive these wheels.

The Shaft is a steel forging large in diameter and short between supports to secure a critical speed well above the operating speed. The stiff shaft ensures long life for the glands and diaphragm packings. The wheels are held to the shaft by keys and by a lock nut, as shown.

The diaphragm packings are made of bronze and are of the labyrinth type as are also the casing glands.

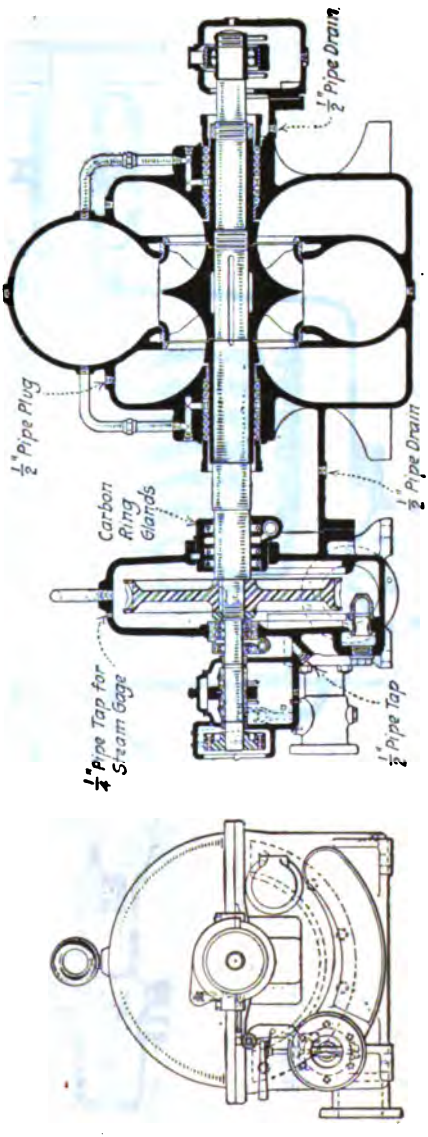


FIG. 246.—Terry Centrifugal Pump Set.

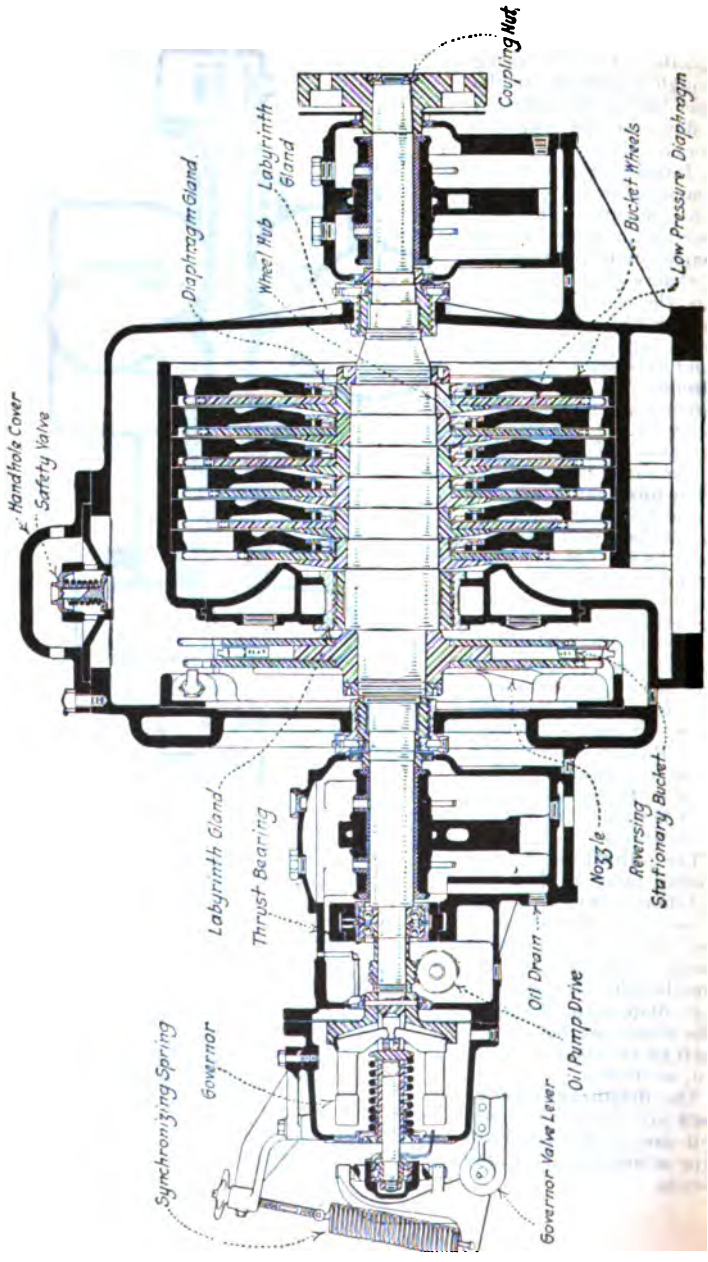


FIG. 247.—Terry "Return Flow" Turbine.

The **blading** is made of drawn bronze. It is inserted in slots in the wheel, as shown in Fig. 248, and is locked by a special method. A shroud is provided over the end of the blades to give added rigidity and to prevent vibration.

The **cylinder** is split horizontally and steam and exhaust connections are placed in the lower half. The diaphragms, glands and bearings can be removed without lifting the rotor. The diaphragm between the h. p. and the exhaust is cast hollow with an ample air pocket to prevent conduction of heat to the exhaust. A long labyrinth packing gland is provided in this diaphragm.

A gear type **oil pump** driven by spiral gears from the main shaft furnishes oil under a pressure of 25 to 40 lbs. for the oil relay. Oil rings for emergency use are also supplied on the bearings. An interesting construction is the cast water jacket under the bearing casing for cooling the oil.

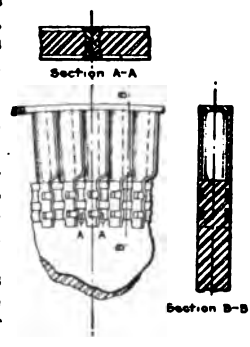


FIG. 248.—Terry "Return-flow" Turbine Blading.

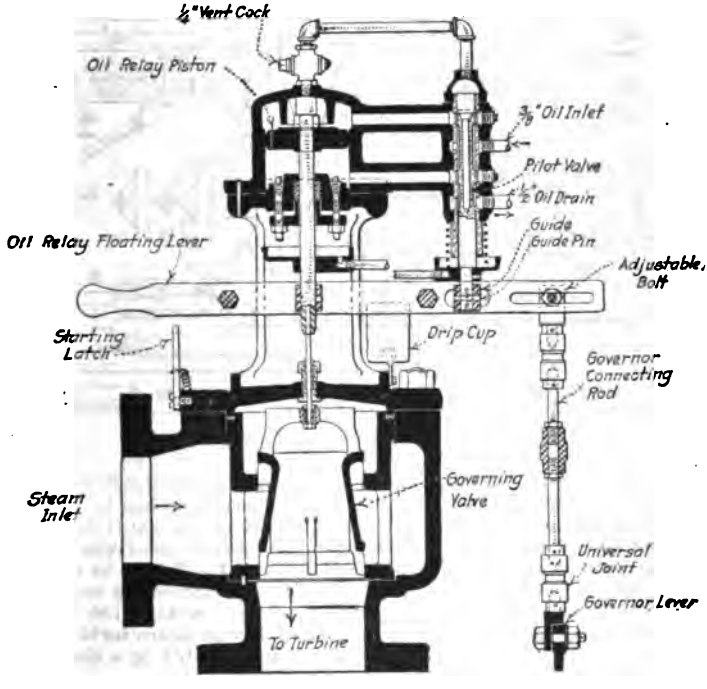


FIG. 249.—Terry Oil Relay Governing Valve.

The **main governor** is the same as on the smaller machines. On small turbines the same type of governor valve is used. Larger turbines use an oil relay governor valve, as shown in Fig. 249. This involves the usual relay construction with compensating lever and connections.

An **overspeed emergency governor** is also provided together with a vacuum breaking device which destroys the vacuum when the main steam to the turbine is shut off.

A ball bearing thrust is furnished to care for any slight end thrust.

The glands may be either of the carbon or labyrinth type. A steam seal is provided to prevent the entrance of air.

This type of turbine when used to drive turbo-alternators is direct connected. When furnished with direct connected generators, helical reduction gearing is used. Large single-stage and multi-stage centrifugal pumps are also driven through helical gearing.

ENGINE ROOM LAYOUTS

The following diagrams and discussion illustrate some typical arrangements of turbine engine rooms on shipboard.

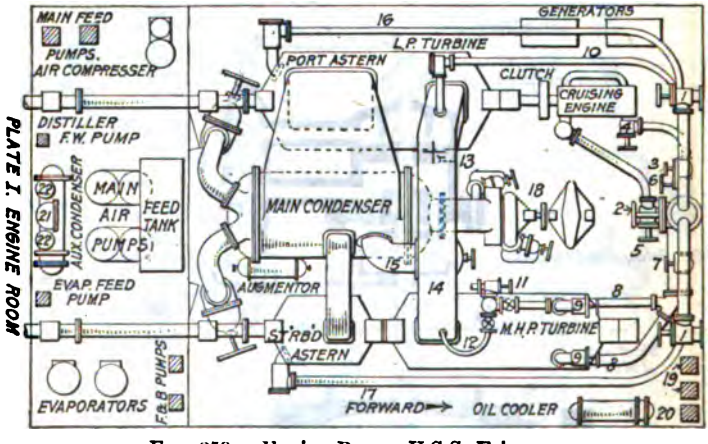


FIG. 250.—Engine Room, U.S.S. Ericsson.

Fig. 250 shows the arrangement of the engine room of U.S.S. Ericsson, a Torpedo Boat destroyer with direct connected Parsons turbines built by New York Shipbuilding Co., Camden, N. J. The Ericsson is 300 ft. long, 30 ft. beam and has a displacement of 1090 tons. Its contract speed was 29 knots with the turbines developing 17,000 S.H.P. at 600 R.P.M. As shown on the layout a vertical direct acting compound reciprocating engine with cylinder 16 in. and 24 in. with 18 in. stroke and with cranks at 180 degree. is arranged on the port shaft forward of the L.P. and astern turbine. This serves as a cruising engine only and may be disconnected by a clutch from the main L.P. turbine.

The condenser is placed directly in the centre of the boat. The single

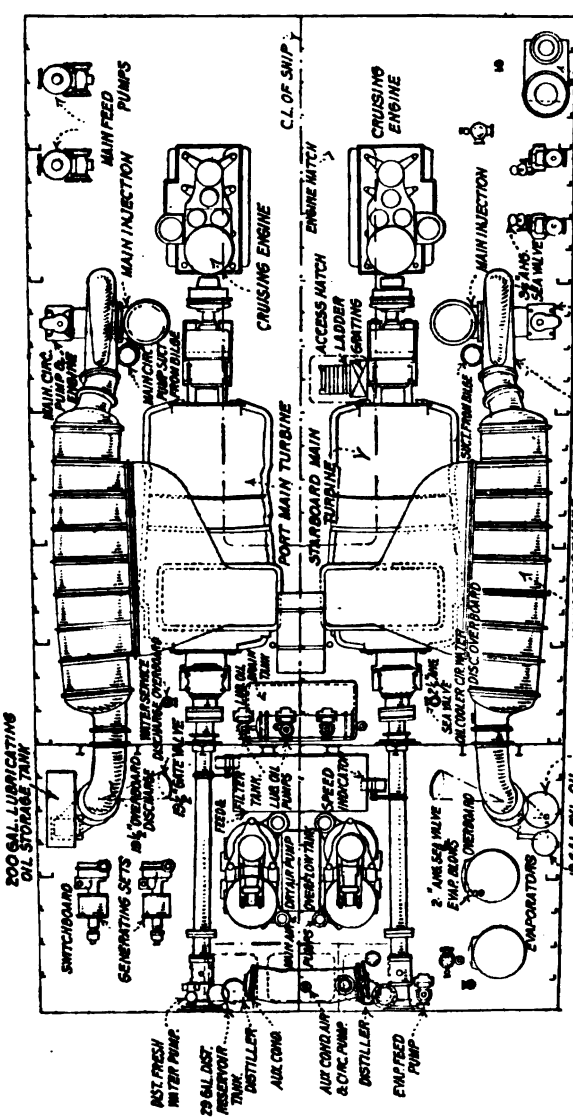


FIG. 251.—Engine Room U.S.T.B.D. Duncan.

condenser for two shaft layouts is not recommended. The circulating pump is of the centrifugal type, driven by a Terry Turbine.

Fig. 251 shows the engine room of the U. S. Torpedo Boat Destroyer Duncan. This ship is 300 ft. long, 30 ft. beam, with a displacement of 1013 tons. The propelling machinery consists of two sets of main turbines and compound reciprocating cruising engines direct connected to the screws which are used in series with the main turbines for cruising speed up to 15½ knots. Curtis marine type turbines of 7500 S.H.P. at 550 R.P.M. are installed. Steam pressure is 250 lbs. gage. The vertical engines are 12½ in. by 26¼ in. and 13½ in. stroke and are connected to the turbine shaft by jaw clutches. The engines can also be used for backing.

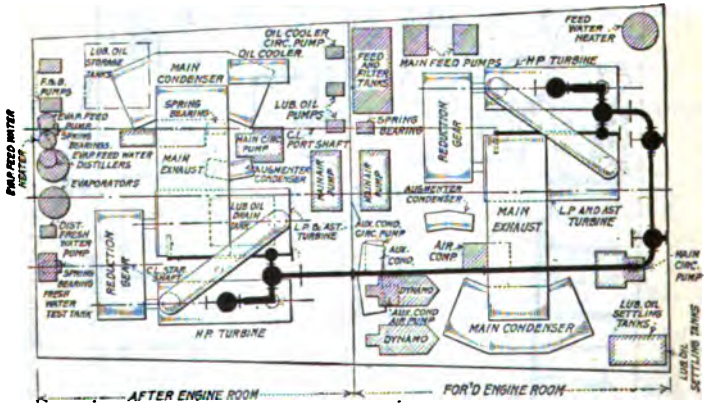


FIG. 252.—Engine Rooms U.S.S. Wadsworth.

Fig. 252 shows the arrangement of the engine room of U.S.S. Wadsworth. The Wadsworth is a torpedo boat destroyer 310 ft. long, 30 ft. 6 in. beam with a displacement of 1050 tons. The propelling machinery consists of Parsons geared turbines for a twin screw drive built by Bath Iron Works. Each shaft is driven through gearing from one ahead H.P. and one ahead L.P. turbine and an astern turbine built in the same casing as the ahead L.P. turbine. The installation is designed to drive the screws at 450 R.P.M. at 30 knots speed, developing 17,500 S.H.P. The H.P. turbines run 2494.5 R.P.M. and the L.P. turbines at 1509.8 R.P.M.

The reduction gears were made on Parsons designs and are of the single reduction type. No cruising turbines have been provided on this ship.

Fig. 253 shows the engine room arrangement of the "A" ships built by the American International Shipbuilding Corp., Hog Island, Pa., for the Emergency Fleet Corporation. These ships are 7500 ton dead weight cargo ships and are equipped with 2500 S.H.P. General Electric single cylinder marine turbines. This turbine operates with 200 lbs. gage pressure at the throttle 50 degrees F. superheat and 28.5 in. vacuum. Its full power speed is 3234 R.P.M. This is reduced through double reduction gearing to 90 R.P.M. at the propeller. The oiling system is fully described on p. 665.

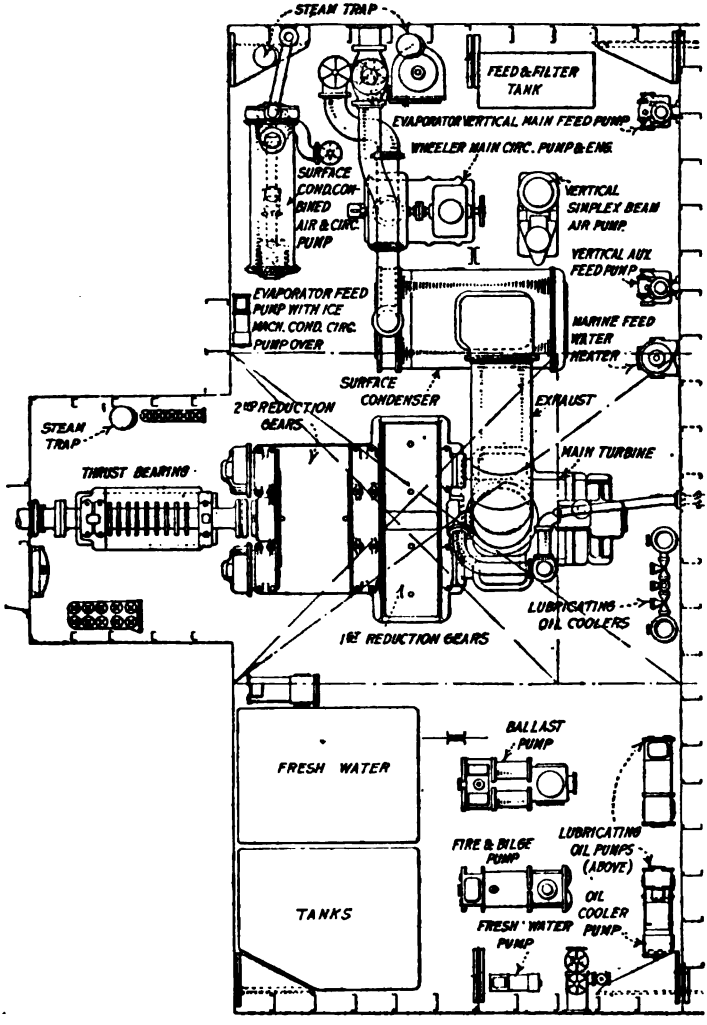


FIG. 253.—Plan of Engine Room "A" Ships American International Shipbuilding Co., Hog Island, Pa.

All operating levers, throttle valves, drip valves, etc., are controlled from one point at the turbine.

Fig. 254 shows a section of the same engine room and shows the relation of the condenser auxiliaries to the turbine and gears. The location of the 15-kw electric lighting sets and the refrigerating machine is also shown on the upper decks.

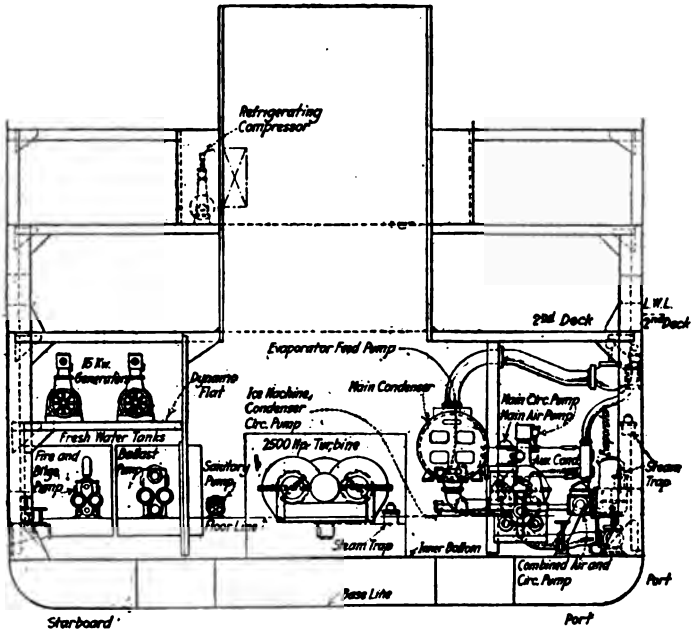
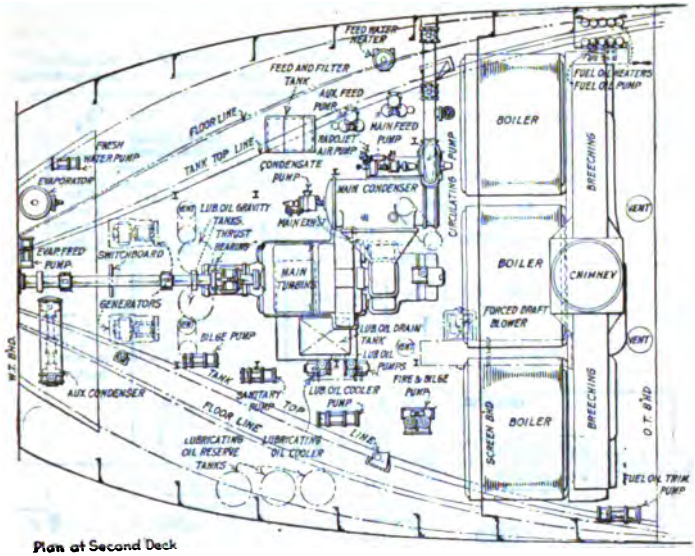


FIG. 254.—Cross-section of Engine Room Hog Island "A" Ships.

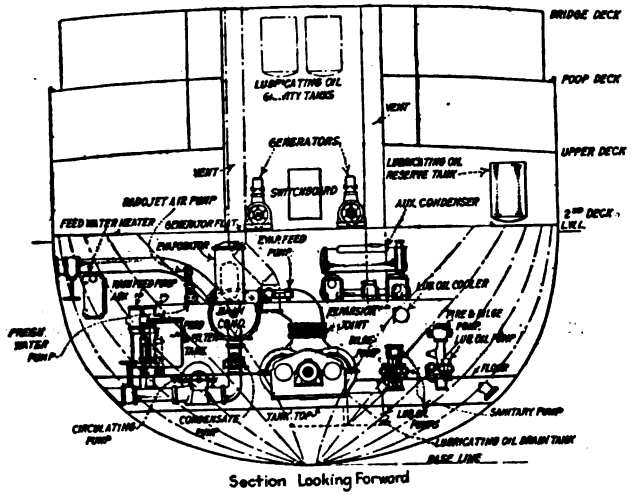
Fig. 255 shows the plan and elevation of the engine room of one of the 10,000 ton tankers built by the Baltimore Dry Dock & Shipbuilding Co. It contains a 3000 S.H.P. Westinghouse turbine geared through double reduction gears to a single propeller. This installation is placed well aft on the boat and the machinery had to be carefully arranged to prevent crowding.

Fig. 256 shows the plan of the "B" ships built by the American International Shipbuilding Co. Hog Island, Pa. These are combined cargo and passenger boats of 8000 tons dead weight capacity. The turbines consist of high and low pressure cylinders driving a single propeller shaft through double reduction gears. The turbines are designed to deliver 6000 S.H.P. They were furnished by the General Electric Co.

The engine room is shown in section in Fig. 257 where the relative arrangement of most of the machinery is shown. An unusual feature of this ship



Plan at Second Deck



Section Looking Forward

FIG. 255.—Engine Room, 10000 Ton Tankers. Baltimore Dry Docks and Shipbuilding Co., Baltimore, Md.

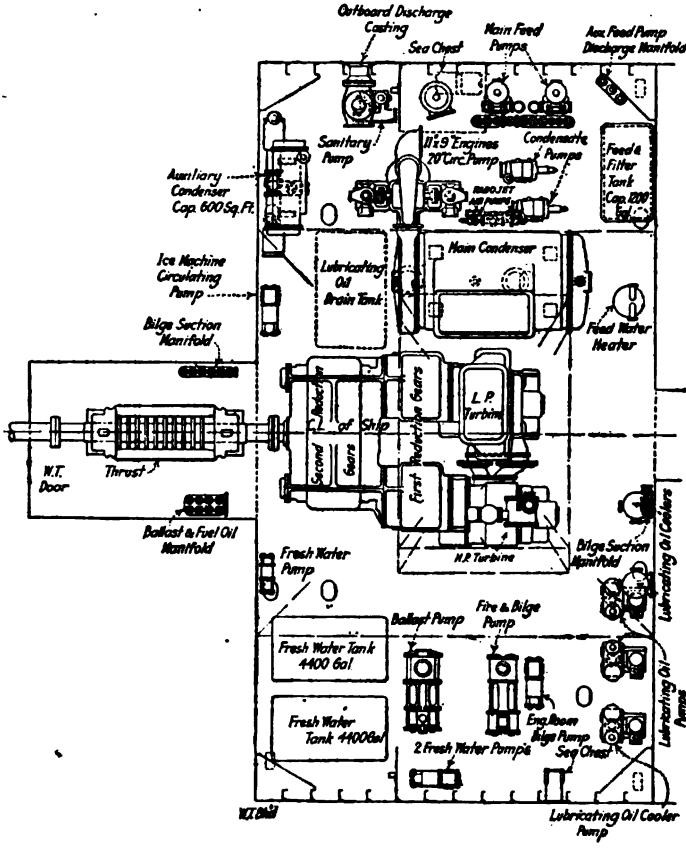


FIG. 256.—Plan of Engine Room "B" Ships, American International Shipbuilding Co., Hog Island, Pa.

is the installation of a travelling crane as part of lifting gear of this turbine and gear set.

The plan of the engine room of S. S. Wulsty Castle, using Ljungstrom turbines and the electric drive is shown in Fig. 177, p. 706. The electric drive has attracted so wide attention that a section of this engine room is shown in Fig. 258. This, together with the plan shown in Fig. 177, gives a very clear idea of how the equipment is arranged on this particular ship.

Figs. 259 and 260 show the engine room of S. S. Merida, a British vessel 435 ft. long, 55 ft. 3 in. beam and with a capacity of 9100 tons.

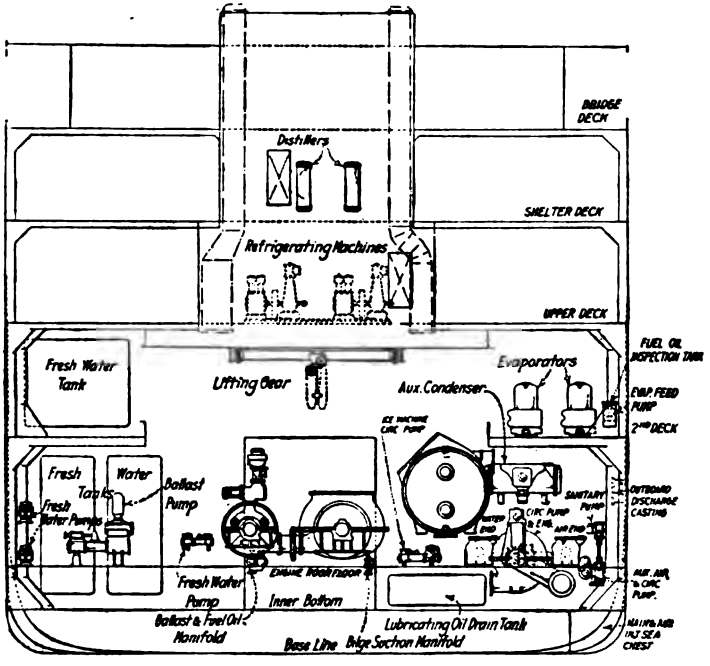


FIG. 257.—Cross-section of Engine Room Hog Island "B" Ships.

This represents British practice with freighters employing double reduction gearing. The turbine has three cylinders with a total power of 2000 S.H.P. of the **Brown-Curtis** type. The high and intermediate turbines are in series on the same shaft driving one pinion, while the L.P. cylinder drives the other pinion. They operate at 220 lbs. gage pressure and 200 deg. F. superheat, and drive a single propeller. The condenser is placed under the L.P. cylinder, as shown in Fig. 260. This is rather an unusual arrangement and requires a considerable amount of underframing.

The turbines differ somewhat from the designs previously shown and represent one **type of modern British construction.** Fig. 261 shows a cross-section of the H.P. and intermediate cylinders and also a cross-section of the L.P. turbine. Astern elements are placed in the intermediate and L.P. cylinders. The H.P. cylinder is of cast steel on account of the high superheat. Emergency governors are placed on the high and the L.P. cylinders.

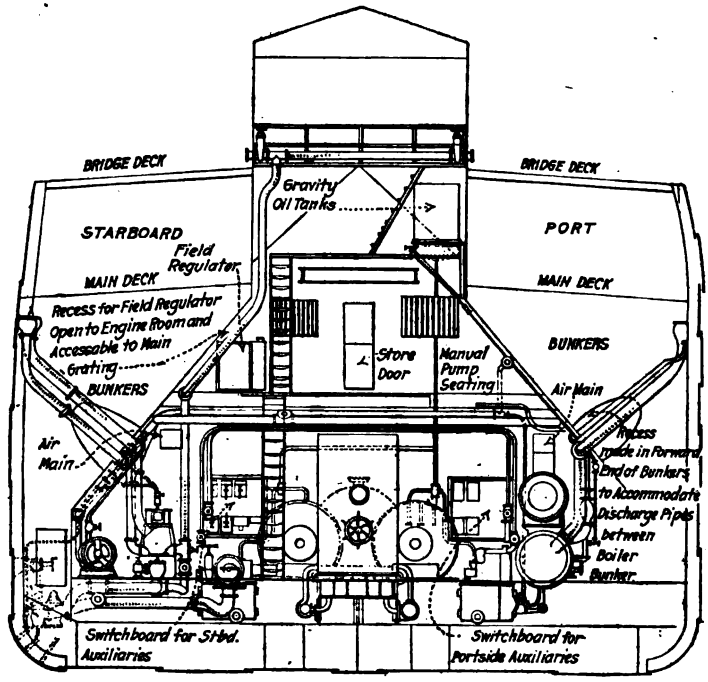


FIG. 258.—Cross-section of Engine Room, S.S. Wulsty Castle. Motors and Reduction Gears Placed in Center of Ship. One Ljungstrom Turbine Shown Above Condenser on Right Hand Side; Arrangement of Auxiliaries Shown on Left Hand Side.

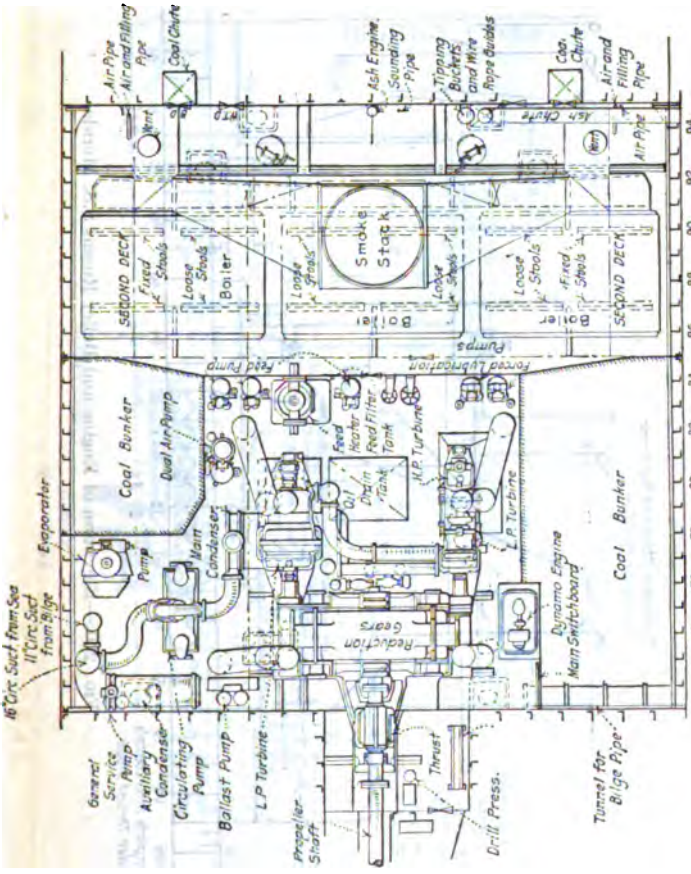


FIG. 259.—Plan of Engine and Boiler Rooms S.S. Merida.

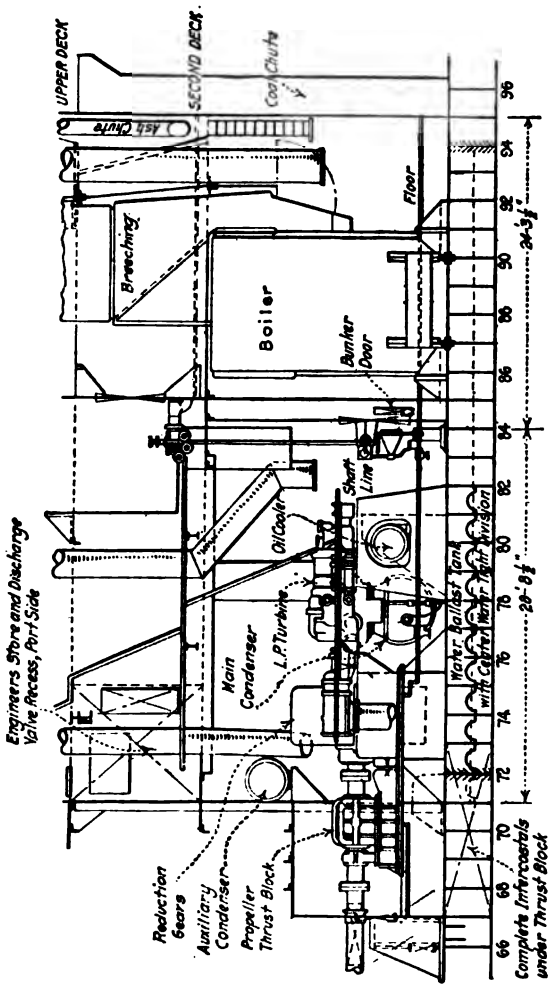


FIG. 260.—Elevation of Engine and Boiler Rooms S.S. Merida.

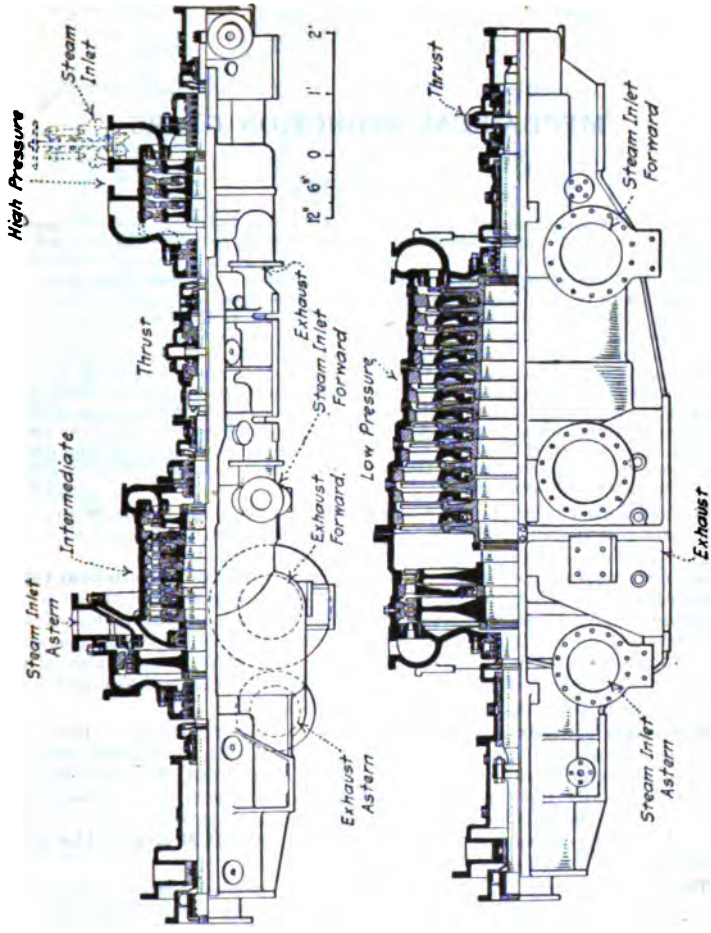


FIG. 261.—Impulse Type of Turbines on S.S. Merida.

MECHANICAL REDUCTION GEARS

BY

J. A. DAVIES

Mechanical reduction gears of the toothed gearing type have been used by engineers for many years. The application of such gearing for the propulsion of ships is of quite recent date. About the middle of the last century spur gears had been used on board ship, but these were increasing gears to permit the use of the early slow-speed reciprocating engines with the higher-speed propellers. Rapid progress in the design of both engines and propellers rendered such gearing unnecessary. Gearing was not again used for the propulsion of ships of large capacity until after the steam turbine had been developed and applied to marine work. The limitations of the direct-connected turbine prohibited its use on any but the fastest ships. In 1910 the *Westinghouse Machine Co.* in the United States built two 3600 H.P. single-pinion reduction gears for the twin screw *U. S. S. Neptune*. During the same year the *Parson's Marine Steam Turbine Company* in Great Britain equipped the single-screw ship *Vespasian* with a two-pinion reduction gear of 1100 H.P. From that date the progress of reduction gears for marine propulsion has been very rapid. Gears have been built and are building for powers exceeding 22,000 H.P. per gear.

While no fixed rules can be made at the present stage in the progress of reduction-gear design, sufficient has been accomplished to indicate the more important dimensions now in favor among gear designers.

All of the gearing now in use on board ship is of **the double-helical type**, a right-hand and left-hand helix being used to balance the end thrust which is present in such gearing. The **involute tooth contour** is universally used for such work. This is largely due to the desirable advantage possessed by involute teeth in that the correct action of the teeth is unimpaired if the centers of the pinion and gear wheel become slightly increased, as might occur due to wear of bearings.

The angle of helix used varies between 23 deg. and 45 deg. There will be more teeth in contact with the latter angle than with the former, assuming the same circular pitch in each case. On the other hand, the time required, and consequently the cost for cutting teeth of the greater angle, is greater than that for the smaller angle, assuming the same type of cutter in each instance. It would appear that a helix angle of **about 30 deg. is the most desirable** one from all points of view.

Types. There are two distinct types of mechanical gearing now in use. One is referred to as the **fixed or rigid-bearing** type (see p. 747), while the other is known as the **flexible or floating-frame** gear, Fig. 1. The latter is an invention of the late *Rear Admiral George W. Melville, U. S. N.*, and *Mr. John H. Macalpine*. It is built in this country by the *Westinghouse Electric & Manufacturing Company*. The rigid type of gearing is best known in the marine field from the work of the *Parson's Steam Turbine Company of England*. The recent unprecedented demand for geared turbine machinery for ship propulsion has brought into the marine field such other well known reduction gear manufacturers as *The De Laval Steam Turbine Company, Trenton, N. J.*, and The *Falk Gear Company, Milwaukee*. The *General Electric Company of Schenectady* have also built a large number of marine reduction gears of the

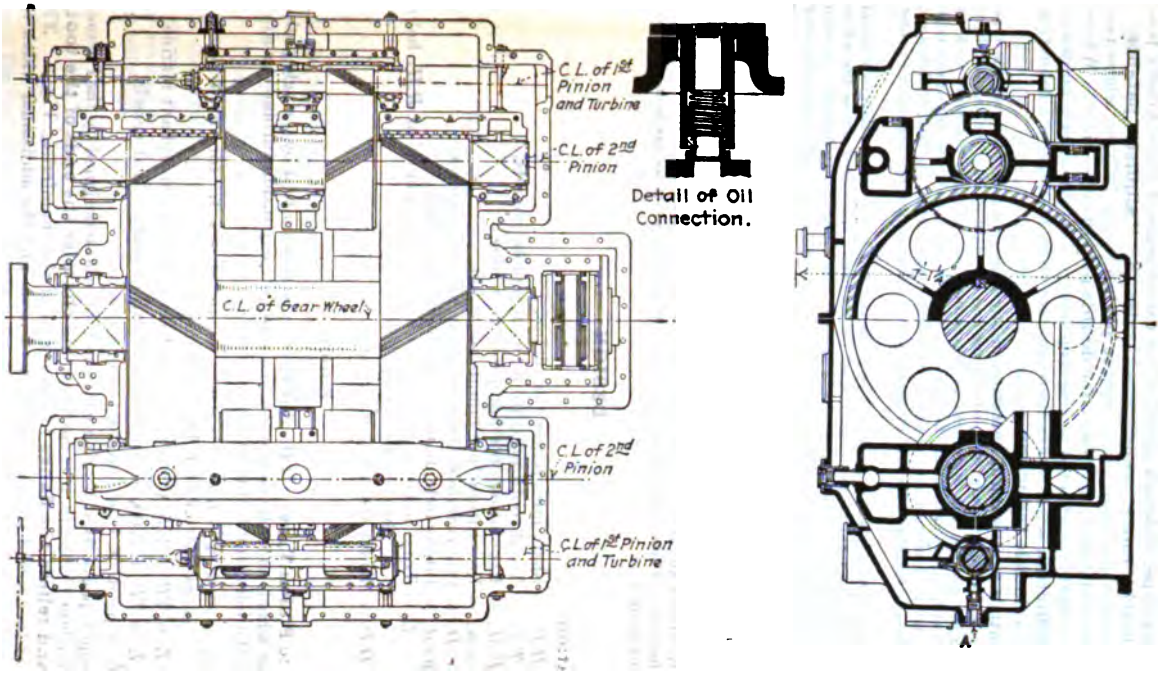


FIG. 1.—Floating Frame, Double Pinion, Double Reduction Gear.

fixed-bearing type, although many of the earlier gears of this company possessed a flexible feature manufactured under the **"Alquist" patents.**

The reduction gear may have a **single pinion** (see p. 747), or, where two turbines drive one gear, may be of the **double pinion** type, Fig. 1. Where the speed reduction is effected in one step (See p. 745), the gear is known as a **single reduction** gear; where there are two steps in the speed reduction the type is a **double reduction** gear.

The essentials to successful operation of any reduction gear whatever the type employed are extreme accuracy of tooth cutting and the maintenance of accurate alignment. In rigid gearing the alignment is determined by accurate machine work in boring the gear housings and fitting the bearings, while in gears of the floating-frame type the pinion and its bearings are mounted in a frame supported on flexible I-beams; this automatically maintains its correct position after being accurately aligned in the first instance.

The important feature which must ever be borne in mind is to so design the gearing and its housing that the total tooth pressure shall be uniformly supported across the whole width of the tooth face. Misalignment will prevent this uniform distribution. So also will excessive bending deflection of the pinion, as frequently occurs where the diameter is small and the tooth face long. This is particularly true of pinions supported by two bearings. Occasionally, where the pinion diameter is large, two bearings can be employed, but in most instances a third bearing placed between the separated helices is essential. In every case it is desirable.

As the alignment must be maintained with undiminished accuracy the design of the bearings in a reduction gear must receive proper consideration. Each detail will be separately treated and its functions briefly discussed. Useful formulæ will be given to aid the designer. These formulæ represent present practice so far as has been disclosed by existing designs.

DESIGN

Notation:

$S.H.P.$ = Shaft horsepower per pinion.
$T.P.$ = Total tooth pressure on each pinion, pounds.
$R.P.M.$ = Revolutions per minute of gear wheel.
$r.p.m.$ = Revolutions per minute of pinion.
$P.C.D.$ = Pitch-circle diameter of gear wheel, in inches.
$p.c.d.$ = Pitch-circle diameter of pinion, in inches.
L = Total axial length of tooth face; *i.e.*, length of $R.H.$ helix, plus length of $L.H.$ helix.
$D.P.$ = Diametral pitch of hob.

Pinion

Single Pinion Type. If the $S.H.P.$ is to be transmitted by a single pinion and gear wheel, as is the case when one complete expansion turbine per shaft is used, the following formulæ are applicable:

(1) $p.c.d. = C \times \sqrt[3]{S.H.P. \div r.p.m.}$, where $C = 7.37$ for fixed bearing gears and 6.72 for floating-frame gears.

(2) $L = TP + 175\sqrt{p.c.d.}$ for fixed bearing gears, merchant marine.
$L = TP + 220\sqrt{p.c.d.}$ for fixed bearing gears, naval vessels.
L = p.c.d. $\times$ 4 for floating-frame gears.

In some types of fixed bearing gears, notably those designed by the *Parson's Steam Turbine Company*, it is customary to **chamfer the ends of the tooth faces and relieve the flanks** in the region of the chamfering, Fig. 2. The length of face found from (2) is the *net* length, and the additional width re-

quired for the relieved and chamfered ends must be added to the net length to obtain the total width of the rim.

(3) $T.P. = S.H.P. \times 126{,}200 \div (p.c.d. \times r.p.m.)$

Double Pinion Type. In a single reduction gear of low ratio driven by two pinions, as for instance when a compound turbine is used for each shaft, it frequently happens that the minimum centers for the proper placement of the turbines will give a *p.c.d.* much larger than that found from (1).

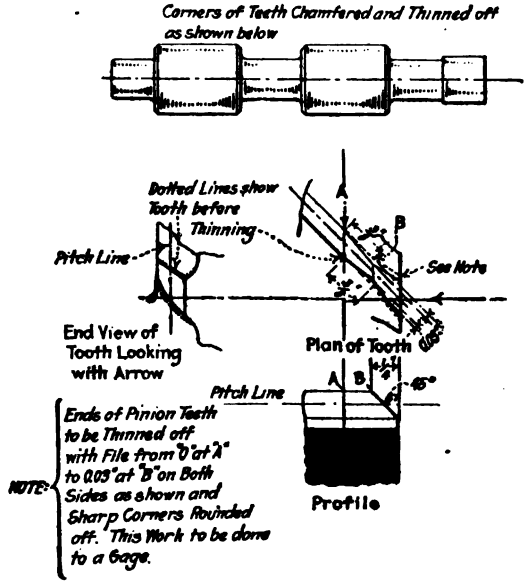

Corners of Teeth Chamfered and Thinned off as shown below
Dotted Lines show Tooth before Thinning
Pitch Line
End View of Tooth Looking with Arrow
A
B
See Note
Plan of Tooth
45°
Pitch Line
Profile
NOTE: Ends of Pinion Teeth to be Thinned off with File from "0" at "A" to 0.09" at "B" on Both Sides as shown and Sharp Corners Rounded off. This Work to be done to a Gage.

FIG. 2.

If the **revolutions per min. of each turbine is the same:**

(4) *p.c.d.* = dimension of pinion centers in inches ÷ (ratio + 1).

(5) $L = T.P. \div (p.c.d. \times 104)$ for floating-frame gears. For fixed bearing gears the formulæ given in (2) apply. It is desirable to keep the minimum dimension of $L = p.c.d. \times 2$.

If the **revolutions per min. of the two turbines are different,** which is the case with many compound turbines where the H.P. element has higher revolutions per min. than the L. P., the gear-wheel diameter is found from the following: *P.C.D.* =

$$\frac{\text{Turbine centers in inches}}{1 + \frac{1}{2}\left(\frac{1}{H.P.\text{ ratio of reduction}} + \frac{1}{L.P.\text{ ratio of reduction}}\right)}.$$

From this the pinion dimensions can be found quite readily. Having found the pinion diameters, where the turbine speeds are unequal, check both

to make sure they are not smaller than that obtained by rule (1). Should one be found smaller, it must be increased to accord with rule (1) and L made in accordance with rule (2). The gear wheel and the other pinion will then be increased in size to give the desired ratios. If both pinions are found to be larger than the dimension obtained from rule (1), select the pinion which is to transmit the greatest tooth pressure and obtain L from rule (5). Both pinions will, of course, have the same tooth face length.

Number of Teeth. $N = p.c.d. \div (D.P. \times \text{cos. angle of helix})$.

For standard tooth dimensions the minimum number of teeth that must be used to avoid interference is 31. For this reason the smallest pitch for a given *p.c.d.* is found from

$$D.P. = p.c.d. \div (31 \times \text{cos. angle of helix}).$$

If the *D.P.* thus found is not a standard size, increase the number of teeth and use the next smaller *D.P.* The *p.c.d.* may also need slight modification to permit the use of standard cutters. Having settled the *p.c.d.* the other tooth dimensions can be found as follows:

Diam. at top of teeth $= p.c.d. + (2 \div D.P.)$.

Diam. at root of teeth $= p.c.d. - (2.314 \div D.P.)$.

Pinion Journals. Max. diam. = p.c.d. $- (2.470 \div \text{D.P.})$.

The diam. thus obtained brings the surface of the journal as close to the root of the teeth as it is practically desirable to go. Nothing is to be gained by reducing the diam. below this unless the peripheral velocity at the maximum r.p.m. is too high. The maximum velocity should not exceed 100 ft. per sec.

$$\text{Total length} = \text{T.P.} \div (\text{diam.} \times 140).$$

When three bearings are used, make the center journal 0.4 and each end journal 0.3 of the total length. The length of the end journals should not be less than 0.75 of the journal diam.

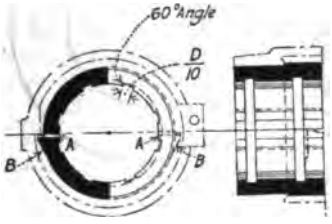
Bearings. The design of bearings for a reduction gear should receive particular consideration. The bearings for the main gear shaft can be of the **plain cylindrical type**, turned on the outside to fit a bored recess in the gear case. Cast-iron or cast-steel **shells** are preferable to bronze. The shells should be as thick and heavy as the particular design will permit. The purpose of this is to prevent the warping that is characteristic of thin shells, particularly thin bronze shells, when heated. Frequent trouble is caused by such shells as they are apt to close in at the horizontal center and grip the journal, still further raising the bearing temperature and usually resulting in damage to the white metal. The shell must be tightly held in the housing. The **bearing cap** should not be a part of the housing cover. A separate bearing cap makes the bearing more accessible and allows of adjustment when fitting the shell to its recess The pinion bearings should be provided with means of adjustment so that the correct running centers of the gears can be set without difficulty.

In some of the early gears of the fixed-bearing type, the pinion bearing shells were of the plain cylindrical type and entire reliance had to be placed on the accuracy of the machine work of the gear housing for the satisfactory operation of the teeth. While it is possible to bore a gear housing so that two sets of bearings are parallel and in the same plane, the most practical method is to provide some adjustment to one set of the bearings and permit the gear centers to be accurately placed when assembling. Furthermore this will be of distinct advantage when spare bearings have to be fitted on board ship.

The **lining** for both pinion and gear bearings should be of good quality of tin-base white metal Lead-base mixtures have been frequently used for the lower speed gear bearings, but such metals are undesirable owing to the

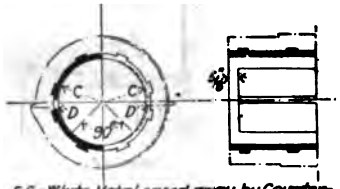
extreme care necessary when melting. The bearing shells should be provided with adequate anchor grooves for holding the white metal, which should be well peened after it is poured. While it may be possible to core the anchor grooves in the bearing shells it is generally found desirable to rough bore the shell and plane the longitudinal anchor grooves. The latter grooves are frequently dovetailed as indicated in Fig. 3, while the circumferential grooves can be cored with parallel sides. The figure shows a suitable construction for a cylindrical bearing. The side lugs on the lower half project under the bearing cap to prevent the shell from turning.

The surface of the white-metal lining must be machined and fitted so that the lubricating oil can have easy access to the whole surface. It should be borne in mind that the oil has to cool as well as lubricate the journal. The first duty demands large, well rounded oil passages and a copious supply of oil. The second is entirely dependent on the formation and maintenance



A. A- Oil Inlet Slots on each Side
B. B. Lugs on each Side to Prevent Turning.

FIG. 3



C.C.-White Metal eased away by Counterboring at each Side, this to extend to within 5/8" of each End and to leave 90° of Surface at Top and Bottom.
D.D.-Oil Passages to be well Rounded

FIG. 4.

of an oil film separating the steel journal and white-metal lining. This is all important in the bearings of modern geared turbines, the journals of which often operate at peripheral velocities as high as 100 ft. per sec. In such bearings the destruction of the oil film, due to failure of the oil or a faulty bearing surface, would result in rapid heating and wiping of the white metal. In a reduction gear this would immediately alter the alignment of the pinion and gear wheel. For this reason the designer should determine the clearances and the shape of the finished white metal surface and not leave such important details to be settled in the machine shop. A clearance of 0.002 in. per in. of diam. should be given to pinion and gear bearings. The clearance on all of the pinion bearings must be equal, for the reason that the pinion changes from the lower to the upper half of the bearing or *vice versa*, when going from ahead to astern. Obviously if the clearances were not equal the pinion would change its alignment with the gear wheel in such circumstances.

Suitable **proportions for the white-metal lining** will be found in Fig. 4.

Pinion Couplings. The coupling fitted between the turbine and gear should be designed to permit an axial movement of the pinion so that the load can be equally divided between the right-hand and left-hand helices. **Two types** are in common use In the **pin coupling**, Fig. 5, round pins on the turbine coupling head drive in bronze bushed holes in the pinion coupling flange.

The jaw coupling, Fig. 6, has claws or projections on each half of the coupling. Either type requires copious lubrication for satisfactory service, and the workmanship necessary is of a high order. All of the pins or claws must bear equally when transmitting power either ahead or astern. The pin type is the cheapest to produce. Greater accuracy of machine work is also possible with this type. Figs. 5 and 6 show typical examples of the two types.

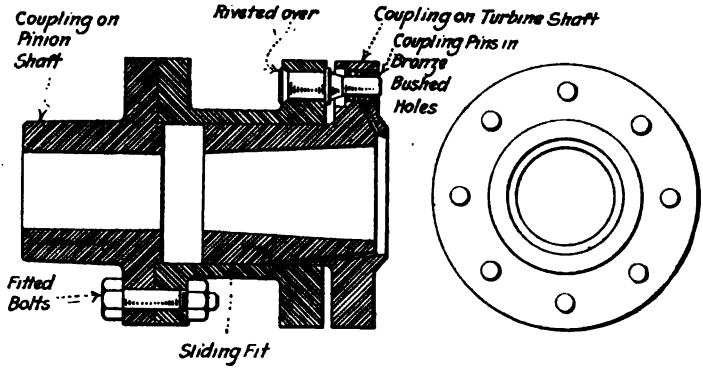


FIG. 5.

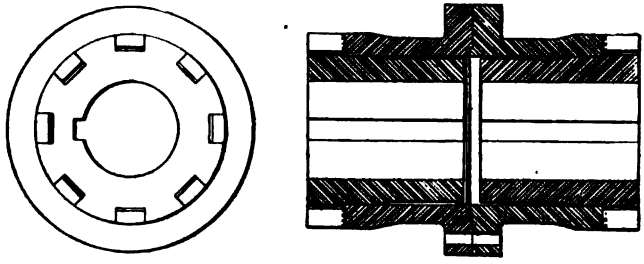


FIG. 6.

Flexible Driving Shafts. While the foregoing indicates two couplings possessing end freedom, they are not, strictly speaking, "flexible" couplings. This term is a misnomer when applied to couplings. Many so-called "flexible" couplings are on the market. The principal claim for each is that misalignment between the pieces coupled will be cared for by the feature of flexibility peculiar to the coupling used. These couplings are usually undesirable in high-powered, high-speed machinery. In any case the alignment of such machinery should be made and checked with the closest accuracy. No excuse can be tolerated for inferior workmanship in this feature. The closer the alignment the more perfect will be the operation. It is, however, possible to augment the pin or claw-type coupling with a flexible shaft,

Fig. 7, and this feature, which *De Laval* used with marked success in his turbine drives, is a desirable link to have between the turbine and pinion. It is used on all gears of the floating-frame type, principally to permit the pinion frame to function without restriction. The *Falk Co.* and the *General Electric Co.* have also used it in their gears. It is a forged-steel shaft of small diam. and considerable length. For economy of length of gear it can be fitted to the interior of the pinion which is made hollow to receive it. Usually an alloy steel possessing a high elastic limit is used. The diam. can then be reduced to a minimum without imposing excessive stresses on the material. Care must be taken in the design to avoid sudden changes of section so that the stress may be uniformly distributed.

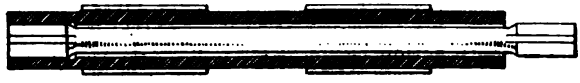


Fig. 7.

When determining the free length of such a shaft the designer should settle on a dimension that will keep the first critical speed of the shaft at least 20 per cent above the maximum running speed of the driving turbine This is important for naval vessels which have a large range of cruising speeds. Fig. 7 shows a suitable design for such a shaft. The diam. and free length can be found from the following formulæ:

$$\text{Diam. in in.} = \sqrt[3]{S.H.P. \times 18 \div r.p.m.}$$

A shaft made with these dimensions should be of chrome-nickel or equivalent alloy steel having a minimum yield point of 80,000 lbs. per sq. in. and an elongation of 20 per cent min. in 2 in. Free length of shaft, in in. =

$$\sqrt{2210 \times 10^3 \times \text{diam. in in.} \div (r.p.m. \times 1.20)}.$$

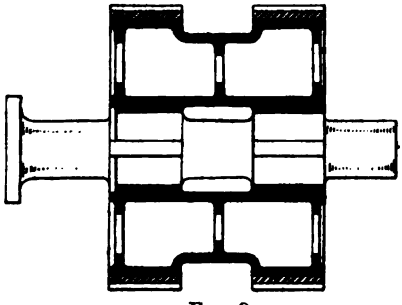


Fig. 8.

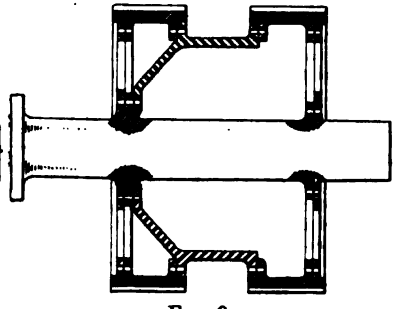


Fig. 9.

Gear Wheels

The large wheels of a reduction gear are usually built up of three pieces, a shaft of forged steel, a centre of cast steel or cast iron and rims of forged or rolled steel, Fig. 8. Occasionally where low weights are of importance, as for instance in destroyer work, the wheels have been built up from a forged-steel shaft, cast-steel hub, boiler-plate disks and forged-steel rims, Fig. 9. Such construction is extremely expensive and requires the most exacting workmanship to obtain a satisfactory wheel.

In addition to rotational stiffness, the wheels must be rigid in a fore-and-aft direction so that no displacement of the helices will occur, when under load, due to the end thrust on the spiral teeth. This **end thrust** = $T.P. \times \tan.$ angle of helix. Thus, the greater the angle of helix, the more it is necessary to support the inner edges of the rims. Figs. 8 and 9 show typical gear-wheel designs.

Teeth

The involute contour is universally used for the teeth of marine reduction gears of the double helical type, Fig. 10.

The **line of tooth action** lies on the tangent to the two base circles and passes through the tangent point of the two pitch circles. The **angle of obliquity**, or pressure angle as it is occasionally termed, is the angle formed by the line of action and the common tangent to the pitch circles. The **radius of curvature of the teeth** is zero at the **base circle** and a maximum at the **addendum circle**. The standard angle of obliquity for spur gears is $14\frac{1}{2}$ deg. Cutters for these standard teeth can be used for helical gears.

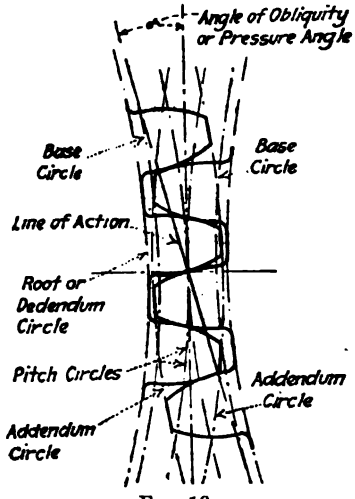


FIG. 10.

The teeth are formed either by a milling process in a hobbling machine or by planning in a specially constructed planer. The merits of both types have been the subject of much controversy, but, judging by the number used for marine gear work, the hobbing machine is to be preferred for the accurate work that is necessary in high-speed gearing. The cutting is done by a milling tool termed a "hob." It is a worm, in the thread of which approximately longitudinal gashes are made to form cutting edges. The wheel to be cut is mounted on a revolving table, which is usually horisontal, and the hob is fed vertically across the face of the gear rim. The helix angle is formed by establishing the correct rate of travel of the table carrying the work and the head supporting the hob.

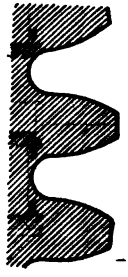


FIG. 11.

The ordinary hobbing-machine table is rotated by a worm and worm wheel, the wheel being bolted to the underside of the table, while the worm is mounted on a transmission shaft supported on the bed of the machine. This worm wheel is thus the "parent" wheel for all of the gears cut on the machine, and any errors in the division of the worm wheel will be reproduced in the work. Consequently, it is essential that the worm wheel be very accurate, as serious errors of division in a gear wheel quite prevent its satisfactory operation at the high pitch-line speeds now common.

It is essential that the **root of the teeth** be provided with substantial fillets. In some gearing the roots of the teeth are continuous radii as indicated in Fig. 11. This is a very desirable construction, as it adds materially to the strength and elasticity of the teeth. This design necessitates special hobs, which require more clearance space between the helices in two bearing pinions. As the tooth depth is greater with such teeth, the pinion journals are correspondingly smaller in diam. than is the case where standard hobs are used.

Lubrication of Teeth. The gear teeth must be copiously lubricated to obtain satisfactory operation. Oil should be introduced at the point where the gear and pinion teeth come into engagement. This is usually accomplished by fitting spray pipes above and below the pinion to lubricate both the ahead and astern faces. Successful practice indicates that a few large holes are preferable to a large number of small spray holes, as the latter are quite easily choked by small particles of lint which occasionally get past the strainers. In arranging these sprays care should be taken to firmly support any pipes that may be used, to prevent fracture due to vibration. On some gears the sprays are individually arranged so that they can be removed and cleaned, if necessary, while the gears are in operation. As such a design involves the use of valves in the oiling system, particular care must be exercised in the arrangement to prevent any possibility of closing all of the sprays.

TESTING

It is a desirable procedure to erect the gear and pinion in the housing and run them under load to determine the excellence of the teeth bearing. Such a test will disclose any irregularities in the tooth surface and these irregularities can either be scraped or ground off to leave a uniform distribution of pressure over the whole face. In making such a test it is preferable to operate

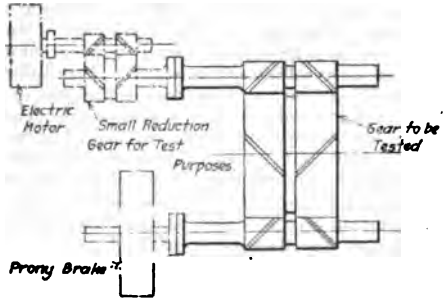


FIG. 12.

at the maximum tooth pressure and not necessarily the maximum speed. To obtain full speed as well as full power would frequently be impossible. On the other hand, full load tooth pressure can be obtained in many cases by the use of an electric motor of comparatively small power. Fig. 12 shows an arrangement for testing a two-pinion gear of 4000 S.H.P. The full speed of the pinions is 3500 r.p.m. The full-load tooth pressure on each pinion is 11,800 lbs., which can be obtained on test in the manner shown, with a 150 B.H.P. motor operating at about 580 R.P.M.

ACCIDENTS AND CHANGES DUE TO WEAR OR OPERATION

The following may occur to a reduction gear for the possible reasons assigned. Suggestions for temporary repairs, if possible, are given:

Rough Handling. It is perhaps needless to state that the utmost care must be used in handling the gears to prevent damage to the teeth. The

teeth are small in pitch and because of this they are more liable to injury if roughly handled. Should the tops of some of the teeth receive a blow of sufficient force to dent the metal, the flanks of the teeth so damaged, in the region of the damage, will be spread, increasing the thickness of the teeth at this point. It will therefore be necessary to scrape both sides of the teeth for a short distance in this vicinity, otherwise a decidedly high spot will be revealed after the gear has been in operation.

Damaged Gear Teeth. This may be caused by a foreign body, such as a nut or wrench, being left in the gears or by a defective tooth breaking out. For such an accident it is hard to prescribe beforehand. It is certain that both the pinion and gear wheel teeth will be spoiled at the damaged section. If the damage has occurred at the ends of the faces, the damaged part of the tooth face on the pinion can be turned off to just below the root. This will enable the undamaged portion of the teeth to work together without interference from the damaged parts of the gear-wheel teeth and avoids the necessity for handling the gear wheel. The pinion can then be replaced and put into operation until such time as a new pinion and wheel are obtained.

Pitting. During the first few months of service there has often been observed on the flanks of the gear teeth of large gears an action known as "pitting." Small flakes of metal break out of the otherwise perfect surface of the teeth, leaving hollow depressions. These depressions, or pit marks, vary in size from the smallest possible mark to as large as $\frac{3}{16}$ in. diam. and 0.015 to 0.030 in. deep. They may appear in any part of the tooth flank from the tip to the root, but as a general average it is most common in the region of the pitch line where only rolling action occurs. It is difficult to say whether this is merely a coincidence, as the cause does not appear to be due to this rolling action. A lack of homogenity in the material at the tooth surface or crushing of minute high spots causing fatigue and failure of small sections of the metal at such high spots are possible reasons for this pitting.

Experience shows, however, that any pitting which may take place usually occurs during the first few months of service and then entirely ceases. No trouble of any description is experienced during this period and the operation of the gear is not in any way delayed or affected.

Scoring of Tooth Surfaces. Should the lubricating-oil supply to the gear teeth sprays fail, the teeth will score and the polished flanks tear into a rough damaged surface. The absolute need for an uninterrupted supply of clean oil for the satisfactory operation of the teeth cannot be emphasized either too strongly or too often. If, after all precautions have been taken, the oil supply should fail and the teeth become scored, the gears must be thoroughly overhauled at the first opportunity. The tooth surfaces should be smoothed down with a carborundum stone and possibly scraped. If the scoring is severe the roughened surfaces may need lightly filing to remove excessive burrs and ridges. The use of a file on a tooth shape must be done with care and judgment and only a well-trained mechanic should be entrusted with the work.

The oiling system will need a thorough cleaning as flakes of steel from the damaged teeth will be found mixed with the oil.

If, after oil failure, the teeth are found to be scored and it is inexpedient to make an immediate overhaul, the oil can be restored and the gears operated without fear of breakage. This fact must not be used to delay proper repairs at the earliest moment.

Corrosion. The gear teeth and journals may show signs of rust with resulting corrosion. This is invariably due to an acid lubricant, or because

of a salt-water leak into the lubricating system, and can only be remedied by changing the oil. No harm will be experienced providing the cause is eliminated without undue delay. The corroded journals should be stoned with an oil stone and the teeth cleaned with a fine carborundum stone.

Wear of Teeth. Under normal conditions the gear and pinion teeth should not wear a measurable amount in a considerable period of time. It is not usual to keep records of the wear of gear teeth, as experience justifies the statement made in the foregoing sentence. It is natural to expect the greatest wear, if any, to occur on the pinion teeth, but any periodic measurements which might be taken should show very small variations. No excessive wear can take place without scoring, as an oil film prevents metallic contact between the surfaces of the teeth.

SECTION 6

RECIPROCATING ENGINES

BY

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CONTENTS

	Page
Classification, Development	821
Efficiency	
Thermal	827
Mechanical	830
Determining Size of the Engine	831
Proportioning of Engine Parts	837
Cylinders	840
Framing and Beds	850
Reciprocating Parts	855
Bearings	864
Shafting	871
Valve Gear	877
Reversing Gear	899
Turning Gear	904
Attached Pumps	906
Piping and Fittings	911
Drafting Room Method of Laying Down the Engine	918
Balancing	920
Analytical Method	921
Graphical Method	926
Means of Improving Balance	932
Turning Effort	934
Weights and Dimensions	938
Machining and Tolerances	939

RECIPROCATING ENGINES

BY

WAYNE T. DIMM

REFERENCES: Bragg, "The Design of Marine Engines and Auxiliaries," D. Van Nostrand Company. Seaton, "Manual of Marine Engineering," D. Van Nostrand Company. Bauer and Robertson, "Marine Engines and Boilers," The Norman W. Henley Publishing Company. Dyson, "Practical Marine Engineering," International Marine Engineering Press. Furman, "Valves and Valve Gears," John Wiley and Sons.

The theory of steam engines has been repeatedly written up, but theory without definite rules to follow is of little value to the average practical man. The intent throughout this section has been to avoid theory except where necessary to demonstrate the reason for a given practice.

American Marine Engine practice has been followed where possible, but where this does not cover the desired ground reference is made to European practice.

Marine Engines are Classified. (1) According to the arrangement of the working parts as (*a*) horisontal, (*b*) inclined, (*c*) vertical. (2) According to the number of expansions as (*a*) simple, with one or more cylinders, (*b*) compound, with two or more cylinders, (*c*) triple, with three or more cylinders, (*d*) quadruple, with four or more cylinders. (3) According to the type of propulsion as (*a*) screw, (*b*) side wheel paddle, beam or inclined, (*c*) stern wheel paddle, horizontal or inclined; and (4) according to the service of the vessel as (*a*) merchant and (*b*) Naval.

Even though several of the earlier types of marine engines had vertical cylinders, the power was transmitted to the paddle wheel shaft through a long horisontal beam and a vertical connecting rod and these engines are spoken of as horisontal engines. Inclined engines are used in paddle steamers either of the side-wheel or stern-wheel type. The modern marine screw engine of to-day is of the vertical type.

Simple engines may be either condensing or non-condensing and may have two or more cylinders working on the same crank shaft. If non-condensing, they are usually spoken of as "High-pressure" engines. Compound engines were originally built with two cylinders, but as the power increased it became necessary for the purpose of construction to use two low-pressure cylinders thus making three-cylinder compound engines. Triple expansion engines are often made with two low-pressure cylinders to secure better balance and more uniform turning moment. In very large triple expansion engines, one low-pressure cylinder would be prohibitive on account of its size. Quadruple expansion engines are not often built with more than four cylinders but there have been noteworthy installations of large quadruple expansion engines having six cylinders among which may be mentioned the American Line S. S. "St. Paul," and the Hamburg-American S.S. "Deutschland," which have six-cylinder, four-crank quadruple engines with two high-pressure cylinders over the two low-pressure cylinders.

Screw engines operate at much higher revolutions than engines driving paddle wheels and hence differ considerably from paddle wheel engines due to the difference in length of stroke.

Merchant and Naval engines differ in that no special effort is made in merchant vessels to save weight or space while in Naval vessels this is of the utmost importance.

As at present, except in certain types of paddle steamers, horisontal and inclined engines are of historical interest only, this section deals chiefly with the modern, vertical, direct-acting screw engine.

Historical Development of Marine Engines. The earliest marine engines were horizontal and for a number of years this type was used in war vessels. In these earlier war vessels, which were without armor, protection for the machinery was obtained by placing it below the water line. This fact led to the development of the many curious and ingenious types of horizontal engines, among which were the **Trunk Engine** (Fig. 1), the **Return Connecting-rod Engine** (Fig. 2), and the **Vibrating or Side-lever Engine** (Fig. 3).

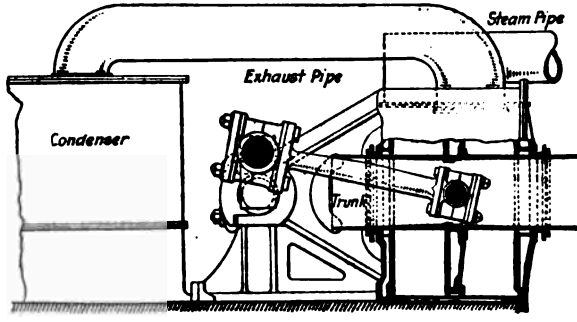


FIG. 1.—Trunk Engine.

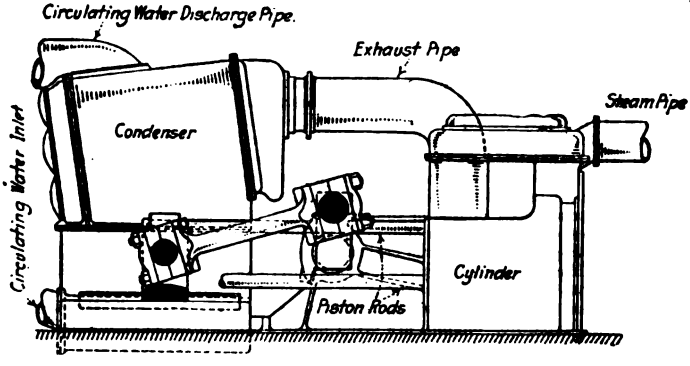


FIG. 2.—Return Connecting-rod Engine.

When the screw propeller first came into use, gearing was used between the slow-turning engine and the propeller shaft to speed up the revolutions in order to obtain proper propeller efficiency.

As the power of marine engines increased, with increase in steam pressure, it soon became evident that marine engines must be vertical in order to be economical in space. Vertical engines were used in merchant vessels long before they were adopted for Naval vessels and it was not until the advent of armor which afforded them protection and of high rotative speeds which enabled short strokes to be used that they came into use for all types of Naval vessels.

The development of the marine engine is covered briefly by R. H. Thurston in his "Manual of the Steam Engine" in which he says "The steady rise in steam pressures during the 19th century is best illustrated by Naval steam-engineering. In the time

of Watt and up to about 1840, the usual pressure in the low-pressure side-wheel engines of that period was from 4 to 7 lbs. and the rude flue-boilers then in use were of the simplest and weakest forms. By the middle of the century the fire-tubular boiler had come into quite common use, and pressures had risen to double those above stated. Between 1850 and 1860, the customary pressures in new engines had become 20 to 25 lbs., and the introduction of the surface-condenser removing the principal difficulty, the later rise in pressure was rapid and has never ceased. The use of the surface-condenser, by reducing the loss due to the deposition of the calcium sulphate contained in sea water produced a gain of 15 or 20 per cent. The type of boiler was next made the cylindrical, Scotch, form with large flues serving as furnaces and the gases returned through tubes, both flues and tubes enclosed in one cylindrical shell, and the compound

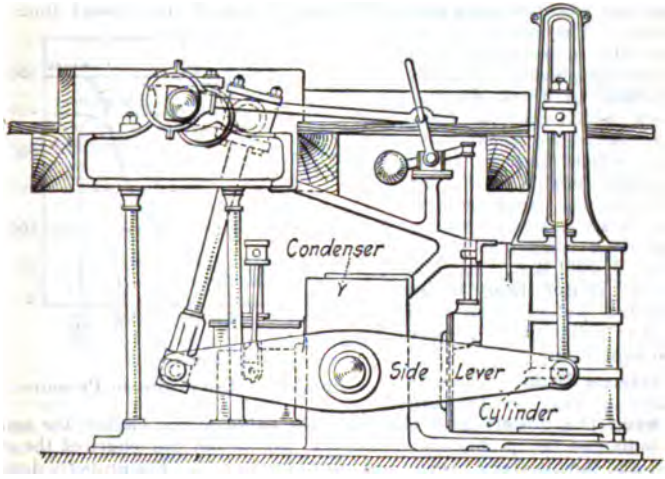


FIG. 3.—Side-lever Engine.

engine introduced, the pressures rising rapidly to 60 or 75 lbs., these changes resulting in a further economy of 30 or even 40 per cent in engines designed during the decade 1860–1870. The next ten years carried pressures for compound engines up to 90 and 120 lbs. and the triple-expansion engine, coming into use, 1875–1880, the pressure has risen one-fourth or one-third more, this type giving a gain of 15 to 20 per cent over the earlier compound engines."

A brief outline of the development of the marine engine is given in Table 1.

Table 1. Historical Development of the Marine Engine

Date	Type of boiler	Steam pressure, lbs. gage	Type of engine	Piston speed, ft. per min.
1820	Flue	4–7	Paddle	175
1840	Tubular	10–15		
1850	Flat sided	20–25	Screw	300
1860	.....	30–35	Surface condensing	
1870	Cylindrical	60	Compound	450
1880	.....	100		
1885	Water tube	130	Triple	600
1887	.....	155		
1890	.....	200	Quadruple	700
1900 to date (Merchant)	.....	250	Quadruple	900
1900 to date (Naval)	.....	300	Four cylinder triple	1000

Steam Pressure. The earliest marine engines followed land practice and worked with a steam pressure of 4 or 5 lbs. in a single cylinder with a cut-off of 0.70 to 0.85. Increase in steam pressure was slow, and it was not until about 1880 that 100 lbs. was reached. The rise in steam pressure was limited by the construction of the boilers on the one hand and by the type of the engine on the other hand. Flat surface or box boilers could not carry pressures over about 30 lbs. With the advent of cylindrical boilers (about 1860) the obstacle to the increase in steam pressure from the boiler side was removed. The limit of pressure for single-cylinder engines in the early days was about 50 lbs. As compound, triple, and quadruple expansion engines came into use, pressures rose until those in use at the present time were reached. The curve, Fig. 4, shows the increase in steam pressure graphically. Except in radical or freak designs, among which may be mentioned the steam yacht "Arrow," which had a triple expansion engine built for 350 to 400 lbs. working pressure, boiler pressures for reciprocating engines have not exceeded **300 lbs.** and this pressure **is not likely to be exceeded** unless means are found to make use of greater total expansion.

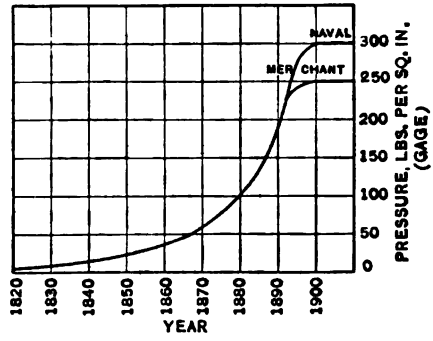


FIG. 4.—Rise in Steam Pressure.

Pressure Drop. There is always a drop in pressure **between the boilers and the throttle valve** at the engine, the amount depending on the quality and velocity of the steam, the length of the steam pipe, and the number of bends and valves in the line. For properly designed piping, this drop **should not exceed** from 5 to 8 lbs. for merchant work, and from 15 to 20 lbs. for Naval work, the larger drop in Naval work being accounted for by the fact that the distance from the boilers to the engines is usually greater, and, in order to reduce pipe sizes and save weight, velocities are higher.

There is also a drop **between the main steam pipe and the cylinder** through the throttle valve, the engine valves, and the ports. In good designs, this drop **should not exceed** from 10 to 15 lbs. in either merchant or Naval work.

Pressures in Cylinders and Valve Chests. For the purpose of designing, it is necessary to know the maximum pressure in each cylinder, and the pressures in the valve chests. The maximum pressure in each cylinder occurs at the beginning of the stroke and corresponds to the initial pressure during admission. With slide valves and with piston valves taking outside steam, the **pressure on the valve chest covers** corresponds to the initial pressures in the cylinders. For piston valves taking inside steam the pressure on the valve chest covers corresponds to the pressure of the exhaust from the cylinder or the initial pressure in the next cylinder. It is usual to express the initial pressure in the cylinders and the pressure in the valve chests as a percentage of the steam pressure. Table 2 gives the value of this factor for different types of engines, the factors being based on the steam pressure at the engine.

Table 2. Pressures in Cylinders and Valve Chests

P = Boiler Pressure (Gage); $p = P$ − drop from Boiler to Engine

Type of engine	Compound		Triple			Quadruple			
Boiler pressure	125 to 175		175 to 200			200 to 250			
Cylinder	H.P.	L.P.	H.P.	I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.
Initial pressure in cylinders	0.85p to 0.9p	0.15p	0.9p to 0.95p	0.3p to 0.35p	0.03p to 0.04p	0.9p to 0.95p	0.35p to 0.4p	0.1p to 0.2p	0.01p to 0.02p
Pressure on valve chest covers for piston valves with outside steam and for slide valves	0.85p to 0.9p	0.15p	0.9p to 0.95p	0.3p to 0.35p	0.03p to 0.04p	0.9p to 0.95p	0.35p to 0.4p	0.1p to 0.2p	0.01p to 0.02p
Pressure on valve chest covers for piston valves with inside steam	0.15p	3 to 4 lbs. abs.	0.3p to 0.35p	0.03p to 0.04p	3 to 4 lbs. abs.	0.35p to 0.4p	0.1p to 0.2p	0.01p to 0.02p	3 to 4 lbs. abs.

Expansion. The number of times the steam admitted to the H.P. cylinder of a multiple expansion engine up to the point of cut-off, increases in volume up to the point of release in the L.P. cylinder, is called the **ratio of expansion** or the number of expansions. If there were no clearance, the expansion in the H.P. cylinder would be the ratio of the total volume swept through by the piston during one stroke, to the volume of the cylinder at the point of cut-off, or $r = 1/b$, r being the expansion in the cylinder and b the cut-off in per cent of the stroke. But **clearance** must always be taken into account and its effect is to reduce the expansion in the H.P. cylinder and hence throughout the engine. The expansion in the H.P. cylinder, considering clearance, is $r = (1 + c) \div (b + c)$, where c is the clearance volume expressed as a per cent of the volume swept through by the piston. The total expansion in a multiple expansion engine is the expansion in the H.P. cylinder multiplied by the ratio of the areas of the L.P. and H.P. cylinders. This is on the assumption that the point of release in the L.P. cylinder is at the end of the stroke, which, for all practical purposes, is sufficiently accurate.

Example. Cut-off in the H.P. cylinder is 0.75, the clearance volume at each end of the H.P. cylinder is 0.12, and the ratio of the areas of the L.P. and H.P. cylinders, 8; the expansion in the H.P. cylinder, if clearance is disregarded, is $r = 1/0.75 = 1.33$, and the total expansion, $1.33 \times 8 = 10.6$. With clearance taken into consideration, $r = (1 + 0.12) \div (0.75 + 0.12) = 1.29$, and the total expansion is $1.29 \times 8 = 10.3$. The reduction in the total expansion of the engine due to this clearance is therefore 3 per cent.

As the number of expansions greatly affects the economy of the engine it is of importance, when designing an engine, that the proper expansion be selected to suit the other conditions. The total expansion to use in design depends upon the boiler pressure, the cut-off in the H.P. cylinder, and the ratio of the areas of the L.P. and H.P. cylinders. The total expansion in the best present-day practice in marine engines is given in Table 3. The **total expansion is approximately** absolute initial pressure in the H.P. cylinder ÷ 22, 17.5 and 16 for compound, triple and quadruple expansion engines.

Table 3. Total Expansions for Different Types of Marine Engines

Type of engine	Boiler pressure, lbs. gage	Cut-off in H.P. cylinder, per cent	Ratio of L.P. to H.P. cylinder	Total expansion
Compound.....	125–175	0.65–0.75	4–5	5.5–7.5
Triple (Merchant).....	175–200	0.70–0.77	7.5–8.5	9.5–11
Quadruple.....	200–250	0.65–0.72	9–12	12–15

Back Pressure. The term "back pressure" is usually applied to the pressure of the exhaust from the L.P. cylinder, although of course there is back pressure in the other cylinders due to the exhausting of the steam into the next cylinder. It is of importance to keep the back pressure in the L.P. cylinder as low as possible as this pressure acts on the piston in opposition to the steam pressure. In order to keep the back pressure low, care should be taken in the design of the L.P. cylinder to make the exhaust passages and exhaust pipes of ample area and free from obstructions and abrupt turns. The back pressure in a well-designed L.P. cylinder **should not exceed** 3 to 4 lbs. absolute. It may be found from the indicator card and is usually about 2 lbs. more than the pressure in the condenser as shown by the vacuum gage.

Vacuum. In recent years, considerable attention has been paid to the design of condensing apparatus in order to obtain a higher vacuum. Formerly it was the belief among many chief engineers that there was no gain in economy in working with a vacuum over 24 in. or 25 in., with the accompanying low temperature of air pump discharge water, as in many merchant steamers there is not enough exhaust steam from auxiliaries to properly heat the feed water. A high vacuum may reduce the steam used by the main engine, but when the total heat used is taken into consideration, in case of insufficient auxiliary exhaust steam to heat the feed water, the fuel consumed may actually increase. There is a limit to the vacuum to which existing engines, designed without special attention being paid to exhaust ports and passages, will respond, and any increase in vacuum over about 25 in. may actually prove a loss due to the increase in power required for air and circulating pumps.

Clearance. Volumetric clearance is the volume enclosed between the piston and the valve face at the beginning of the stroke and is expressed as a percentage of piston displacement. This percentage varies with the size of the engine, the ratio of diameter to stroke, the location of the valves, and the size and length of ports. The percentage of clearance is greater in a fast running engine, as the ports are made larger to avoid excessive steam velocities. The **actual clearance volumes** of an engine **are found** by measuring the amount of water or other liquid required to fill the clearance spaces. Clearance volume may also be calculated from the engine drawings with a fair degree of accuracy. **Linear or piston clearance** (see p. 844) is the distance between the piston and the cylinder at each end and may account for only a small part of the volumetric clearance. The steam in the clearance space does no work except during expansion, and as it is not practicable to carry compression completely to admission pressure a certain amount of fresh steam must be supplied each stroke to fill the clearance space; clearance therefore **increases steam consumption.** Clearance also **reduces the ratio of expansion,** see *Expansion*, p. 825, and consequently **reduces the work** done per pound of steam.

Although clearance should be kept as small as possible a certain amount is necessary at each end of the cylinder to provide space for the water which may be condensed when starting the engine, or which may be carried over from the boilers. By keeping the valves close to the cylinders and making the ports horizontal the values given under "short ports" in Table 4 may generally be realised.

Table 4. Volumetric Clearances in Per Cent of Piston Displacement

	Short ports	Medium length ports	Very long ports
H.P. cylinders with piston valves	10–12	16–18	22–26
I.P. cylinders with piston valves.....	10–12	12–14	18–22
L.P. cylinders with piston valves	10–12	12–14	16–18
L.P. cylinders with slide valves	6–8	8–10	12–14

EFFICIENCY OF MARINE ENGINES

Thermal Efficiency

The transformation of heat energy into mechanical energy or work is accomplished by a series of processes called a cycle, during which the working fluid passes through successive states of pressure, volume and temperature, which, after completion, leaves it in its initial state. The efficiency of a cycle is the ratio of the heat energy converted into useful work to the heat energy supplied.

The Carnot cycle as shown in Fig. 5 consists of two isothermal lines *ab* and *cd* representing the application and rejection of heat at constant temperature and two adiabatic lines *bc* and *da* representing expansion and compres-

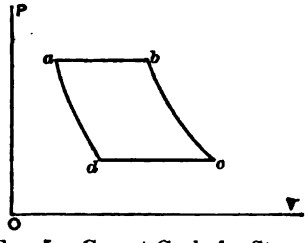


FIG. 5.—Carnot Cycle for Steam.

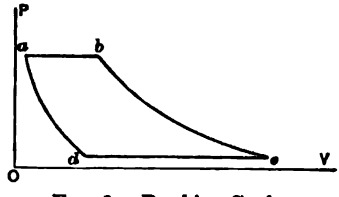


FIG. 6.—Rankine Cycle.

sion without transmission of heat to or from the fluid. It is impracticable to construct an engine to operate on this cycle but it is important as being the most efficient cycle. The efficiency of the Carnot cycle, $E_c = (T_h - T_c) \div T_h$; where T_h and T_c are the absolute temperatures of the steam at admission and exhaust respectively.

The Rankine or ideal cycle more nearly approaches the cycle of events in an actual engine. An engine operating on this cycle, Fig. 6, would receive vapor at constant pressure (*ab*), expand adiabatically to exhaust pressure (*bc*),

exhaust against constant back pressure (cd), and compress adiabatically to admission pressure (da). In such an engine the exhaust valve would not open until expansion is complete, the walls would be non-conducting, there would be no friction, no leakage past the valves or the piston and no drop in pressure through the throttle, admission and exhaust valves. The efficiency of the Rankine cycle, $E_r = q_r \div q$; where q_r = heat converted into work by this cycle (adiabatic heat drop) and q = heat supplied.

In the actual engine however the above conditions are not realised and the actual efficiency is much less than the ideal. **The actual thermal efficiency,** $E_a = q_a \div q$ where q_a = heat converted into work by the actual engine.

It is usual to compare the actual efficiency of an engine with the efficiency of the Rankine or ideal engine. This is called the cylinder efficiency or efficiency ratio, $E_r \div E_a = q_a \div q_r$, and is the ratio of the heat converted into work in the actual engine to the heat which an ideal engine (Rankine cycle) operating with the same admission and exhaust pressures, could transform into work.

All of the above efficiencies are based on 1 lb. of steam, and the heat supplied to the steam, q, is the heat supplied by the boiler and the feed water heater above the heat of the liquid at exhaust pressure.

Example: The steam consumption of a marine engine at full power is 14 lbs. per I.H.P.-hr. The pressure in the main steam pipe is 200 lbs. gage with 50 deg. superheat and the vacuum is 27 in. Find (a) Carnot efficiency, (b) Rankine or ideal efficiency, (c) actual efficiency and (d) efficiency ratio.

(a). Absolute temperature of steam at 200 lbs. gage and 50 deg. superheat, $T_h = 438 + 460 = 898$ deg. Absolute temperature of steam at 27 in. vacuum, $T_c = 115 + 460 = 575$ deg. F.

$$\text{Carnot efficiency} = \frac{898 - 575}{898} = 0.36.$$

(b) The adiabatic heat drop from 200 lbs. gage and 50 deg. superheat to 27 in. vacuum is best found from an entropy chart.

Heat content of steam at 200 lbs. gage and 50 deg. superheat	= 1232 B.t.u.
Heat content of steam at 27 in. and same entropy	= 901 B.t.u.
Adiabatic heat drop, q_r	= 331 B.t.u.
Heat content of steam at 200 lbs. gage and 50 deg. superheat	= 1232 B.t.u.
Heat content of water at 27 in. vacuum (115 deg.-32 deg.)	= 83 B.t.u.
Heat supplied, q	= 1149 B.t.u.

$$\text{Rankine or ideal efficiency} = \frac{331}{1149} = 0.29$$

(c) B.t.u. per H.P.-hr. $= \frac{33{,}000 \times 60}{778} = 2545.$

$$\text{Actual heat turned into work, per lb. of steam, } q_a = \frac{2545}{14} = 182 \text{ B.t.u.}$$

$$\text{Actual efficiency} = \frac{182}{1149} = 0.16.$$

(d) Efficiency ratio $= \frac{0.16}{0.29} = \frac{182}{331} = 0.55.$

Aside from improvement in mechanical features the **efficiency** of an engine **may be increased** by: (1) increasing the initial steam pressure; (2) reducing the back pressure; (3) reducing condensation and re-evaporation in the cylinder by (a) superheating, (b) steam jacketing, (c) compounding.

The effect of increasing initial pressure and reducing back pressure is illustrated by the curves in Fig. 7. It will be noted that a large increase in initial pressure is required to produce an effect equal to a small reduction in back pressure.

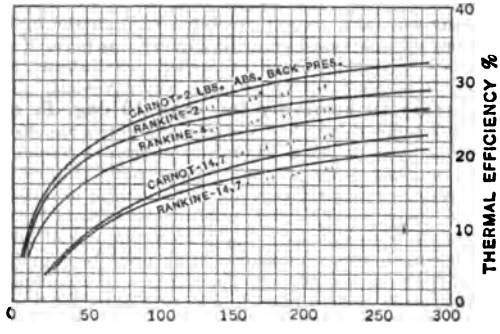


FIG. 7.—Carnot and Rankine Efficiency with Dry Saturated Steam.

Cylinder Condensation and Re-evaporation: The following treatment of this subject is from *Barton's "Naval Engines and Machinery,"* U. S. Naval Institute: Excepting the loss of heat to the condenser in the exhaust steam, initial condensation in the cylinders is the most serious loss connected with the use of steam as a working medium. During admission, expansion and exhaust, there is a complex interchange of heat between the cylinder walls and the steam, greatly increasing the steam consumption and reducing the efficiency. Steam at admission comes in contact with the metallic surfaces of the cylinder walls which have been chilled by contact with the cool steam of expansion and exhaust of the previous stroke. Condensation immediately commences and continues until the temperature of the cylinder walls approaches that of the steam. Since the throttle is still open, fresh steam is supplied to take the place of the steam liquefied, so that at the point of cut-off, much more steam has been supplied than corresponds to the volume of the admission space. After cut-off the deposited moisture and water begins to re-evaporate as soon as the temperature of the steam falls below that of the cylinder walls, and the weight of the steam present in the cylinder as steam gradually increases toward the end of the stroke. This heat transferred to the steam by re-evaporation during expansion and before exhaust does a small amount of work on the piston, but the heat transferred to the steam during exhaust goes directly to the condenser thus increasing the already large exhaust loss. In general in a single cylinder condensation and loss from re-evaporation is increased by an increase in the range of temperature between admission and exhaust.

Superheat. The practical advantages of superheated steam are primarily due to the reduction of the losses caused by cylinder condensation. If the steam is kept dry to the point of cut-off these losses are practically eliminated. This would probably require higher superheat than is otherwise desirable in marine work. In early days with the very low boiler pressures in use, considerable gain in economy resulted from the use of superheated steam as a

large quantity of heat could be added without increasing the temperature to a dangerous extent. As boiler pressures rose to about 60 lbs. the total temperature became too high for the construction and materials then in use and superheat was discontinued on the ground that the high temperature dried up the unguents, destroyed the packing in the stuffing boxes (metallic packing was unknown), and caused excessive wear in the cylinders. Later on however the use of superheat in marine engines was revived and during the past 20 years there have been many successful installations using moderate and high superheat in both Merchant and Naval vessels. There is about **1 per cent increase in economy for each 10 deg. F. of superheat.**

As steam at high temperature has a chemically corrosive effect on all brass and copper besides causing other troubles, the present tendency with reciprocating marine engines is to use moderate superheat (not exceeding 100 deg. F.) to insure dry or slightly superheated steam at admission.

With saturated steam the cylinder walls are lubricated by the condensed steam and by the oil entering the cylinder from the use of oil swabs on the rods but with superheated steam oil lubrication must be supplied and this oil must be removed from the feed water before it enters the boilers. On account of the high temperature special metallic packing is required at least for the piston rods and valve stems of the H.P. cylinder.

Steam Jackets. By constructing the cylinder with jacket spaces and filling these spaces with live steam, cylinder condensation may be reduced. The jacket steam, to be effective, must be kept in circulation and this causes a loss by condensation in the jackets which may entirely offset the gain produced by the prevention of condensation within the cylinders. Tests made to determine the economy of steam jackets have usually indicated some gain. However, it is doubtful if in every day practice there is much economy in the use of steam jackets on account of general defects in the system and lack of attention by the engineers. Also in large engines the thickness of the walls interferes with the rapid interchange of heat. But even if there be no economy in the use of steam jackets, the jacket provides a method of warming the cylinders before starting up. At present, steam jackets are very **seldom used in merchant engines,** and when used, only those cylinders are jacketed which are fitted with liners which provide the jacket spaces **In Naval work it is the practice to jacket** all cylinders. In order to prevent excessive condensation in the jackets, jacket steam for intermediate and low pressure cylinders should be reduced to a pressure corresponding to the temperature of admission steam within the cylinders.

Compounding. To obtain the full benefits of high pressures and large ratios of expansion, it is necessary that the expansion be carried out in two or more cylinders in order to reduce cylinder condensation by reducing the range of temperature in each. Furthermore the steam re-evaporated during the exhaust stroke in one cylinder is available for expansion in the next instead of passing directly to the condenser. The usual pressure ranges for compound, triple and quadruple expansion engines are given in Table 3.

Mechanical Efficiency

The mechanical efficiency of an engine is the ratio of the horsepower developed in the cylinders, I.H.P., to the horsepower delivered to the shaft, S.H.P., as determined by the use of torsion meters. The difference between the two is the **frictional loss.** The mechanical efficiency of marine engines varies from 0.90 to 0.93 at full power to 0.94 to 0.95 at low powers. *Bauer,*

in "*Marine Engines and Boilers*" cites several marine engines in which the mechanical efficiency runs from 0.91 for a 1640 I.H.P. engine to 0.985 for one of 4500 I.H.P. *Mr. Archibald Denny* in a paper read before the *Institution of Naval Architects*, March, 1907, gives a curve of mechanical efficiency of a certain marine engine which shows values of about 0.925 at 1325 I.H.P., 0.937 at 400 I.H.P. and 0.947 at 230 I.H.P. The mechanical efficiency of **engines with attached pumps** will be at least 5 per cent lower than the above figures.

Steam Consumption. It is usual to rate the economy of marine engines in terms of steam consumption per I.H.P.-hr. In general, the steam consumption per I.H.P. will be lower for engines with high ratios of expansion if properly proportioned as to ratios of cylinders, cut-off, clearance, etc., than for engines with low ratios of expansion. Other considerations which improve economy are superheating, lubrication, good workmanship and alignment, and proper care and operation. Very few tests of steam consumption of reciprocating merchant engines have been made and published but Table 7 gives the steam consumption of several merchant and naval engines of different types.

The steam consumption per I.H.P. **increases as the power decreases.** It is also usually **higher at overload than at designed full power.** These characteristics are shown by the curves in Fig. 8, in which steam consumption per I.H.P. is plotted against per cent of power, calling designed full power 100 per cent. The steam consumption per I.H.P. of merchant engines, which are proportioned to meet the conditions given in Table 3, exclusive of auxiliaries, is approximately as follows: compound 500 to 1000 I.H.P., 18 to 22 lbs.; triple 1000 to 2000 I.H.P., 16 to 17 lbs.; 3000 to 6000 I.H.P., 15 to 16 lbs.; quadruple 2000 to 4000 I.H.P., 12.5 to 13.5 lbs. The use of superheat should reduce these values about 1 per cent for each 10 deg.

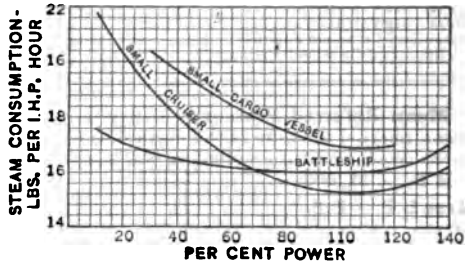


FIG. 8.—Characteristic Reciprocating Engine Steam Consumption Curves.

DETERMINING SIZE OF THE ENGINE

Size of L.P. Cylinder. As the L.P. cylinder must be of sufficient size to pass all the steam used by the engine at terminal pressure it is logical, in determining the size of an engine for a given horsepower, that the diameter of the L.P. cylinder be determined first. The length of the stroke, revolutions or piston speed (P.S.), and the mean effective pressure referred to the L.P. cylinder (M.R.P.) must first be selected. Then **area L.P. cylinder in sq. in.** $= \frac{\text{I.H.P.} \times 33{,}000}{\text{P.S.} \times \text{M.R.P.}}$. The diameters of the H.P. and I.P. cylinders are determined by the ratios of their areas to the area of the L.P. cylinder.

Length of Stroke is chosen to suit the size of the engine and the conditions under which it is to operate and is ordinarily 0.6 to 0.7 of the diam. of the

L.P. cylinder. When there are two L.P. cylinders this factor refers to the diam. of one equivalent cylinder.

Revolutions per Minute (R.P.M.) also vary with the size of the engine and the class of service. Table 5 gives the revolutions and piston speed generally adopted for reciprocating engines for vessels of various types. **Piston speed (P.S.) in ft. per min.** $= \frac{\text{R.P.M.} \times \text{stroke} \times 2.}{12}$

Table 5. Revolutions and Piston Speed of Reciprocating Engines

Type of vessel	Revolutions per min.	Piston speed ft. per min.
Tugboats and other small craft	100–200	400–550
Screw river and bay steamers	100–150	500–700
Large passenger steamers	75–100	700–1000
Large cargo steamers	70–85	550–750
Small cargo steamers	65–85	500–700
Beam paddle wheel engines	15–35	400–700
Inclined paddle wheel engines	35–60	400–700

Mean Referred Pressure (M.R.P.). In a multiple cylinder engine the mean referred pressure is the mean effective pressure that would be required if the work of all the cylinders was done in the L.P. cylinder. **M.R.P.** $= \frac{\text{I.H.P.} \times 33{,}000}{\text{Area L.P.} \times \text{P.S.}}$. The area of the piston rod is not deducted. The M.R.P. is always calculated when the power of an engine is determined from indicator cards as a means of comparison with other engines and it is also used in determining the size of the L.P. cylinder of a multiple expansion engine. The designer in determining the M.R.P. must properly coördinate the factors upon which the M.R.P. depends, viz.; the cut-off in the H.P. cylinder, the initial steam pressure, the back pressure, and the ratio of the areas of the L.P. and H.P. cylinders. The initial pressure in the H.P. cylinder depends upon the drop in pressure between the boilers and the cylinder as discussed under "Pressure Drop" on p. 824. Back pressure in the L.P. cylinder has also been discussed on p. 826. The M.R.P. to suit the given conditions is found by calculating the theoretical mean effective pressure and multiplying this by the "**card factor.**" The mean referred pressures for engines of different types are given in Table 7.

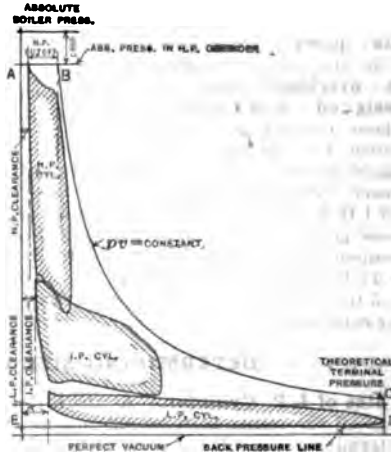


FIG. 9.—Combined Actual and Theoretical Indicator Diagram for a Triple Expansion Engine.

Theoretical Mean Effective Pressure. This is the mean pressure of admission and expansion as shown by the theoretical diagram in Fig. 9. This diagram represents the pressure and volume of the steam admitted to the H.P. cylinder, the expansion through all the cylinders, the volume of the steam at release and the theoretical pressure at exhaust. It is usual to assume that the steam expands in accordance with the law pv = constant in which case the expansion curve is hyperbolic and the theoretical mean effective pressure, $P_m = P_1 \frac{(1 + \text{hyp. log } r)}{r} - p_b$ in which P_1 = absolute initial pressure, r = total expansion, and p_b = absolute back pressure. To simplify the calculation the values of $\frac{1 + \text{hyp. log } r}{r}$ for different values of r may be taken from the curve in Fig. 10. In calculating P_m it is advisable to use a drop of about 10 lbs. between the boilers and the engine and a drop of about 15 lbs. from the throttle valve to the cylinder for the ordinary arrangement of machinery and piping. With these assumed drops the absolute initial pressure, P_1, will be 10 lbs. less than the boiler pressure by gage. Ordinarily a back pressure of 4 lbs. absolute may be assumed. The value of r is calculated as shown under "Expansion" on p. 825.

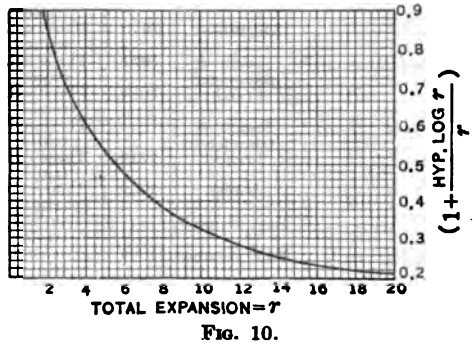


FIG. 10.

Card Factor. The ratio of the areas of the actual indicator cards to the theoretical combined card *ABCDE* as shown in Fig. 9 or the ratio of the actual mean referred pressure to the theoretical mean effective pressure (M.R.P. ÷ P_m) is called the "card factor." This factor represents the losses due to cylinder condensation, drop in pressure through ports and passages, and all other losses between the heat available in the steam and the heat actually converted into work in the cylinders. This factor varies with the number of cylinders in which expansion takes place, and various other considerations. For the usual ratios of cylinders and total expansions the values in Table 6

Table 6. Ratio of Actual to Theoretical Mean Referred Pressure (Card Factor)

Type of engine	Card factor
Single cylinder, L-P. paddle engines	0.68–0.72
Two cylinder, compound paddle engines	0.62–0.66
Two cylinder, small compound screw engines	0.58–0.66
Three cylinder, triple expansion merchant engines	0.58–0.63
Four cylinder, triple expansion merchant engines	0.57–0.62
Four cylinder, quadruple expansion merchant engines	0.60–0.64

may be used for saturated steam. Methods of calculating card factors differ in that P_1 is sometimes taken as the boiler pressure and sometimes the back pressure and clearance are neglected. With superheat the specific volume of the steam is increased and larger cylinders are required for the same power.

Table 7. Data and Performances of Marine Engines

Vessel	Type vessel	Date	Type engine	Diam. cylinders, in.	Stroke
Virginia	River steamer	....	Three cyl. triple	2[17-27-43]	24
Iona	.....	....	Three cyl. triple	21.9-34-57	39
Vespasian	Freight	1909	Three cyl. triple	22¼-35-59	42
Brookline	.....	....	Three cyl. triple	23-25-57	36
Mariana	Freight	1915	Three cyl. triple	21½-36-63	42
Meteor	.....	....	Three cyl. triple	29.4-44-70.1	48
Topila	Oil tanker	1913	Three cyl. triple	24½-41½-72	48
Madison	Freight & passenger	1911	Three cyl. triple	26½-44-74	54
Neches	Freight	1914	Three cyl. triple	29-49-84	54
Minnesota	Freight & passenger	....	Three cyl. triple	2[29-51-89]	57
El Sol	Freight	1910	Three cyl. triple	34½-57-96	60
Idalia	Yacht	1909	Four cyl. triple	11½-19-22¹ ½₆-22¹ ½₆	18
Florida	Bay steamer	....	Four cyl. triple	24½-40-47-47	42
Southland	Bay steamer	1909	Four cyl. triple	26-41-50-50	42
Birmingham	U. S. cruiser	1909	Four cyl. triple	28¼-45-62-62	36
Wilhelmina	Freight & passenger	1909	Four cyl. triple	35-58-69-69	60
South Carolina	U. S. battleship	1909	Four cyl. triple	2[32-52-72-72]	48
Delaware	U. S. battleship	1909	Four cyl. triple	2[38½-57-76-76]	48
James C. Wallace	Freight	1905	Quadruple	18½-28½-43½-66	42
Huron	Freight	1914	Quadruple	19½-28¼-41-60	42
J. D. Rockefeller	Oil tanker	1914	Quadruple	24-35-51-75	51
Pres. Grant	River steamer	....	Quadruple	2[25-36-52-75]	54
Manoa	Freight & passenger	1913	Quadruple	27-39-58-87	54
Saxonia	Freight & passenger	1901	Quadruple	2[29-41½-59-84]	54
Mongolia	Freight & passenger	1904	Quadruple	2[30-43-63-89]	60
Cap. Finisterre	Freight & passenger	....	Quadruple	2[30-44-62-90]	55
Geo. Washington	Freight & passenger	1908	Quadruple	2[38.2-56.6-79.9-112.2]	66.9
Germania	Tug	....	Compound	13-24	18
Rush	Revenue cutter	....	Compound	24-38	27
Fusi Yama	.....	....	Compound	27.4-50.3	33
Colchester	.....	....	Compound	30-57	36
Ville De Douvres	.....	....	Compound	50.1-97.1	72
Gallatin	Revenue cutter	....	Single	34.1	30
Tivoli	River steamer	....	Beam	38	102
Lancaster	River steamer	....	Beam	48	132
Smithfield	River steamer	....	Beam	51	96
Adirondack	River steamer	....	Beam	81	144

Table 7. Data and Performances of Marine Engines (Continued)

Ratio L.P. / H.P.	Cut-off H.P. Cyl.	Pressure: At boiler	Pressure: At engine	Superheat, deg. F.	I.H.P. one engine	R.P.M.	Piston speed, F.P.M.	Referred M.E.P.	Vacuum	Steam per I.H.P. per hr.
6.4	.....	200	.....	.....	1,250	190	760	37.8	.....	.....
6.78	.....	.....	165	.....	645	61.1	397	21.0	28.5	13.4
7.03	0.710	149	.....	.....	1,007	70.1	490	24.7	26.5	17.2
6.15	.....	.....	145	.....	1,137	93	558	26.3	21.7	17.0
8.58	0.752	174	164	.....	1,791	84	588	32.3	25.5	.....
5.68	.....	.....	145	.....	1,994	71.8	575	29.8	24.4	15.0
8.64	0.652	201	195	.....	2,320	73	584	32.2	28.1	.....
7.88	0.765	.....	194	.....	3,815	91	819	35.8	26	.....
8.39	0.587	197	185	.....	4,102	89	801	30.5	27	.....
9.42	.....	250	240	.....	5,970	89.5	850	37.3	23	.....
7.74	0.75	195	190	.....	5,541	75	750	33.8	26	.....
7.8	.....	.....	190 203	105	572.3 502.2	194 193.1	583 579	36.0 35.4	25.5 25.2	18.3 15.5
7.35	.....	175	.....	.....	2,582	105.4	738	38.2	.....	.....
7.41	0.815	175	173	.....	3,547	115	805	37.1	24.5	.....
9.62	.....	.....	246	.....	7,200	150	900	43.8	27.9	15.3
7.76	0.763	.....	190	.....	5,559	74	740	33.15	25.7	.....
10.15	0.795	285	275	47.5	8,826	121.3	970	37.04	26.24	.....
7.8	.....	300	286	61.6	14,289	128.4	1,025	50.9	26.31	13.38
12.75	0.574	238	234	86.9	1,559	79	553	27.2	24.5	.....
	0.574	237	235		1,741	80.3	563	29.9	24.6	.....
9.47	.....	208	.....	.....	1,731	84.9	594	34	21.2	.....
9.75	0.625	219	214	.....	2,614	71.5	608	32.1	28.7	12.8
9.00	.....	215	.....	.....	4,250	86	774	41	.....	
10.4	0.624	220	212	.....	3,959	80	720	30.5	26.2	.....
8.4	.....	199	192	.....	4,550	78	702	39	25.35	13.47
8.82	.....	205	200	.....	5,200	85	850	32.5	26	.....
9.00	.....	213	.....	.....	5,650	82	750	39.1	.....	.....
8.62	.....	213	.....	.....	10,000	83	926	36.1	.....	.....
3.41	.....	.....	104	.....	256	170	510	36.6	25.5	.....
2.51	.....	.....	69.1	.....	266.5	71	319	24.3	26.5	18.4
3.88	.....	.....	56.8	.....	371	55.6	306	20.1	25.1	21.2
3.61	.....	.....	80.5	.....	1,022	86	516	25.6	24.8	21.7
3.75	.....	.....	105.8	.....	2,977	36	432	30.7	20.3	29.8
....	.....	.....	65.4	.....	260.5	51	306	36.9	25.1	22.0
....	.....	60	59	.....	826	37.0	629	38.2	.....	.....
....	.....	55	.....	.....	810	20¼	446	33.2	.....	.....
....	0.358	60	.....	.....	703	28.9	462	24.8	.....	21.35
....	.....	55	.....	.....	4,000	26	623	.....	.....	.....

It therefore follows that smaller card factors must be used. Data from actual engines show that the card factor is reduced approximately 1 per cent for each 10 deg. of superheat. The use of steam jackets increases the card factor.

Ratios of Cylinders. The ratio of the areas of the L.P. and H.P. cylinders of a multiple expansion engine must be taken into consideration in connection with the total ratio of expansion. The ratios for different boiler pressures and ratios of expansion for well proportioned marine engines are given in Table 3. The ratios of the areas of successive cylinders, in connection with the cut-off in each cylinder, affect the power distribution in the different cylinders. In practice it is usual to determine the sizes of the intermediate cylinders more or less arbitrarily and proportion the cut-offs to suit the ratios. In triple expansion engines the ratio of the areas of successive cylinders is approximately $\sqrt{\text{area L.P.} \div \text{area H.P.}}$ and in quadruple engines approximately $\sqrt[3]{\text{area L.P.} \div \text{area H.P.}}$ However, it is usual to increase somewhat the ratios of the areas of successive cylinders, toward the L.P. cylinder approximately as follows:

	$\frac{\text{1st I.P.}}{\text{H.P.}}$	$\frac{\text{2d I.P.}}{\text{1st I.P.}}$	$\frac{\text{L.P.}}{\text{2d I.P.}}$	$\frac{\text{L.P.}}{\text{H.P.}}$
Triple	2.65 to 2.8		2.85 to 3.0	7.5 to 8.5
Quadruple	2 to 2.2	2.1 to 2.3	2.2 to 2.4	9 to 12

The ratios of cylinders for engines of different types are given in Table 7.

Cut-off. The cut-off in the H.P. cylinder regulates the total horsepower of the engine and the cut-off in the other cylinders regulates the distribution of power in the different cylinders. Decreasing the H.P. cut-off decreases the total power as less steam is admitted per stroke. Similarly increasing the H.P. cut-off increases the total power. An early cut-off in the H.P. cylinder increases the total expansion, improves the economy of the engine, and affects the size of the engine for a given horsepower. The cut-off in each cylinder after the H.P. depends on the ratio of its volume to the preceding one. In order to obtain a uniform turning moment, it is desirable to distribute the power equally on the cranks. This distribution may be obtained approximately by varying the cut-offs in the cylinders after the H.P. by adjusting the valve gear. In a triple expansion engine, if the cut-off in the I.P. cylinder is increased the back pressure in the H.P. cylinder is decreased and more power will be developed in that cylinder and less in the I.P. cylinder. Similarly increasing the cut-off in the L.P. cylinder will reduce the back pressure in the I.P. and more work will be done in the I.P. cylinder and less in the L.P. The cut-off usually used in the H.P. cylinder of different types of engines is given in Table 3.

Data and Performance of Marine Engines. Table 7 giving the sizes, proportions and performances of a number of marine engines for vessels of various types will prove useful for reference. Due to the scarcity of steam consumption data this table lists the engines of several old vessels on which steam consumption tests were made.

Example: Determine the size of a three cylinder triple expansion engine for a large cargo steamer to develop 5000 I.H.P. with 200 lbs. steam pressure at the boilers and 27 in. vacuum in the condenser. From Table 5 we will select the R.P.M. as 75 and the piston speed as 750 for an engine of this class and I.H.P. The length of stroke will therefore be $\frac{750 \times 12}{2 \times 75} = 60$ in. The next step is to determine the proper M.R.P.

to use in calculating the area of the L.P. cylinder. Assume a drop of 10 lbs. from boilers to engine and 15 lbs. from throttle valve to cylinder. The absolute initial pressure in the cylinder, P_1, will therefore be 190 lbs. Assume a back pressure of 4 lbs. absolute. From Table 3 th ratio of the areas of the L.P. and the H.P. cylinders for this pressure may be taken as 8.5 and the cut-off in the H.P. cylinder at full power as 0.75. Table 4 indicates that the clearance in the H.P. cylinder may be 12 per cent. Expansion in H.P. cylinder $= \frac{1 + 0.12}{0.75 + 0.12} = 1.29$ and total expansion $r = 1.29 \times 8.5 = 11$. Theoretical mean effective pressure, $P_m = 190 \frac{(1 + \text{hyp log } 11)}{11} - 4$. From Fig. 10 the value of $\frac{(1 + \text{hyp log } 11)}{11} = 0.31$. Therefore $P_m = 190 \times 0.31 - 4 = 55$. From Table 6 select a card factor of 0.58 as it is well in determining the size of an engine to allow some margin by assuming a safe card factor. M.R.P. then $= 55 \times 0.58 = 31.9$. Area L.P. cylinder $= \frac{5000 \times 33{,}000}{750 \times 31.9} = 6900$ sq. in. and diam. $= 94$ in. Area H.P. cylinder $= 6900 \div 8.5 = 810$ sq. in. and diam. $= 32$ in. If the diameter of the I.P. cylinder is taken 54 in., the ratios of successive cylinders will be 2.85 and 3.03.

PROPORTIONING THE PARTS OF MARINE ENGINES

Throughout this section the details of the engine are proportioned by the calculation of pressures, stresses and factors of safety as is done by the practical designer rather than by rules and empirical formulæ. As reciprocating engines are no longer being designed for the fighting ships of the Navy, the proportions given here are for engines of merchant vessels or of Naval auxiliary vessels, which, when using reciprocating engines, follow merchant practice.

Stresses in Marine Engines are kept low and large **factors of safety** are used to make the engines safe against the strains caused by rolling and pitching. Furthermore, low stresses and massive parts guard against break downs at sea with their accompanying dangers and delays.

Indicated and Effective Thrust. It is usual in designing thrust bearings for reciprocating engines to base the stresses and pressures on the indicated thrust and the proportions of thrust bearings in this section are on this basis. The actual or effective thrust may be found by multiplying the indicated thrust by 0.55 which is the combined efficiency of engine and propeller assuming the efficiency of the propeller 65 per cent and the mechanical efficiency of the engine, including shaft bearings and attached pumps, 85 per cent.

Destructive Loads for Columns. The sizes of piston rods, connecting rods and other parts acting as struts are determined by treating them as columns. This involves the ratio of length (L) to radius of gyration (ρ), and the destructive load for the material of which the part is made. The column calculation may be greatly simplified by the use of Table 8, giving the destructive loads for various values of $L \div \rho$ for different classes of material. These values are based on the results of an extensive series of experiments on iron and steel struts made by the Pencoyd Iron Works and published in their handbook. The steel used in merchant engines averages about 70,000 lbs. ultimate tensile strength and this value is used in the safe stresses and factors of safety given in this section. These figures must be corrected if steel of different ultimate tensile strength is used.

Table 8. Destructive Loads for Iron and Steel Struts

(Adapted from Pencoyd's Handbook)

L = Length of column in in. ρ = Rad. of gyration in in.

$\frac{L}{\rho}$	Wrought iron or soft steel of about 46,000 to 55,000 lbs. tensile strength			Medium steel of about 70,000 lbs. tensile strength			Hard steel of about 100,000 lbs. tensile strength		
	Fixed ends	Hinged ends	Round ends	Fixed ends	Hinged ends	Round ends	Fixed ends	Hinged ends	Round ends
20	46,000	46,000	44,000	70,000	70,000	66,900	100,000	100,000	95,600
30	43,000	43,000	40,250	51,800	51,000	47,700	74,000	74,000	69,300
40	40,000	40,000	36,500	46,000	46,000	41,900	62,000	62,000	56,600
50	38,000	38,000	33,500	44,000	44,000	38,800	60,000	60,000	52,900
60	36,000	36,000	30,500	42,000	42,000	35,600	58,000	58,000	49,100
70	34,000	33,750	27,750	40,000	39,700	32,600	55,500	55,100	45,300
80	32,000	31,500	25,000	38,000	37,400	29,700	53,000	52,200	41,400
90	31,000	29,750	22,750	36,100	34,700	26,500	49,900	47,800	36,600
100	30,000	28,000	20,500	34,200	31,900	23,400	46,800	43,700	32,000
110	29,000	26,150	18,500	33,100	29,800	21,100	44,700	40,400	28,500
120	28,000	24,300	16,500	31,900	27,700	18,800	42,600	36,900	25,100
130	26,750	22,650	14,650	30,100	25,500	16,500	39,400	33,500	21,600
140	25,500	21,000	12,800	28,200	23,200	14,200	36,300	29,900	18,200
150	24,250	18,750	11,150	26,400	20,700	12,300	34,280	26,580	15,700
160	23,000	16,500	9,500	25,300	18,100	10,400	32,200	23,100	13,300
170	21,500	14,650	8,500	23,400	15,900	9,240	29,800	20,300	11,800
180	20,000	12,800	7,500	21,400	13,700	8,030	27,400	17,500	10,300
190	18,750	11,800	6,750	19,400	12,200	6,990	25,100	15,800	9,060
200	17,500	10,800	6,000	17,900	11,000	6,120	22,900	14,100	7,860

Section Modulus is required in the calculation of bending stresses, and as the sections of the parts of engines figured as beams are usually complicated, the method of finding the section modulus is presented.

The moment of inertia and section modulus of a compound section may be obtained by treating it as a combination of elementary sections of which the properties are already known or can be found. The first step is to find the centre of gravity, which, in the case of a symmetrical section, is at the centre of the figure. For an unsymmetrical section the centre of gravity is determined by multiplying the areas of the component parts by the distances of their centres of gravity from an assumed axis, and dividing the sum of these products by the sum of the areas, which gives the distance of the centre of gravity of the whole section from the assumed axis. The moment of inertia of the compound section about an axis through its centre of gravity is the sum

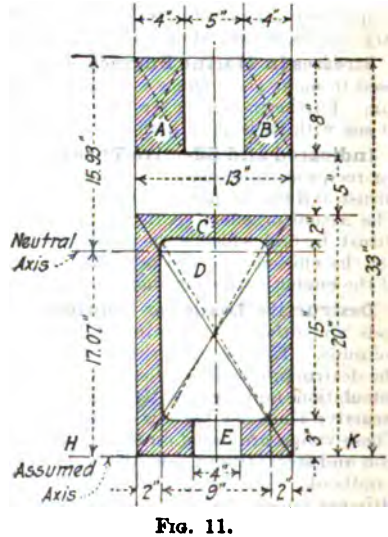


FIG. 11.

of the moments of inertia of each component part about an axis through its own centre of gravity parallel to the axis of the compound section, plus the sum of the products of each area by the square of the distance of its centre of gravity from the axis through the centre of gravity of the compound section. The section moduli for tension and compression are obtained by dividing the moment of inertia of the whole section by the distances from the neutral axis of the extreme fibres on the tension and compression sides. Table 9 gives the calculations for the moment of inertia and section moduli for the section shown in Fig. 11.

Table 9. Method of Finding Section Modulus of a Compound Section

Element	Width (b)	Height (h)	Area (A)	Distance C. of G. of element from HK (l)	Moment ($A \times l$)	Moment of inertia about its own axis I_x	Distance from C. of G. of element to neutral axis ($l - d = D$)	AD^2
A	4	8	32	29.0	928	171	11.93	4,560
B	4	8	32	29.0	928	171	11.93	4,560
C	13	20	260	10.0	2,600	8,665	7.07	13,000
D	9	15	−135	10.5	−1,417	−2,535	6.57	−5,825
E	4	3	− 12	1.5	− 18	− 9	15.57	−2,910
Total . .	..	..	177	....	3,021	6,463	.....	13,385

d = Distance of centre of gravity above assumed axis HK = 3021 ÷ 177 = 17.07 in. I_x of rectangular section = Ah^2 ÷ 12. Total moment of inertia I of section = I_x + AD^2 = 6463 + 13,385 = 19,848. Y_t = distance to extreme fibre in tension = 17.07 in. Y_c = distance to extreme fibre in compression = 15.93 in. Z_t = section modulus (tension) = $I \div Y_t$ = 1163. Z_c = section modulus (compression) = $I \div Y_c$ = 1245.

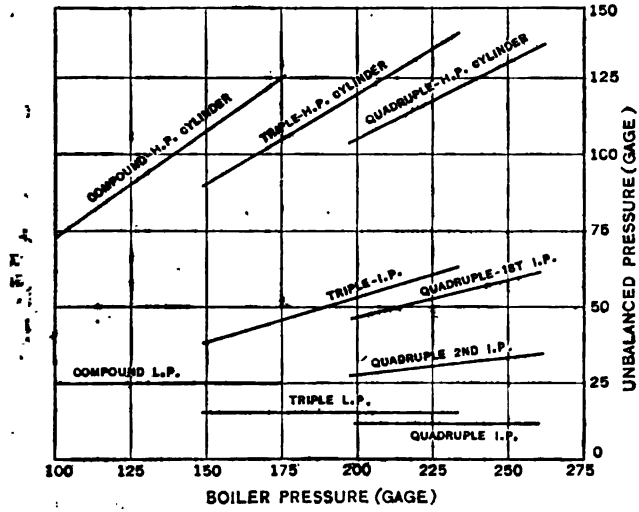


FIG. 12.—Unbalanced Steam Pressures in Marine Engines.

Unbalanced Load. The unbalanced load (P) on the H.P. piston is taken as the basis for proportioning and calculating the strength of the piston rods, cross-heads, connecting rods and housings and for calculating the pressure on the crosshead pins, crosshead slippers, crank pins and main bearings. The unbalanced pressure is the difference in the absolute pressure on the opposite sides of the piston at the time of admission and is found from the indicator card. The unbalanced load then is the area of the cylinder multiplied by the unbalanced pressure. The unbalanced loads in the different cylinders vary with the ratios of the cylinders and the valve setting, the load on the I.P. piston of a triple expansion and on the 2d I.P. piston of a quadruple expansion engine being higher than the load on the H.P. piston, but for all practical purposes the H.P. unbalanced load may be used for all cylinders. The unbalanced pressure in all cylinders except the L.P. vary with the boiler pressure and the curves in Fig. 12, taken from the indicator cards from a number of engines, show the unbalanced pressures in the cylinders for different boiler pressures. The piston rods, crossheads and connecting rods are usually made the same size for all cylinders. For the purpose of calculating the strength of these parts for the L.P. cylinders of four-cylinder triple expansion engines, it is advisable to assume the unbalanced load in these cylinders as 60 per cent of the unbalanced load in the H.P. cylinder as the actual unbalanced load often runs higher than the theoretical 50 per cent.

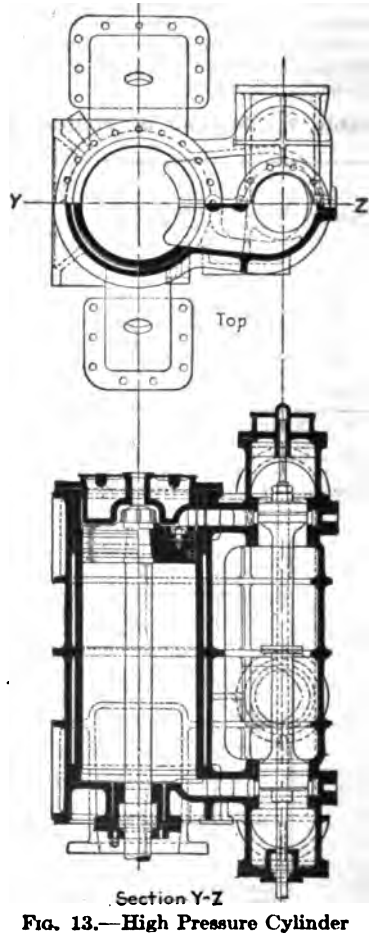
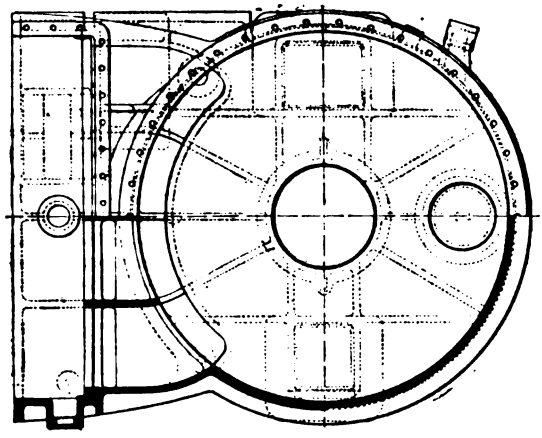


FIG. 13.—High Pressure Cylinder

Cylinders

Cylinder Liners and Barrels. The cylinder barrel, bottom and valve chest are cast in one piece which makes a complicated casting and for this reason a good quality of close grained cast iron that will run easily must be used. This quality of cast iron does not resist the wear of the piston rings as well as will a harder iron but hardness must be sacrificed to insure a sound casting. On account of the high pressure and temperature in the H.P. cylinder it is more liable to fracture and wear than are the other cylinders and so



Part Elevation and Section A – B

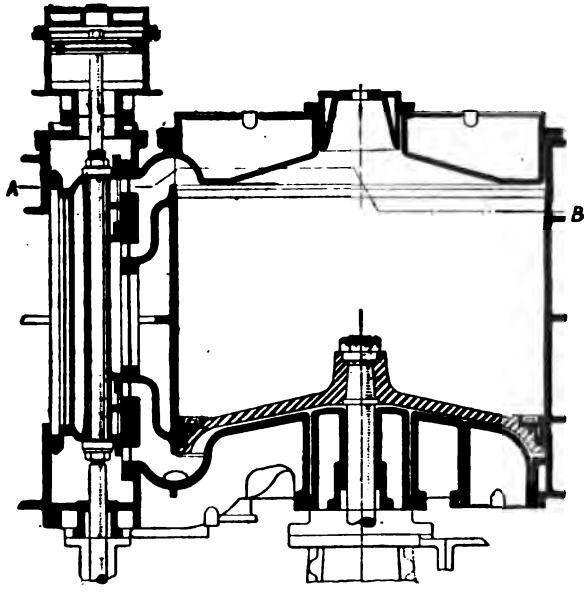


FIG. 14.—Low Pressure Cylinder.

it is customary to fit this cylinder with a close grained cast iron liner as hard as can be conveniently machined. In merchant practice, it is not usual to fit liners in L.P. cylinders, but they are often fitted in I.P. cylinders. Some owners, however, prefer to have liners in all cylinders to lessen the cost of renewal in case of fracture or serious wear of the cylinder bore.

Fig. 13 shows a **H.P. cylinder** with a **piston valve chest** and with straight and direct ports. This gives a much smaller **volumetric clearance** than is possible with the long curved ports formerly used. It will be noted that the top and bottom of the valve chest extend above and below the cylinder, but this is less objectionable than large clearances. The space between the liner and the cylinder is made small to reduce to a minimum the volume for steam in case the joints between the liner and the cylinder leak. For examining the valve laps without removing the top valve chest cover, inspection holes fitted with covers are provided on the side of the valve chest in line with the top and bottom steam ports.

Fig. 14 shows a **L.P. cylinder** from another engine with a **double ported slide valve chest** and with the ports as short and direct as is possible with this type of valve. The volumetric clearance is small and the slide valve permits the top of the valve chest to come in line with the top of the cylinder. A manhole in the bottom allows access.

Thickness of Liners and Barrels. In determining the thickness of liners and of cylinders without liners, allowance is always made for reboring. The following formula, in which P = boiler press., D = diam. in in., f = allowable fibre stress, and c = a constant for reboring, is used in determining the thickness of H.P. cylinders and liners: **Thickness** $= (P \times D) \div 2f + c$. The constant c is usually taken as $\frac{1}{8}$ in. and the stress is kept low; 1500 lbs. for small and 2500 lbs. for large H.P. cylinders is good practice. In determining the thickness of cylinder barrels in which liners are fitted, the same formula and the same stress is used but no allowance is made for reboring; in this case D = the actual cylinder barrel diameter. The thickness of different sizes of H.P. cylinder liners and of cylinders fitted with liners for a steam pressure of 200 lbs. are given in Table 10. For pressures above or below 200 lbs the thickness should be increased or decreased accordingly. It is usual to make the thickness of cylinders and liners for the other cylinders the same as the H.P. In designing cylinders much depends on the judgment of the designer in giving due consideration to factors such as the minimum thickness of a large cylinder to insure a sound casting, the ribbing of flat surfaces, and the avoidance of lumps of metal which may cause cracks or spongy spots in the casting. Table 11 gives the thickness of L.P. cylinders with and without liners.

Table 10. Thickness of H.P. Cylinders Fitted with Liners

For 200 Lbs. Steam Pressure.

Diam. of bore	Thickness cylinder	Thickness liner
15	1 1/16	1 5/16
20	1 1/8	1 1/4
25	1 1/4	1 3/8
30	1 3/8	1 1/2
35	1 1/2	1 5/8
40	1 5/8	1 3/4

Table 11. Thickness of L.P. Cylinders

Diam. of bore	If liner fitted	If no liner fitted
40	1	1 1/16
50	1	1 5/16
60	1 1/16	1 5/16
70	1 1/16	1 3/8
80	1 3/4	1 1/2
90	1 3/8	1 3/8
100	1 1/2	1 3/4
110	1 5/8	1 7/8

Thickness of Cylinder Walls. Representing the thickness of metal in the cylinder without liners by t, the thickness of the different walls of the cylinder may be taken as follows: Thickness of metal in bottom $= t$; in steam ports and valve chest $= 0.85t$; in slide valve face $= 1.3t$; in cylinder feet $= t$ to $1.1t$; in ribs in cover or bottom $= t$.

The cylinder bottom should be strongly ribbed and a good connection made to the cylinder feet as the unbalanced load is transmitted through the feet to the engine framing. Sometimes for very large cylinders or when jackets are used the bottom is made double. In large cylinders a manhole is fitted in the bottom for access as shown in Fig. 14.

Water Testing of Cylinders. After the cylinder castings are machined they should be tested by water pressure for strength and soundness as follows: H.P. cylinders to 1.5 times the boiler pressure, I.P. cylinders to 1.5 times the initial pressures given in Table 2, and L.P. cylinders to 25 lbs.

Steam Ports are made as **wide** as practicable to keep the **height** a minimum and thereby reduce the valve travel. The **area** is determined by the steam velocities given in Table 13. The steam ports are made as **short and direct** as practicable to keep the volumetric clearance small and to reduce resistance and condensation. Abrupt turns and projections in the ports should be avoided.

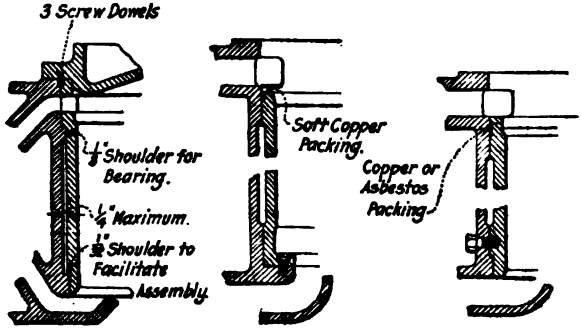


FIG. 15. FIG. 16. FIG. 17.

Methods of Securing Cylinder Liners.

Cylinder Bolting. Cylinder cover studs vary from 1¼ in. for a 15-in. cylinder to 1¾ in. for a 40-in. H.P. cylinder. Valve chest cover studs vary from 1 in. to 1½ in. for the same size cylinders. One size of studs is used for all cylinder covers and one size for all valve chest covers in the same engine. **Pitch of studs** in diameters to ensure tight joints should not exceed: 3 to 4 for H.P., 4 to 5 for I.P. and 5 to 6 for L.P. cylinders. The stress in the studs should be based on the initial pressures in the cylinders and valve chests given in Table 2. Stresses should not exceed 3000 lbs. for small and 5500 lbs. for large studs. The **flanges for cylinder and valve chest covers** should be 2.6 to 2.8 times the diameter of the studs in width and 1.4 to 1.8 times the diameter of the studs in thickness. For **bolting in the feet** see "Framings," p. 850.

Securing Cylinder Liners. It is very difficult to make joints between the cylinder and the liner which will remain steamtight after the engine is

in commission. It is therefore important that the space between the cylinder and the liner, which may be filled with steam, be reduced to a minimum. Fig. 15 shows a liner supported by a shoulder turned near the upper end. The liner extends to the top of the cylinder and is held in place by the cylinder cover and expands downward. Three screw dowels should be fitted at the top to keep it from working loose. On account of the small space between the liner and the cylinder the liner must be finished on the outside. Fig. 16 shows a liner with a flange on the lower end through which tap bolts secure it to the cylinder bottom. At the top is a dovetailed recess into which is calked a ring of soft copper. A modification of this type is shown in Fig. 17 which reduces the volumetric clearance. All of these methods allow the liner to expand and contract freely.

Counterbore. To prevent shoulders being worn in its working surface by the piston rings the liner or cylinder is counterbored about ⅛ in. larger in diameter at each end as shown in Figs. 13 and 14, the length of the wearing surface being such that the piston rings will either come flush or overrun a short distance at each end, say 0 in. to ⅛ in. at the bottom and ⅛ in. to ⅜ in. at the top. On the other hand the length of the counterbore must be limited so that when striking clearances of the piston are taken the piston rings will not entirely overrun and stick on the shoulders.

Piston Clearances. Clearance must be allowed between the piston and the cylinder at each end. This clearance is usually made larger at the bottom than at the top as the wear of the bearings in time will lower the piston. The clearances allowable should be based on the length of stroke rather than on cylinder diameter as the clearance is made the same in all cylinders of the same engine. Clearances should not be less than given in Table 12 unless the pistons and cylinder ends are machined. The cylinder cover joints will slightly increase the top clearances given.

Table 12

Stroke in in.	Piston clearances in in.	
	Top	Bottom
15 to 21	¼	3/16
22 to 31	¼	¼
32 to 41	3/16	¼
42 to 48	3/16	5/16
49 to 54	¼	5/16
55 to 66	¼	5/16

Cylinder Relief or Safety Valves. Relief valves are provided on top and bottom of all cylinders to prevent facture of the cylinder parts in case excessive pressure is built up or water hammer occurs. The diameters of relief valves for the different cylinders bear the following relations (to the nearest half inch) to the diameters of the cylinders: H.P., 0.10 to 0.115; I.P., 0.065 to 0.085; L.P., 0.045 to 0.06.

Safety valves are usually fitted to the receivers and are the same size as or slightly less than the cylinder relief valve for the corresponding cylinder. Relief and safety valves for H.P. cylinders are set at 1.10 to 1.15 times the boiler pressure, for I.P. cylinders at 1.25 to 1.5 times the initial pressures in the cylinders as given in Table 2 and for L.P. cylinders of triple and quadruple expansion engines at about 30 lbs.

A cylinder relief valve of the type generally used is shown in Fig. 18. The casing prevents the scalding of attendants by the escaping water, the exit being on each side near the bottom. If it is desired to reduce volumetric clearance to a minimum the type of relief valve shown in Fig. 19 should be used. The seat of this valve is fitted in the inner wall of the cover and the exit is through the cored passage (*a*) to a pocket in the cylinder cover. This pocket is drained by a pipe with its flange on the same level as the relief valve flange.

Cylinder Drains. Drain cocks or valves are fitted at the lowest point in each cylinder and valve chest and gear should be provided for operating them from the working platform. The drains are piped to the hot well or condenser. The sizes of cylinder drains, to the nearest quarter inch, for the different cylinders are: H.P., 0.04 to 0.055; I.P., 0.03 to 0.04; L.P., 0.02 to 0.03 times the diam. of the cylinders. The valve chest drains are usually made ¼ in. or ½ in. smaller than the corresponding cylinder drains.

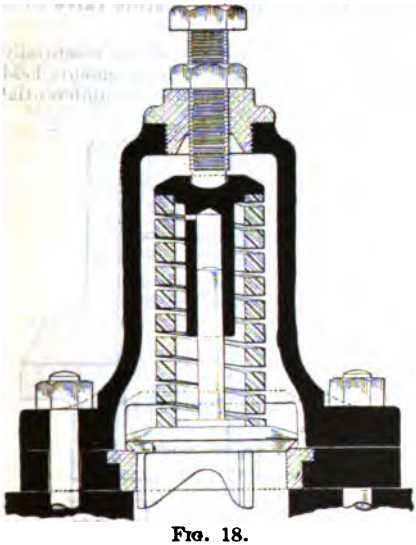


FIG. 18.

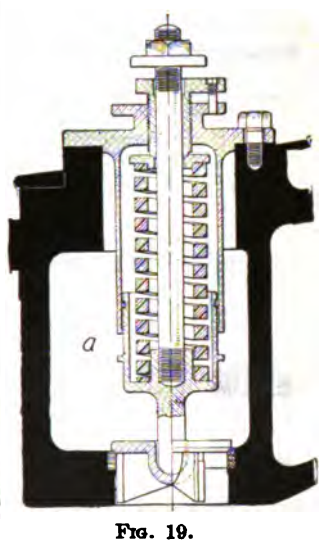


FIG. 19.

Relief Valve.

By-pass Valves. For admitting live steam to the I.P. or L.P. cylinders for starting the engine easily or for starting the engine in case the H.P. crank should be on its dead centre by-pass valves are fitted. Steam is taken from the main steam pipe or from the inlet side of the throttle valve. Two-inch to 3-in. valves are used for engines of 1000 to 3000 I.H.P. and 3-in. to 4-in. for larger engines.

Indicator Connections of ¾ in. or 1 in. for indicator pipes are provided at each end of the cylinder, care being taken that the connections are not near the cylinder ports so that the inrushing steam during admission will not effect the pressure recorded by the indicator. The three-way cock to which the indicator is connected should be located near the middle of the cylinder. Valves should be fitted close to the cylinder at top and bottom so the pipes can be closed off when the indicators are not in use.

Lagging. All cylinders and valve chests are lagged all over with 1½-in. to 2-in. magnesia block or plaster and covered over with galvanized or planished steel secured to the lagging flanges by screws.

Stuffing Boxes. The **piston rod stuffing box** is usually a separate casting bolted to bottom of cylinder, Figs. 13 and 14. The opening

thus provided permits the use of a heavy bar in boring the cylinder. The stuffing box should be designed to accommodate any of the standard makes of metallic packing to allow change in the type of packing. **The diameter of stuffing box** for metallic packing is approximately diam. of rod plus 2 in. for 4-in. rods to diameter plus 3 in. for 9 in. rods. **The depth of box** varies with the design of packing. The space required for soft packing is about the same as for metallic. The **stuffing box for a piston valve** stem is usually cast with the bottom valve chest cover, Fig. 13, and **for a slide valve** stem with the valve chest bottom, Fig. 14.

Metallic Packing. There are many types in use but all are essentially the same in principle, consisting of one or more systems of ring segments held securely against the rod by pressure induced by axial, radial or circumferential

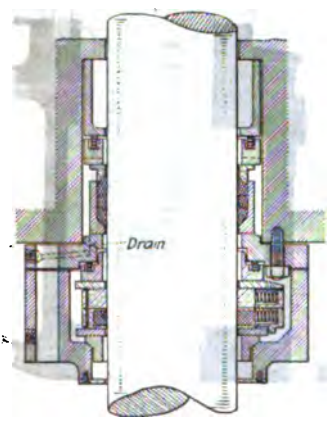


Fig. 20.

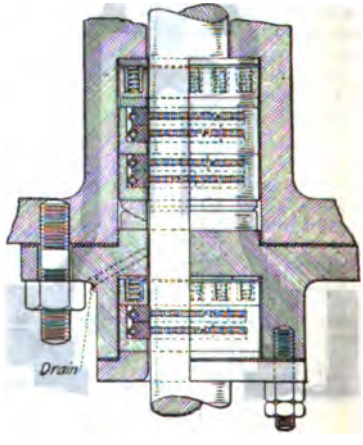


Fig. 21.

Types of Metallic Packing.

springs. Fig. 20 illustrates a typical example of double metallic packing used **for high pressure.** White metal packing rings of the inner set are held against the moving rod by axial springs and conical seatings in a retaining cup and those for the outer set by a combination of radial and axial springs without the use of the conical seating. **For lower pressure,** the outer set is often made a duplicate of the inner set. Fig. 21 is a **double packing** in which the elements of the inner and outer sets are **interchangeable.** The packing rings are held against the moving rod by axial and circumferential springs. When double metallic packing is used the chamber for holding the outer set is a separate casting bolted on the outside of the stuffing box. For H.P. cylinders using **superheated steam** or saturated steam above 190 lbs. the packing rings must be of a special composition.

A drain must always be fitted to take off the leakage or condensation which would otherwise cause an objectionable drip from the stuffing box; these drains are piped to the condenser, except the L.P. piston rod drain which is generally carried into the L.P. receiver to prevent air being drawn into the

condenser. A separator or collector is sometimes fitted to prevent cylinder oil from being drawn into the condenser. A commercial brand of soft packing is sometimes used for the L.P. piston rods and valve stems. When metallic packing is used **in a new engine** it is very important that the steam ports and passages be thoroughly cleaned, preferably by pickling; otherwise sand and grit will enter the stuffing boxes and wear the packing. Sometimes the stuffing boxes are filled with soft packing until the grit has all worked out and then the metallic packing installed. In **designing stuffing boxes** for metallic packing, it is advisable to prepare tentative sketches, giving fixed dimensions and limits, and to submit these to packing manufacturers for recommendations and criticisms in order to use a standard size. For **materials for metallic packings,** see p. 78.

Main Steam, Receiver and Exhaust Pipes and Passages. Steam velocities are based on piston displacement. Velocity in ft. per min. = area of cylinder × piston speed ÷ area of pipe or passage; piston speed is in ft. per min. and areas in sq. in. To avoid drop in pressure the areas of pipes and passages should be such that velocities will not exceed those given in Table 13 but velocities should be kept lower if possible.

Table 13. Velocities of Steam in Pipes and Passages
(Ft. per Min.)

	H.P.	First I.P.	Second I.P.	L.P.
Main steam pipe	5500			
Steam to first I. P. cylinder	....	6500		
Steam to second I. P. cylinder	....	....	7500	
Steam to L. P. cylinder	....	....	....	8500
Exhaust to condenser	....	....	....	6000
Steam through valve liner at maximum port opening	5000	6500	8000	9000
Steam or exhaust through ports of cylinder	5000	6500	7500	8500
Exhaust through total area of valve liner	4000	5000	6500	8000

On account of the increased **resistance through the throttle valve** it should be ⅛ in. to 1 in. larger in diam. than the main steam pipe. **Velocities through cylinder ports** should be fairly high to reduce condensation and permit of small clearance volumes but the **exhaust through the valve liners** should be low on account of the increased resistance. Instead of receiver pipes the exhaust from one cylinder is often led to the next through a passage cast with the cylinders. For long receiver pipes expansion joints must be fitted unless expansion can be taken care of by pipe bends. (See p. 1267.) The exhaust pipe from the L.P. cylinder to the condenser should be of ample area and as short and direct as possible as it is important that excessive back pressure be avoided. Receiver and exhaust pipes may be cast iron, steel or copper.

Cylinder Covers in merchant vessels are of cast iron with a **single wall** with deep radial ribs on the outside unless it is desired to used steam jackets in which case **double walls** are used. Double walls may also be used for covers of very large cylinders. As it is difficult to calculate the stresses in cylinder covers it is usual to proportion them from successful practice. **Thickness of metal in cover** is about $0.85t$ for small cylinders and t for large cylinders where t is the thickness of cylinder without liner. The **ribs** are the same thickness or sometimes slightly heavier. Cylinder covers are conical to follow the shape of the piston and to obtain greater strength. They

should be well ribbed to avoid too large flat surfaces between ribs. **The width of the steam joint** is about 2.8 × d and the **thickness of bolting flange** about 1.35 × d where d is diam. of the studs. In large cylinders a **manhole** is fitted in the cover, Fig. 14, so that the piston and cylinder may be examined without lifting the cover. A **false cover** of cast iron or checkered steel plate is fitted on top of the ribs and the space between ribs is filled with magnesia. If a tail rod is used, the cylinder cover is fitted with a stuffing box or a closed cast iron casing in which the tail rod works.

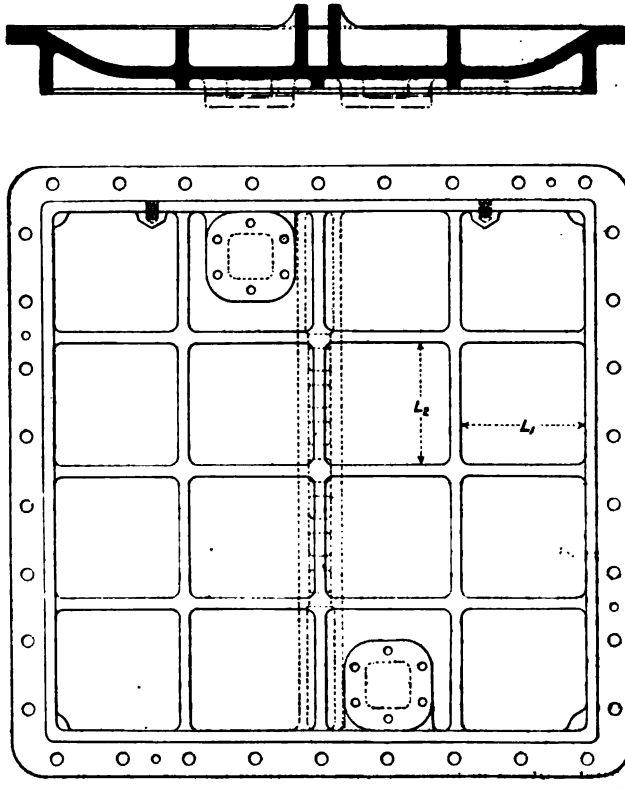


FIG. 22.—Slide Valve Chest Cover.

Valve Chest Covers for piston valve chests are of the same construction as cylinder covers. They contain the balance piston cylinder, when fitted, as shown in Fig. 14. Valve chest covers for slide valves are rectangular in shape and on account of the large area must be heavily ribbed as shown in Fig. 22. The **width of the steam joint** should be about 2.6 × d and the **thickness of the bolting flange** about 1.25 × d where d is the diam. of

the studs. The thickness of metal in cover is usually the same as the cylinder walls. Slide valves are used only on cylinders where the pressure is low and therefore the design of this cover is not as difficult as if it carried full pressure steam.

It is well however to calculate the **stress in the cover** by the following method given by *Bauer:* Experiments have shown that square or rectangular plates when subjected to a uniform strain under hydraulic pressure, split across the diagonal and should therefore be calculated for a bending stress along that line. Load on cover (see Fig. 23), $P = a \times b \times p$ where p = test press. in lbs. per sq. in. and a and b = dimensions of cover to centerline of bolts. Half of the load may be considered as acting at the centre of gravity S of half the cover; then the bending moment about the diagonal is $\frac{P}{2} \times \frac{c}{3}$. This load is counteracted by the resultant of two forces which act along the sides a and b. Then the moment about the diagonal $= M_b = \frac{P}{2} \times \frac{c}{2} - \frac{P}{2} \times \frac{c}{3} = \frac{Pc}{12}$. Expressing c in terms of a and b and $P = a \times b \times p$, then $M_b = \frac{pa^2b^2}{12a^2 + b^2}$. Stress $= \frac{Mb}{Z}$ where Z = modulus of the diagonal section found by the method given on page 838. Stresses calculated by this method may safely run as high as 6000 lbs. per sq. in. If a guide for the slide valve is cast with the cover and is a strength member it should be considered in calculating the section modulus. The pressure is considered as acting to the centreline of the bolts.

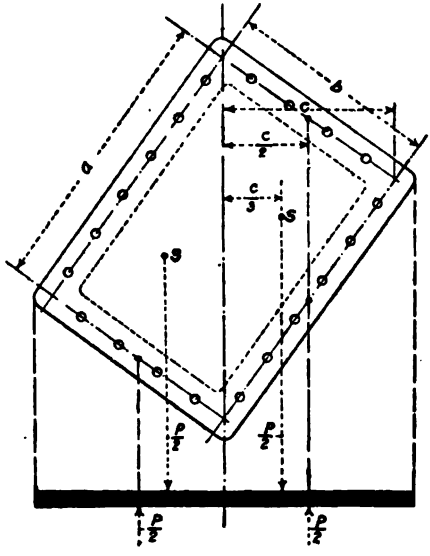


FIG. 23.

Strength of Flat Plates. In designing cylinders and cylinder and valve chest covers it is often necessary to calculate the strength of flat sections of metal between ribs. A number of formulæ for flat plates have been published but rarely is the same formula satisfactory for both thick and thin plates. In cylinder parts the flat plates are fixed around the edges and uniformly loaded and it is usual to calculate them for the test rather than for the working pressure. Probably the simplest and most satisfactory method of figuring flat plates is by equating deflections, although the strength of a flat plate can be determined only approximately. If the plate to be figured is not rectangular an equivalent rectangle should be assumed. Total load, W = pressure per sq. in. × area. Part of the total load, W_1, is carried by the edge L_2, Fig. 22, and part of the load, W_2, is carried by the edge L_1. Equating the deflections at the middle of the rectangular plate $(W_1 + W_2) = (L_2^4 + L_1^4)$. Therefore

$W_1 = WL_2^4 + (L_1^4 + L_2^4)$ and $W_2 = WL_2^4 + (L_2^4 + L_1^4)$. Stress due to $W_1 = \frac{6W_1L_1}{12L_2t^2}$ and stress due to $W_2 = \frac{6W_2L_2}{12L_1t^2}$, where t = thickness. Stress figured by this method should not exceed 2000 lbs. per sq. in. for cast iron, based on test pressure, but it will usually be much less than this.

Framing and Beds

Framing of merchant engines usually consists of cast iron housings of box section either straight or inverted Y, Fig. 24. In small engines, and in

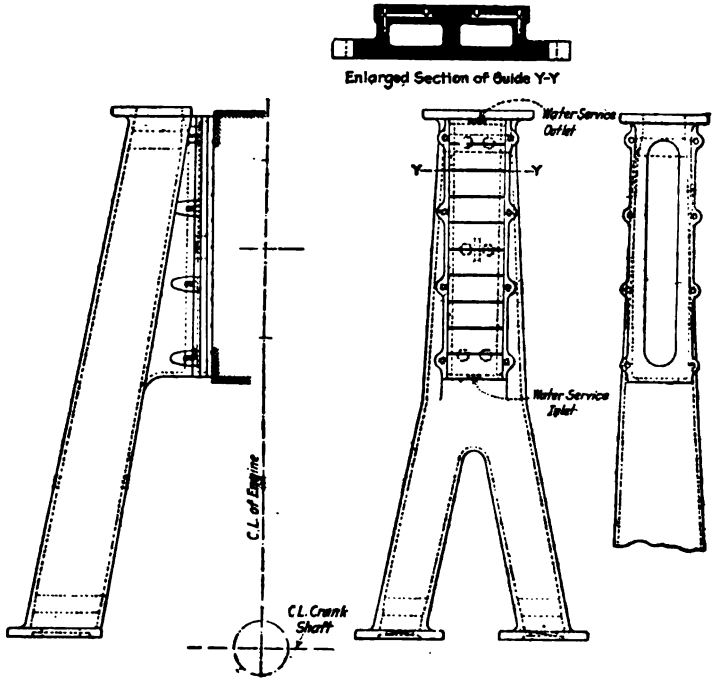


FIG. 24.—Cast Iron Housings.

larger engines where it is desired to keep the front of the engine open, forged steel front columns are used with straight or **Y box housings** carrying the crosshead guides in the back. In engines of intermediate and large sizes **straight box housings** are generally used in front and Y housings in the back, each housing carrying a crosshead guide. In very large engines four cast iron or cast steel housings may be used for each cylinder each carrying a guide. In **naval and yacht engines**, where saving of weight is important, the framing is made up of **forged steel columns** and braces. Housings are

subjected to direct tensile and compressive stress from the unbalanced load, to direct compressive stress from the weight of the cylinders, which may be greatly augmented by rolling of the vessel, and to bending stress from the crosshead pressure on the guides. As the stresses are always low, housings must be proportioned for rigidity rather than strength to avoid deflection. The bending stress from the crosshead is ordinarily only about 200 lbs. per sq. in. and need not be computed. The direct stress at the top of the housing from the unbalanced load and the weight of the cylinders should not exceed 800 lbs per sq. in. The **thickness of metal in housings** is about 1 in. for a 20-in. and 1⅝-in. for a 35 in. H.P. cylinder. Back housings should have ⅛ to ⅓ more sectional area than front housings.

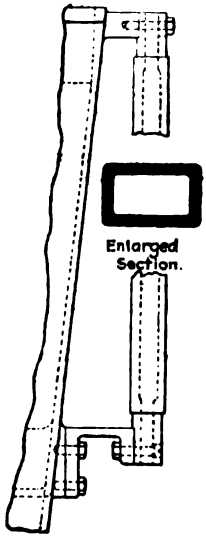


Fig. 25.

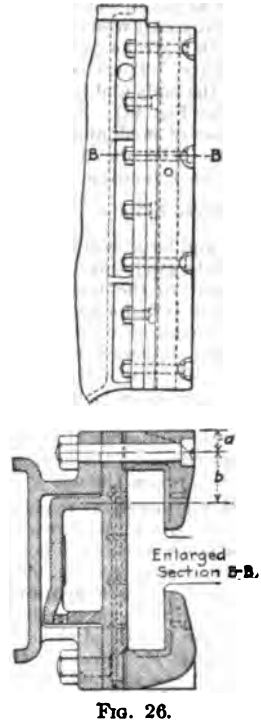


Fig. 26.

Types of Crosshead Guides.

The thickness of the metal should be increased at top and bottom to provide better attachment for the flanges. The **thickness of flanges** is about 1.4 times the diam. of the bolts. **Number and diameter of bolts** securing the housings to the cylinders and bedplates are determined from the unbalanced load, P. Stresses should be kept low to insure rigidity, about 2400 lbs for the upper and 1800 lbs. for the lower bolts being good practice. Diameters average about 1½ in. for engines with a 20-in. and 2¼ in. for a 35-in. H.P. cylinder, the lower bolts usually being made somewhat larger than the upper. Forged steel front columns are proportioned from the unbalanced load, P, considering them as columns

fixed at the ends by the method given under piston rods, p. 858. The full unbalanced load, P, should be used with a factor of safety of about 25 to allow for the weight of the cylinders and the increased load due to rolling of the vessel. If the main condenser is built in as a part of the engine framing, the back housings rest on top of the condenser.

Crosshead Guides. Small engines usually have guides of the **bar type**, Fig. 25. This type is favored for tug boat and other engines which do a lot of backing as the surface is the same for backing as for going ahead. The guide is generally cast iron of box section for water circulation, but when used on engines with a stroke as long as 42 in. cast or forged steel should be used to insure rigidity. The strength of the bar should be computed as a beam supported at the ends and loaded in the middle by the load on the crosshead slippers, P_2 (See Fig. 44). Stress for cast iron should not exceed 2750 lbs. per sq. in. Engines of small and intermediate size often have guides of the **single type**, Fig. 26. The ahead guide, which is hollow for water circulation, is bolted to the back housing, the same bolts also securing the backing guides. The load on these bolts $= \frac{P_2(a+b)}{2a}$ in which a and b are as shown in Fig. 26.

Intermediate and large engines usually have **guides on both front and back housings.** The guides are usually separate castings, cored for water circulation and bolted to the housings, Fig. 24. Very large engines with four housings to each cylinder have a guide on each housing. The **surface of the guides** is proportioned to suit the area of the crosshead slippers and the **length** is such that the slippers overrun about 1 in. at each end to prevent wearing of shoulders. The bar and single slipper types of guides require no adjustment to allow for expansion of the cylinder; this is required with guides on both housings, the adjustment being made in the latter type, to prevent crossheads from pounding after the engine has been running for some time. The working surfaces of the guides are not lined with white metal, but usually have grooves to retain the oil.

Bedplates for merchant engines are cast iron of box section with either flat or dropped bottom as shown in Figs. 27 and 28. The **dropped bottom type** is preferable as the housings are shorter and the shaft lower while ample space is allowed for building rigid foundation girders under the bedplate. Bedplates **are proportioned** for mass and rigidity rather than for strength, but it is usual to calculate the stress in the cross girders as beams supported at the middle of the side girders and loaded in the middle by the unbalanced load, P. Stress $= Pl + 2 \times 4R$ where R is the modulus of section found by the method on p. 838. To insure rigidity, stress should not exceed 1200 lbs. The **overall dimensions of bedplates** and the distance of the side girders and bottom from the centre of the shaft, to insure sufficient clearance from the path of the connecting rod, are determined when laying down the engine. The **depth of the side girders** is from 1.5 to 2 and the width from 1.25 to 1.6 times the diam. of the shaft. The **width of the cross girders** is 2 in. to 4 in. less than the lengths of the main bearings which they carry. The **thickness of metal in bedplates** for an 8-in. crank shaft is about 1 in. on top and bottom and ¾ in. on the sides. For an 18 in. shaft the corresponding figures are 1½ in. and 1¼ in. The **thickness of flanges** for the holding down bolts should be 1.4 to 1.8 times the thickness of the adjoining metal and the width about 2.75 times the diam. of the bolt. **To facilitate casting and handling,** bedplates are often made in two or more sections bolted together. If the main condenser is built in as part of the framing, it may rest on top of the bedplate or extend to the foundation

in which case the bedplate is bolted to it. Bedplates are secured to the foundations by bolts passing through cast iron or steel liners of thickness to suit

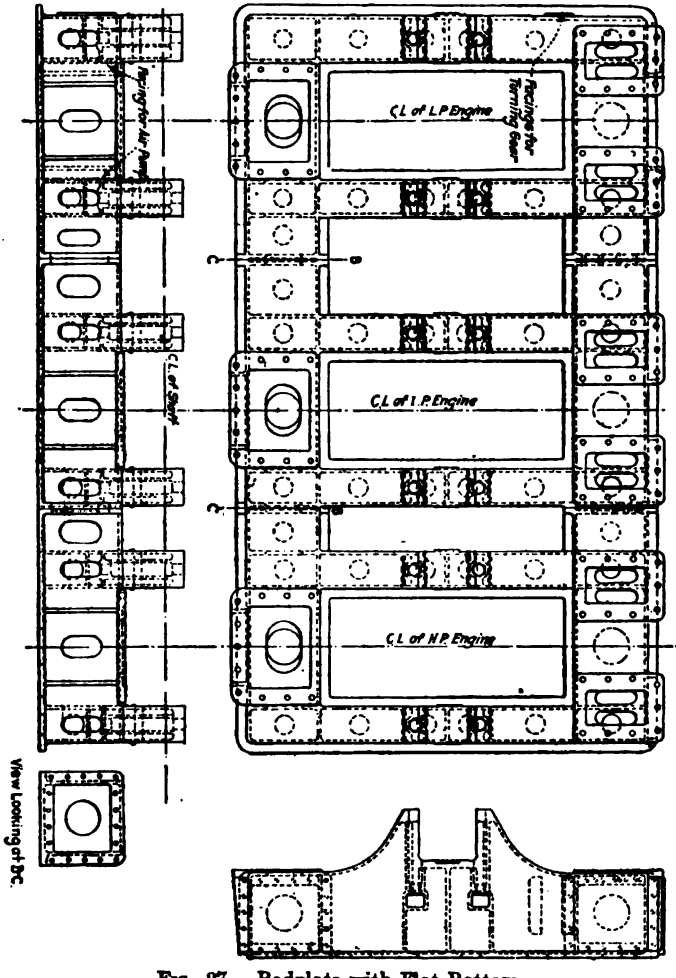


FIG. 27.—Bedplate with Flat Bottom.

the alignment. The size and number of holding down bolts must be sufficient to keep the engine in alignment when the vessel is rolling. The diameters run from 1⅛ in. for a 1500 to 1½ in. for a 7500 I.H.P. engine;

pitched 8 or 9 diameters. A certain number should be fitted bolts or all of them should be a close fit. Sometimes cast steel chocks, riveted to the

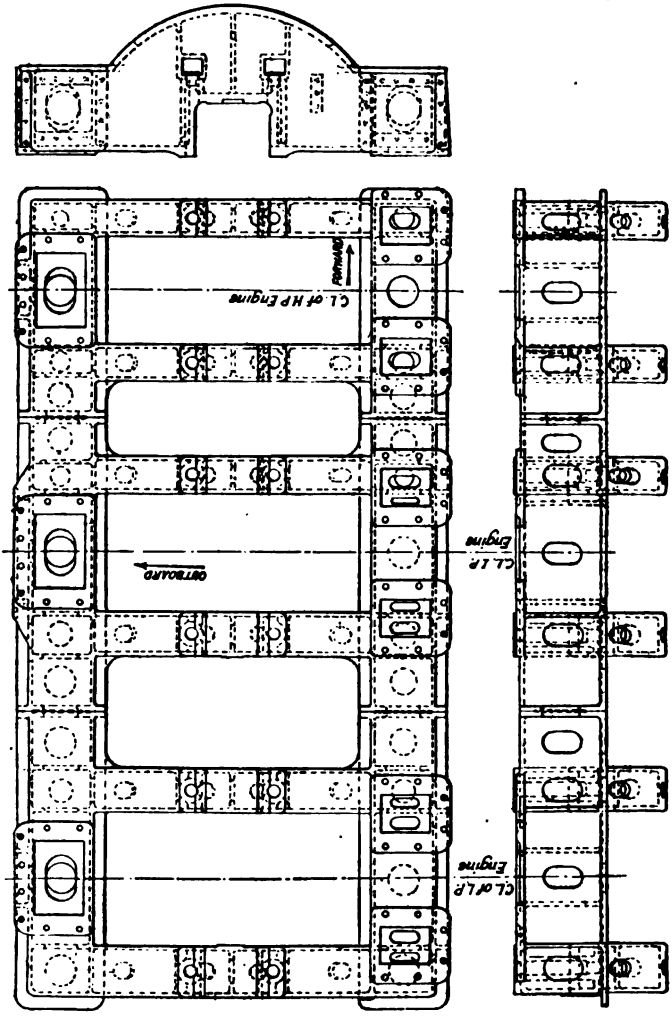


FIG. 28.—Bedplate with Dropped Bottom.

structural foundation, are fitted at each corner of the bedplate in lieu of fitted bolts.

Reciprocating Parts

Pistons may be of **cast iron of box section, Fig. 29,** or of **cast steel conical in form, Fig. 30.** Either form combines lightness and strength. As the relative weights of the different pistons greatly affect the balance of the engine, the H.P. piston is often made of cast iron almost or entirely solid, and the L.P. piston of cast steel as light as consistent with the necessary strength. The curves in Figs. 31 and 32 give **proportions of cast iron and cast steel pistons.** If the H.P. piston is of cast steel it should be much thicker than indicated by the curves to increase the weight. It is usual to make the depth of face and depth of hub the same for all cylinders. In

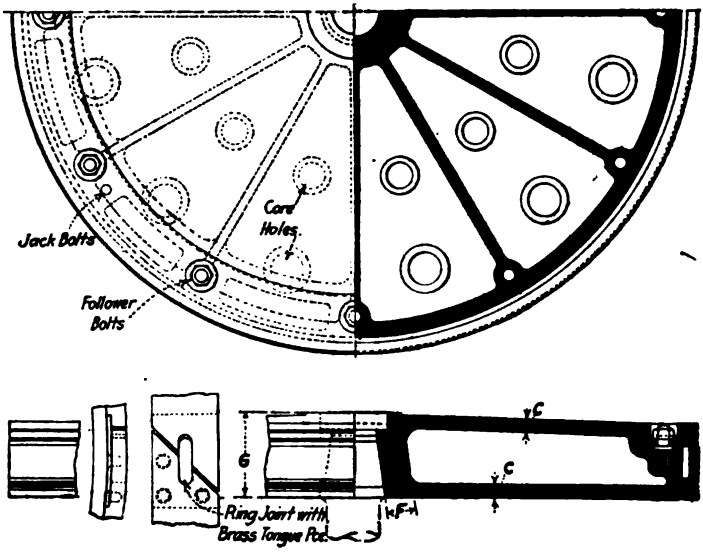


FIG. 29.—Cast Iron Piston.

hollow cast iron pistons the top, bottom and ribs are generally made the same thickness and the number of follower or junk ring bolts corresponds to the number of ribs. For solid cast iron and for cast steel pistons the pitch of the bolts in diameters is 6 for H.P., 9 for I.P. and 12 for L.P. pistons. Bolts and nuts must be securely locked against backing out and should not project above the follower face. To prevent bolts from rusting fast in the piston, brass plugs are sometimes fitted in the piston into which the bolts are screwed. Piston castings up to 35 in. diam. should be turned about $\frac{1}{32}$ in. less than the diam. of the cylinder bore and larger pistons about $\frac{1}{16}$ in. The top and bottom walls of a cast iron piston must be of sufficient strength to withstand the shock due to water collecting in the cylinder. The strength of a cast steel piston should be calculated by considering half the piston as a cantilever loaded at the centre of pressure and breaking through the centreline. The

cross-section is divided into a number of triangles or rectangles as shown in Fig. 33 and the section modulus found by the method on p. 838. Stress $= \frac{P \times A \times r}{2 \times 2 \times Z}$ in which P = press. per sq. in., A = total area of piston in sq. in., r = rad. to centre of press. = 0.424 × rad. of piston, and Z = modulus of half section. Stress figured by this method should not exceed 3000 lbs. per sq. in. It is usual to use a pressure of 30 lbs. in the calculation for L.P. pistons.

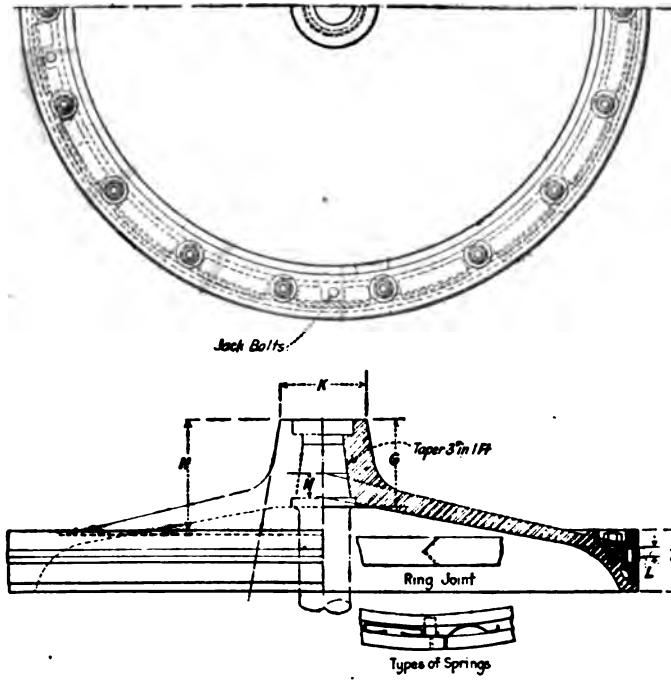


FIG. 30.—Cast Steel Piston.

Piston Rings. No parts of the reciprocating engine require more care in their design than the piston rings, the function of which is to prevent leakage of steam past the piston by exerting a uniform pressure all around without excessive friction. To accomplish this was a relatively simple matter when low pressure wet steam was used, but with high pressure and superheated steam the duties of the piston rings, especially for H.P. cylinders, have become much more exacting. The unequal expansion of the piston and the cylinder walls and the necessity of providing for wear render some means of adjustment necessary.

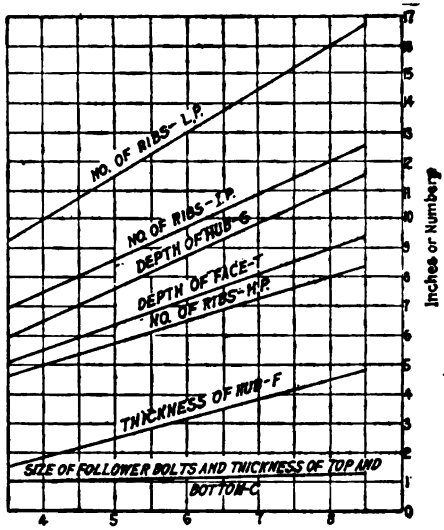

NO. OF RIBS-L.P.
NO. OF RIBS-I.P.
DEPTH OF HUB-G
DEPTH OF FACE-T
NO. OF RIBS-H.P.
THICKNESS OF HUB-F
SIZE OF FOLLOWER BOLTS AND THICKNESS OF TOP AND BOTTOM-C
Inches or Numbers
Diam of Piston Rod, Inches.

FIG. 31.—Proportions of Cast Iron Pistons (Letters refer to Fig. 29).

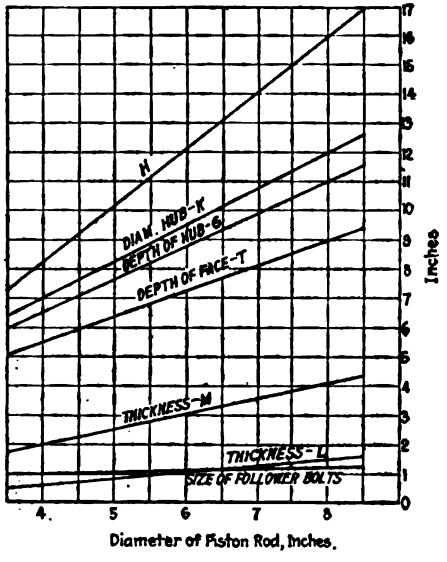


FIG. 32.—Proportions of Cast Steel Pistons. (Letters refer to Fig. 30).

Ordinary **concentric rings** of square or rectangular section, cut and sprung into position, do not exert a uniform pressure but bear much harder near the ends. This has a tendency to wear the cylinder bore oval. To overcome this and at the same time to improve the steam joint by making the pressure of the rings uniform all around the cylinder wall, springs are often fitted behind the rings as shown in Figs. 30 and 34. Rings of this type should not be used as **floating snap rings** unless they are restrained by clamping of the follower or by **non-elastic bull rings**, which do not bear against the cylinder walls, Figs. 30, 35, 36 and 37. Holes are tapped into the bull ring so that it may be removed with the rings without drawing the piston.

Eccentric or **ramsbottom rings**, thicker in the middle than at the ends, must be used for all snap rings. To prevent steam leaking past them, the rings are usually dowelled together so that the joints will come on diametrically opposite sides of the piston, and in addition a brass tongue piece is sometimes fitted as shown in Fig. 29. Piston rings are always made of hard cast iron to resist wear. Fig. 37 shows a very satisfactory type of piston packing for high pressures in which the rings are so fitted that when the follower bolts are loosened the rings will expand and fill the cylinder bore and when the bolts are tightened the rings will be pinched in position. This arrangement gives the effect of an accurately fitted solid piston with provision for adjustment. Figs. 35 and 36 show other satisfactory types of H.P. packing rings. In these the rings float but the amount is limited by shoulders on the bearing rings. Figs. 30 and 34 show satisfactory types of I.P. and L.P. piston rings and Fig. 29 a very satisfactory L.P. packing ring. There are a number of patented piston packings of various designs in use.

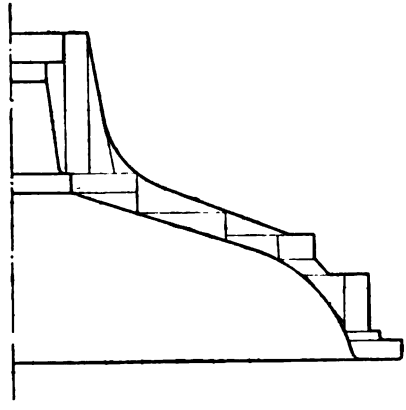


FIG. 33.

Piston Rods. The **length** is found when laying down the engine and is usually about 1½ times the stroke. The rods for all cylinders are made **interchangeable**, if possible. The **diameter** is determined from the unbalanced load, P, treating the rod as a column with round ends. The stress in the body of the rod $= P \div$ area and is usually about 2000 lbs. per sq. in. The factor of safety, which is usually about 20, is found by dividing the destructive load per sq. in., from Pencoyd's Tables given on p. 838, by the stress in the rod. These destructive loads are based on $L \div \rho$, where L is the length of the column and ρ, the least radius of gyration both in inches. $\rho = D \div 4$ for solid rods and $\sqrt{D^2 + d^2} \div 4$ for hollow rods in which $D =$ diam. of rod and $d =$ diam. of hole. The stresses are kept low and the factors of safety high, to permit turning down if the rods should be scored by the metallic packing. The piston rod is **tapered** about 3 in. per foot on diam. in the piston and 1 in. in the crosshead, Fig. 38. The large taper facilitates

the removal of the piston. A collar at the end of the taper against which the piston rests prevents this large taper from splitting the piston. The diameter of the thread at each end is about 0.7 to 0.8 times the diam. of the rod

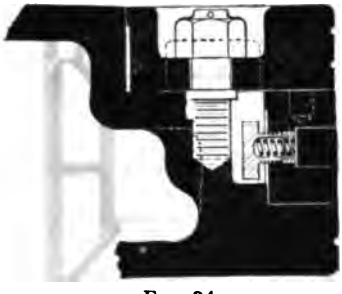


FIG. 34.

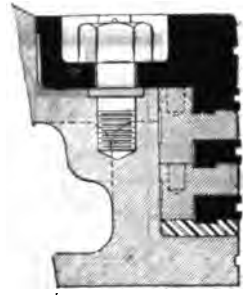


FIG. 35.

Types of Piston Packing.

FIG. 36.

FIG. 37

Types of Piston Packing.

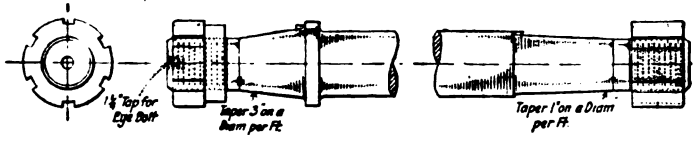


FIG. 38.—Piston Rod.

and is proportioned from the unbalanced load, P, for a stress of 3500 to 4000 lbs. Four threads per inch are generally used for all sizes. The diameter of tail rod, if fitted, is about 0.7 times the diam. of the piston rod, or as large as the threads of the piston rod nut will permit.

Crossheads. In merchant engines the **central block type** of crosshead with two pins is generally used, Fig. 39. The **depth of the central block** is 1.6 to 1.8 and the **length** 1.5 to 1.8 times the diam. of the piston rod.

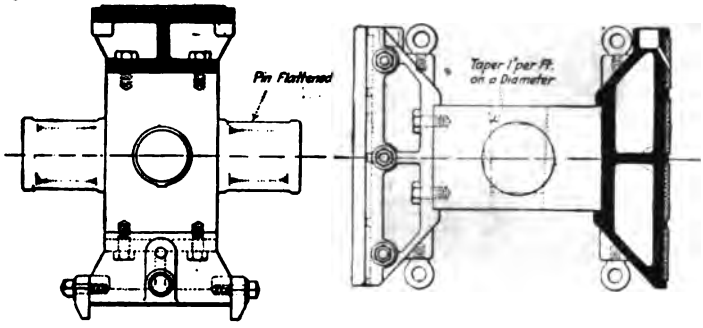


FIG. 39.—Central Block Type of Crosshead.

The **width**, when one slipper is used, is about the same as the depth; with two slippers it is made much wider to keep the slippers clear of the connecting rod. The central block should be figured for strength as a beam supported at the middle of the pins and loaded by the unbalanced load on the piston rod.

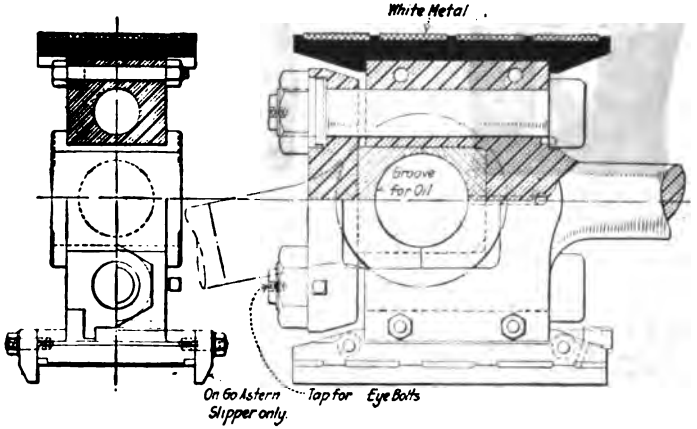


FIG. 40.—Single Pin Type of Crosshead.

P. Stress = $PL \div 4R$ where L = length between middle of pins and R = resistance of the section = $bd^2 \div 6$; b = width less the average width of the hole for the piston rod and d = depth. Stress may safely run as high as

3000 lbs. The **single pin type** of crosshead, which usually consists of a bearing box on the end of the piston rod, is also used, Fig. 40. In this type the pin is shrunk into the fork of the connecting rod. Stresses in the bolts and caps, due to the unbalanced load P, should be calculated as shown under Main Bearings, p. 867. The same stresses are allowable.

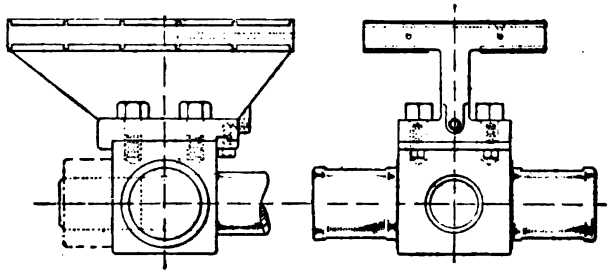


FIG. 41

Crosshead Pins. As the crosshead pin bearings are oscillating instead of rotating bearings, higher pressures per in. are allowable than on the crank shaft and crank pin bearings. The pressure is produced by the load on the

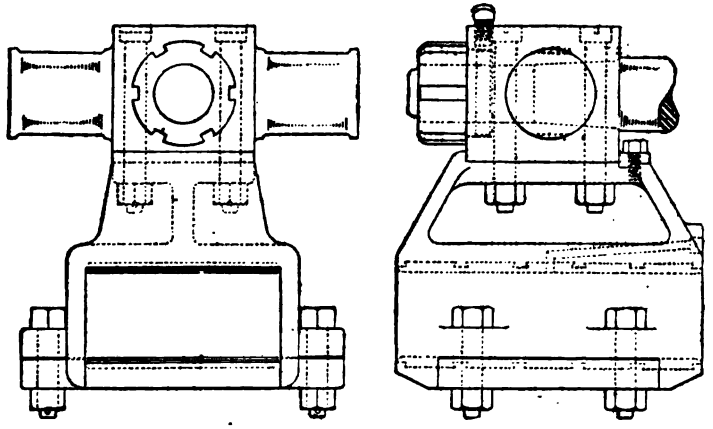


FIG. 42.

connecting rod P_1 (see Fig. 44), and pressures of 500 to 600 lbs. per sq. in. are allowable. The pins must be proportioned for strength as well as area. **The diameter of the pins** of the type shown in Fig. 39 is 1.1 to 1.2 times the diam. of the piston rod and the length 1.1 to 1.25 times their diam. The

pin is figured as a cantilever loaded at the middle of its length by one-half the load on the connecting rod, P_1. Stress $= P_1 l \div 2 \times 2R$ where l = length of pin and R = resistance to bending $= d^3 \div 10.2$. Crosshead pins are flattened on the sides to facilitate the distribution of oil from top to bottom.

In the single pin type, Fig. 40, diameter of the pin is 1.2 to 1.5 times the diam. of the piston rod and the length about 1.4 to 1.6 times its diam. The

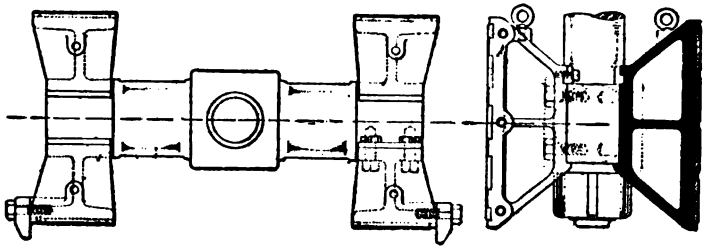


FIG. 43.

pin is figured as a beam fixed at the ends and loaded at the middle of its length by the load on the connecting rod, P_1. Stress $= P_1 l \div 8 \times R$, where l = length of pin and R = resistance to bending $= d^3 \div 10.2$. Stresses in pins of either type may safely run as high as 3500 lbs. per sq. in.

Crosshead Slippers. Crossheads with two slippers, Fig. 39, are generally used in **intermediate and large engines.** For **smaller engines** one slipper of the type shown in Fig. 41 or the box type, Fig. 42 are used. For **very large engines** with four housings to each cylinder four slippers are used, Fig. 43. Slippers are usually of cast iron but sometimes of cast steel and lined with white metal. When one slipper is used the bolts securing the slipper to the central block must be proportioned to carry the load when backing. When two or four slippers are used the ahead or astern slippers must be fitted with gibs on each side as shown in Figs. 39, 40 and 43 to keep the crosshead in alignment. To facilitate the removal of the slippers keys are fitted in the upper side of the central block as shown in Figs. 41, 42 and 43.

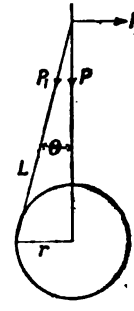


FIG. 44.

The **pressure of the slipper on the guide,** P_2, is the component of the load on the connecting rod, P_1, perpendicular to the guide as shown in Fig. 44; $P_2 = P_1 \sin \theta$ or $P \tan \theta$. The **surface of the slipper** should be proportioned for a pressure of about 40 lbs. per sq. in. When single slippers of the type shown in Fig. 41 are used the **backing surface is** 65 to 75 per cent of the ahead surface.

Connecting Rods. The usual type used in merchant engines is shown in Fig. 45. The crank end and crosshead end boxes and caps are lined with white metal and are usually of cast steel, although sometimes of brass. Caps are generally cast in one piece but cast iron or brass boxes and forged steel binders are sometimes used as shown in Fig. 46. Set screws are fitted in the crank pin cap and in the upper end of the connecting rod to keep the bolts from falling out when the nuts are removed enabling caps and bolts to

be handled together. For intermediate and large engines there should be two oil holes to the crank pin with pipes led to the oil boxes on top of the crosshead pin bearings. If the single pin type of crosshead is used the fork of the

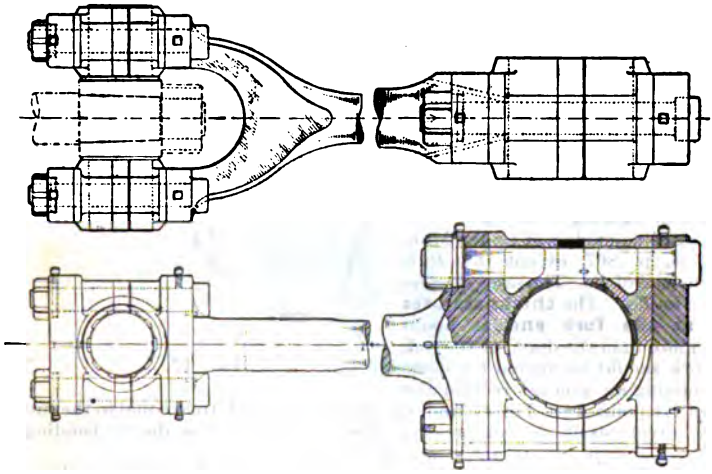


FIG. 46.

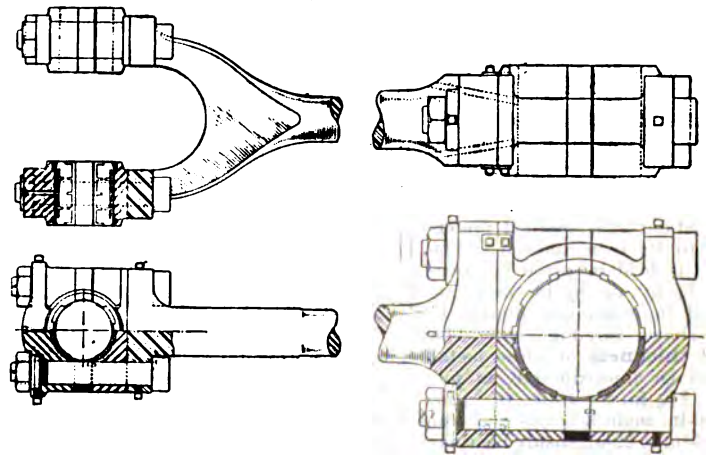


FIG. 45.

Connecting Rods.

connecting rod is made with solid eyes and the pin shrunk in. The **length between centres** of connecting rods for main engines, is usually 2.25 times the stroke, but this ratio is often made 2.5 for small engines and 2 for large

engines. The **diameter at the upper end** is 0.90 to 0.95 times the diam. of the piston rod and the **diameter at the lower end** about 1.25 times the diam. of the upper end. The rod must always be figured for strength as a column with hinged ends. This is done by the method explained under Piston Rods, p. 858, except that the load is taken as $P_1 = P$ sec. θ, where P is the unbalanced load and θ the angle whose sine is equal to the crank radius ÷ length of rod, Fig. 44. In figuring the factor of safety the stress and radius of gyration must be taken at the middle of the rod. The factor of safety is usually about 20. The method of figuring bolts and caps is the same as explained under Main Bearings, p. 867, except that P_1 is used instead of P. The same stresses are allowable. The **thickness over flats at the fork end** is about ½ in. more than the diam. of the rod. The fork should be figured for stress by assuming an average rectangular section at a point $A-A$, Fig. 47, and considering the total stress equal to the sum of the direct tensile or compressive stress and the stress due to bending.

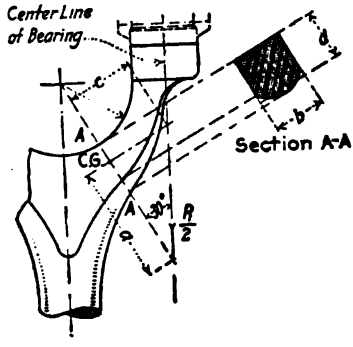


FIG. 47.

Total stress $= \dfrac{P_2 \times \sin 30°}{2b \times d} + \dfrac{6P_1 \times \cos 30° \times c}{2b \times d^2}$ which may be reduced to the form $\dfrac{P_1}{4bd^2}(d + 6a)$. The stress figured by this method may safely be 5000 to 6000 lbs.

Bearings

White metal is cast in the bearing boxes at least ¼ in. thicker than required by the finished bearing to allow for hammering in. After the bearing is bored out it is scraped preferably to its own journal. The metal is held in place by dovetails ¼ in. to ½ in. deep arranged in a manner similar to that shown in Fig. 48. The **total thickness** of white metal should be approximately as follows: 9/16 in. for 10-in. to ¾ in. for 20-in. main bearings and crank pins; ½-in. for 8-in. to 15-in. and ⅝ in. for 16-in. to 22-in. steady bearings; ⅜ in. for 5-in. to ⅝ in. for 10-in. crosshead pins; 9/16 in. to ¾ in. for thrust shoes and about ½ in. for crosshead slippers and other flat surfaces.

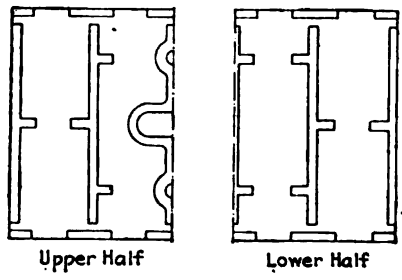


FIG. 48.—Dovetails Securing White Metal in Bearings.

Main Bearings. Two bearings to each cylinder or crank is usual although, in order to shorten the engine and save weight, one bearing is often used be-

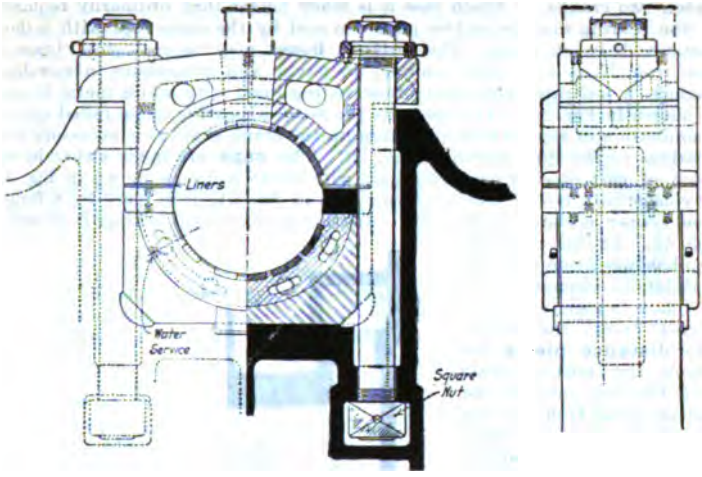


FIG. 49.

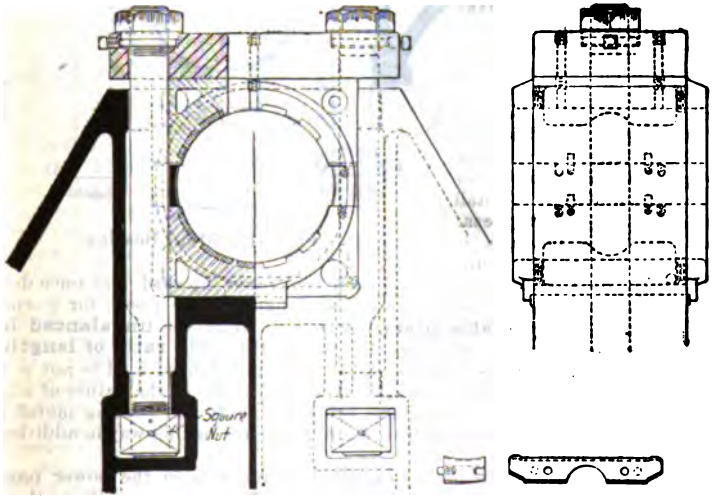


FIG. 50.
Types of Main Bearings.

tween two cranks, in which case it is made longer than ordinarily required, or the bearing may be in two parts covered by the same cap, with a short distance between them. The **bottom boxes** may be cast steel, brass or cast iron, lined with white metal. In naval, and sometimes in merchant engines, a separate cylindrical brass casting lined with white metal is used as shown in Fig. 49. This permits the bottom bearing to be rolled out for examination or repairs without taking up the crank shaft as is necessary with bearings of the type shown in Fig. 50. The **caps** are made either in one piece of cast steel or cast iron lined with white metal as shown in Fig. 49, or consist of a box of cast steel, cast iron or brass held in place by a forged steel keeper as shown in Fig. 50. Another type of main bearing is shown in Fig. 51. In this type the cap has lugs fitted over the bedplate to strengthen the bedplate against side thrust from the shaft. The **distance pieces** between the bottom brass and the cap prevent the bottom brass from nipping the shaft in case the bearing heats. Unless bearings are unusually long, two **bolts** are sufficient to secure the cap. The caps are fitted with **oil boxes** and a **handhole** with cover for the purpose of feeling the shaft and for applying cold water directly to the shaft in case the bearing warms up. The bottom boxes and sometimes the caps are cored for water circulation except in small engines. The **max. press.** on the main bearings is produced by the unbalanced load and occurs twice each rev., once upward and once downward and is usually taken as the basis of figuring the press. for purposes of design. The **allowable press.** per sq. in., from the **unbalanced load** taken by two bearings, run from 125 to 175 lbs. The **ratio of length to diam.** of main bearings is 1.0 to 1.15 The unbalanced load is not a true measure of the load on the bearings, but as it operates in the nature of a blow tending to destroy the oil film and bring the shaft and bearing metal into direct contact, it is always well to figure pressures in this way in addition to any other method used.

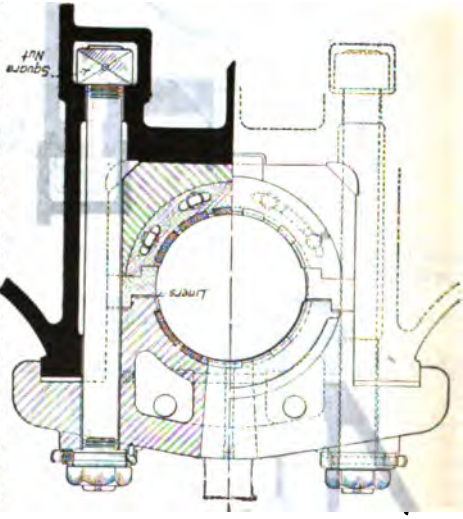


FIG. 51.—Main Bearing.

The true load on the main bearings is a function of the power passing through them and increases from forward to aft, as the power from the forward cylinders exerts a turning force on the after crank webs, which is transmitted to the bearings. Higher press. per sq. in. are allowable on the after bearings as the turning force transmitted through them is more uniform and, therefore is more evenly distributed around the bearing. In designing, however, it is usual to proportion the bearings on the average load and make all

bearings the same length except the extreme forward bearing, which having no power passing through it, frequently is made 10 per cent shorter than the intermediate bearings. For a similar reason it is desirable to make the extreme after bearing, through which all the power passes, 10 per cent longer than the intermediate bearings. The average load on each bearing, from the power is $\frac{I.H.P. \times 33{,}000}{P.S. \times \text{No. bearings}}$. The **allowable pressure** per sq. in. **from mean load** run from 80 to 120 lbs. In the bearing pressures given above consideration has not been given to the weight and inertia of the crank shaft and reciprocating parts, or the centrifugal force. The **pressure due to weight** amounts to an additional 40 to 60 lbs. per sq. in. but the effects of inertia and centrifugal force are small. Some designers, in determining allowable bearing pressures take into account the product of the pressure and the rubbing speed of the bearings but for merchant marine engines this is not considered necessary unless the engines are of a size and speed beyond the range of ordinary practice.

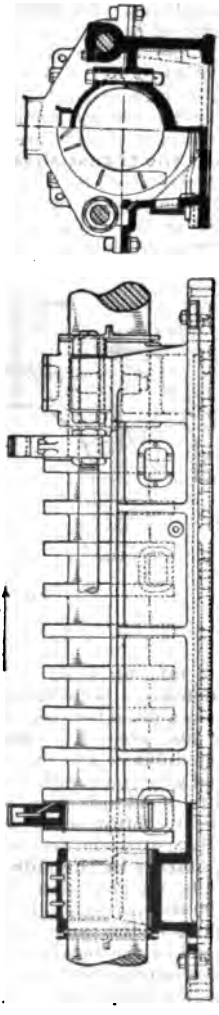


FIG. 52.—Thrust Bearing. Horseshoe Type.

Main bearing caps and bolts must be proportioned for the max. unbalanced load, P. The tensile **stress in the bolts** $= P + (2n \times a)$ where n = No. of bolts in one bearing and a = area at bottom of threads. The stress in the bolts, on account of the intermittent and suddenly applied load should not exceed 3500 lbs. The **cap** is figured as a beam loaded in the middle and supported at the ends. **Stress** $= Pl + (8 \times 2R)$ for caps cast in a single piece, where l = dist. between bolts, and R = section modulus which is determined by the method given on p. 838. The section modulus is always figured under the white metal and the handhole and oil hole taken out. If the cap consists of a box with a keeper, it is considered that the separate box has the effect of making the load on the keeper a mean between concentrated and distributed load and the load is assumed as all being taken by the keeper and stress $= Pl + (10 \times 2R)$. On account of the intermittent and suddenly applied load and to secure stiffness the stresses in caps should not exceed 2500 lbs. per sq. in. for cast steel, 1500 for cast iron, and 3500 for a forged steel keeper.

Thrust Bearings. The **horseshoe type** of thrust bearing, Fig. 52, is generally used with reciprocating engines as the nuts on the side rods provide means of adjustment. The **number of shoes and their area** is determined

from the indicated thrust T, which $= \frac{I.H.P. \times 33{,}000}{R.P.M. \times P \times \%}$ or $\frac{I.H.P. \times 33{,}000}{S \times 101.3}$ where P = pitch of propeller in feet, % = per cent slip and S = speed of the vessel in knots. For **diam. of thrust collars** see "thrust shaft," p. 874. In this type of bearing the effective area of the thrust shoe is only about 70 per cent of the annular area of the thrust collar. The **press. on the shoes** is usually 40 to 60 lbs. per sq. in., but with very stiff thrust collars and shoes press. as high as 65 lbs. may be carried safely. The thrust shoes, Fig. 53, are cored for water circulation and are usually of cast iron although sometimes of cast steel and are lined with white metal on both faces. The **thickness of the thrust shoe**, including the babbitt, should be 2.3 to 2.6 times the thickness of the thrust collar for cast iron and 1.4 to 1.6 for cast steel. Sometimes instead of the customary dovetailed babbitted surface, a loose babbitt or bronze face is secured to the "go astern" side of the thrust shoe by screws as shown in Fig. 54. Liners may then be inserted behind the

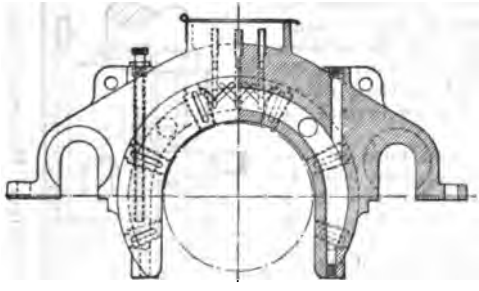


FIG. 53.

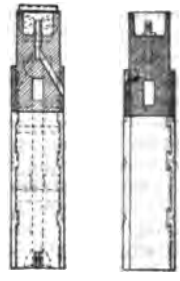


FIG. 54.

Thrust Shoes.

face to take up wear. In calculating the **stress in the thrust shoe** it is considered as a beam supported at the centres of the side rods, the load being taken as a mean between concentrated and distributed. Bending moment $= \frac{TL}{10n}$, where L = dist. between side rods in in. and n = No. of shoes. The modulus of the section at the centre is found in accordance with the method given on p. 838. Stress = bending moment ÷ section modulus. In order to avoid deflection, which would cause unequal bearing on the surface, stresses in thrust shoes should be kept low, good practice being not over 1500 lbs. per sq. in. for cast iron and not over 3000 lbs. for cast steel. The **tensile stress in the side rods** is calculated by dividing the total indicated thrust by the area of the rods at the bottom of the thread. A stress of 5000 to 6000 lbs. per sq. in. is allowable for either steel or bronze rods. **Steady bearings** of the same design as the line shaft steady bearings are fitted at each end of the thrust bearing bedplate as shown in Fig. 52. The length of these bearings is 1.0 to 1.1 × diam. of shaft. Sometimes the cap is lined with white metal but this is unnecessary. A stuffing box is fitted around the lower half of the shaft to retain the oil in the thrust bearing bedplate. A pipe coil for water circulation for cooling the oil is fitted in the bottom of the thrust bearing bedplate.

The all-round type of thrust bearing is occasionally used. To permit of adjustment, the "go ahead" and "go astern" bearings are separate, although both are fitted on a common bedplate. In this type the bearing surface covers the entire annulus of the collar, and as the collars on the shaft, due to their form, are less liable to deflect they may be made of larger diameter, necessitating the use of fewer shoes to maintain the same pressure per square inch. These bearings, however, lack the adjustability of the horseshoe type and are now seldom used in marine work.

Sole Plate. The thrust bearing is bolted down, through elliptical holes, on a cast iron sole plate which is riveted to the hull foundation. Adjusting wedges are fitted at each end between the bearing flanges and projections on the sole plate. It is very important that the foundation under the sole plate be strong and rigid to avoid deflection. Sometimes to insure rigidity it is preferred to cast the sole plate with V-shaped legs on the forward end for bolting to the main engine bedplate. For merchant vessels of low power it is satisfactory to omit the sole plate and to bolt the thrust bearing to the hull foundation on liners through which the bolts pass, chocks being riveted forward and aft of the bearing and liners driven up hard against the bearing flanges to lock the whole solid after bearing is adjusted. Low **stresses** (not over 4000 lbs. per sq. in. in single shear) should be allowed in the rivets securing the sole plate and in the holding down bolts securing the bearing.

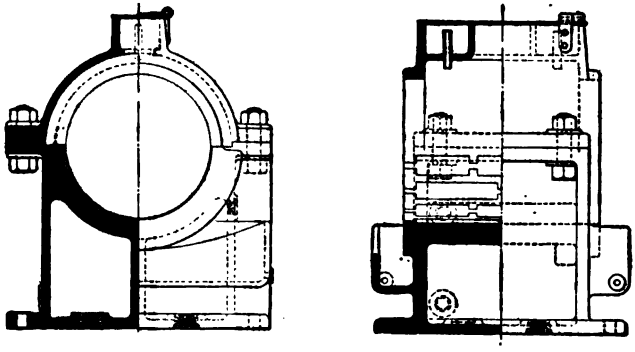


FIG. 55.—Line Shaft Steady Bearing.

Line Shaft Steady or Spring Bearings. It is customary to make steady bearings the same diam. as the shaft, it being unnecessary to increase the shaft in the bearing even if it is desired to leave the rest of the shaft rough turned; only one bearing is generally used to each section of shaft, but with large shafts a bearing is sometimes fitted at each end of each shaft to facilitate lining up and handling. The **lengths of steady bearings** vary from 1.25 to 1.5 diameters and the **pressures per sq. in. of projected area** due to the weight of the shaft run from 35 to 50 lbs. If the distance between bearings is too great the shaft may deflect due to its weight and cause heating of the bearings. With only one bearing per shaft a safe **length between centers of bearings** in ft. $= c\sqrt{d}$ in which d = diam. of shaft in in. and $c = 5$ to 5.5. Fig. 55 shows a typical steady bearing. The **pedestal** is of cast iron

pined with white metal, and provided with an oil receptacle. The **cap** which is usually merely a shell almost touching the shaft at each end to keep out dirt, has wick feed **oil boxes** and a **handhole** for feeling the shaft. The bearings are sometimes piped for water circulation. Caps are secured by four **bolts**, the diam. of which vary from 1 in. for an 8-in. bearing to 1½ in. for an 18-in. bearing. For the same size bearings the **holding down bolts** vary from 1 in. to 1¼ in. diam. pitched about 6 diam. Ring oiling is sometimes used.

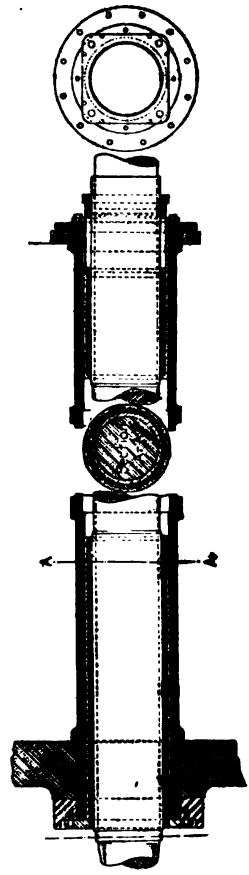


Fig. 56.—Stern Tube.

Stern tubes in merchant work are of cast iron and in one piece unless very long, Fig. 56. The **thickness of the casting** in the stern frame and between bearings varies from 1½ in. for a 10-in. shaft to 2½ in. for a 20-in. shaft. The **min. length of the after bearing** is 4 × diam. of propeller shaft under the bearing sleeve according to Lloyd's Rules, but this sometimes is made longer. The **length of the forward bearing** depends on the length of the stern tube, but the min. length is 1¼ × diam. of shaft. The **pressure due to weight** of the shaft, sleeve and propeller should not exceed 30 lbs. per sq. in. for the after bearing and 20 lbs. for the forward bearing. In all cases a steady bearing is fitted on the line shaft as close as possible to the inboard coupling. The stern-tube bearings are usually of lignum vitæ contained in brass bushings, made in halves, the joints being in the horizontal plane slightly inclined fore and aft to facilitate withdrawal. The **thickness of the wood**, which always bears on the end of the grain, runs from 1 in. for a 10-in. shaft to 1¼ in. for a 20-in. shaft. The **thickness of the brass bushing** under the wood for the same size shafts is about ¼ in. and ⅝ in. The stern tube is held in place in the stern frame by a cast or forged steel nut in which there are holes or slots for a spanner wrench for tightening up. Four threads per in. are used for nuts of all sizes. The thickness of the nut should be about 3¼ in. for a 10-in. shaft and 6½ in. for a 20-in. shaft. At the forward end of the stern tube is fitted a **stuffing box** of the usual type for soft packing. Very often a bronze or cast-iron bushing is substituted for the forward lignum vitæ bearing.

No **lubricant** other than water is required for stern tubes with lignum vitæ bearings. There should be a pipe connection to one of the salt water pressure pumps in the engine room for washing out the stern tube and also a connection for draining the space between the forward and after bearings. Sometimes the after bearing is babbitted and the forward bearing bronze bushed in which case a protecting stuffing box must be fitted between the propeller and the after end of the stern tube to keep out sand and dirt. With these stern tubes oil is used for lubrication. There are several patented stern tubes and protective stuffing boxes on the market.

Strut Bearings. In twin screw vessels in which the stern tube is not carried back to the propeller a separate bearing is carried in the strut for taking the weight of the shaft and propeller. The length of the bearing, the thicknesses of the wood and the brass bushing and the pressure per sq. in. correspond to the figures given for the after stern-tube bearing.

Lignum Vitæ is a greenish-brown, hard, heavy wood of specific gravity, 1.10. It is used for stern tube and strut bearings, and makes an excellent low pressure bearing when **lubricated by** the free circulation of water. Lignum Vitæ **for use in bearings** should be generally straight, sound and well seasoned; it **should be free from** injurious shakes, wormholes, excessive sap, large or unsound knots, and other injurious defects. Ordinary season checks, small amounts of sap, slight heart and ring shakes are not generally considered defects.

Bulkhead Stuffing Boxes of cast iron are fitted where line shafting passes through a watertight bulkhead, Fig. 57. The box and gland are in halves. The stuffing box is carried on a portable plate in halves large enough to ship the shafts and bearings. Soft **packing** is used.

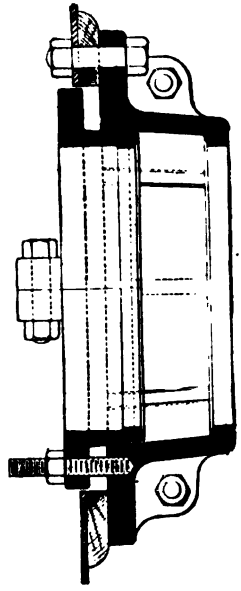


FIG. 57.—Bulkhead Stuffing Box.

Shafting

Arrangement. The location of the **after end** of the shafting is determined by the diam. and the depth of immersion of the propeller. In single screw vessels a clearance of 4 in. to 6 in. must be allowed between the tips of the propeller and the stern frame shoe. In twin screw vessels ample clearance between the tips of the propellers and the hull must be allowed. (See p. 1033.) The height of the **forward end** of the shafting is determined by the depth necessary for a strong and rigid foundation under the main engine bedplate. Shafting is kept parallel with the base line of the vessel except where the inner bottom or the water tanks under the engine makes this impossible. In twin screw vessels it is often impossible due to the arrangement of the machinery, to keep the shafting parallel in the horizontal plane.

The **shaft tunnel** is fitted its entire length with a platform from which the bearings may be attended. The forward end of the shaft tunnel is isolated from the engine room by a sliding watertight door operated from above. Overhead in the tunnel are located eye-bolts or other lifting gear for handling the propeller and thrust shafts and the stern-tube bearings. The propeller shaft is drawn inboard for examination unless its length is prohibitive. A water service pipe is carried through the tunnel with a connection to each bearing and to the stern-tube stuffing box. This pipe is usually led so it can be used as a hand rail. Fig. 58 shows the arrangement of shafting for a single-screw vessel with the machinery located in the stern.

Crank shafts for merchant engines are usually of the "built up" type, that is, the pieces of shaft, webs and pins are forged separately and shrunk together. For Naval engines, yachts and other engines where saving of

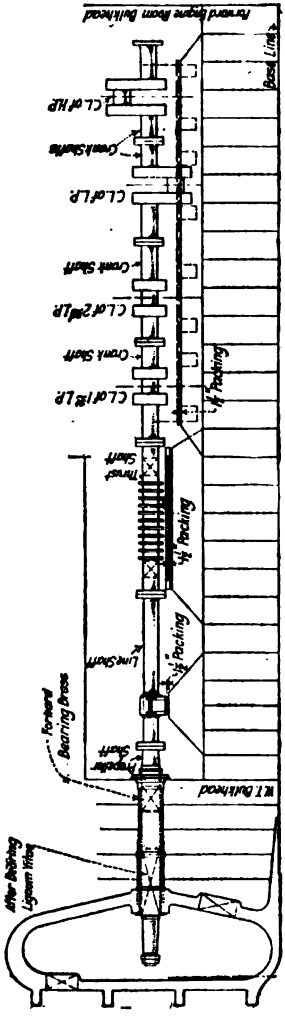


FIG. 58.—Arrangement of Shafting.

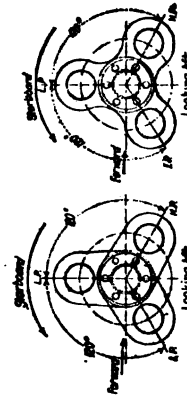


FIG. 60.

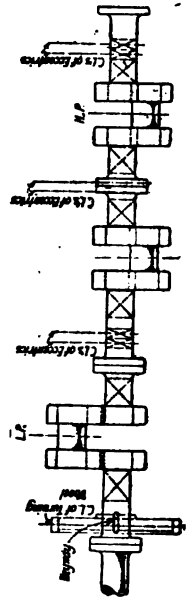


FIG. 59.
Crank Shaft.

weight and space is important, crank shafts are "solid forged" and usually hollow. In order that one spare section of crank shaft may serve for any crank, the sections are often made interchangeable and sometimes interchangeable and reversible, that is, one spare section will serve for any crank with either end forward. The overall dimensions of cylinders and bedplate are increased if the sections of crank shaft are made interchangeable, or interchangeable and reversible. This question is discussed under Drafting Room Method of Laying Down a Marine Engine, p. 918.

The **diameter** of the crank shaft for a merchant engine is determined from the rules of the Classification Society under which the vessel is constructed, but the designer should always calculate the stress in the after end of the shaft produced by the combined bending and twisting moment. Max. twisting moment in in.-lbs., $T = \frac{63{,}000 \times I.H.P.}{R.P.M.} \times C$, where C = ratio of the max. to the mean twisting moment. This ratio varies with the number of cranks, as shown in Table 14. The bending moment is produced by the twisting moment, which, for all practical purposes, may be taken the same as T, considered at the middle of the crank pin. Then max. bending moment, $B = T \times L + 8 \times r$, where L = dist. between edges of bearings of after crank in in. + 1 in. and r = rad. of crank in in. Then Equivalent Max. Twisting Moment, $T_e = B + B\sqrt{1 + (T + B)^2}$.

Stress in shaft = $T_e + R_t$, where R_t = resistance of the shaft to twisting = $D^3 + 5.1$ for solid shafts and $(D^4 - d^4) + 5.1D$ for hollow shafts, in which D = diam. of shaft and d = diam. of hole. Stress should not exceed 6000 lbs. per sq. in., although it is often less than this due to the desire of owners to have shafts larger than required by the Classification Society Rules.

Table 14. Ratio of Maximum to Mean Twisting Moment

Type of engine	Ratio for avg. cut-off of 70 per cent in H.P. cyl.
One-crank	2.00
Two-crank	1.50
Three-crank	1.33
Four-crank	1.25

The effect of inertia is not considered in the above ratios. According to Bragg, inertia may increase the ratio for three-crank engines as much as 18 per cent, while for engines with cranks at 180 deg. and 90 deg., the inertia effects are nearly balanced.

Crank webs for merchant engines are usually of the type shown in Fig. 59, with straight sides, although they are sometimes made as shown in Fig. 60 to save weight; the weight saved, however, does not justify the extra labor involved. Crank webs are subjected to a bending stress in the direction of rotation due to the max. twisting moment and also to a bending stress in a fore and aft direction due to the unbalanced load on the crank pin. These stresses, however, are small and need not be computed, if the webs are proportioned by Table 15.

In built-up shafts the shafts and crank pins are shrunk into the webs. Square or round keys are always fitted in the shafts and sometimes in the pins.

Table 15. Proportions of Crank Webs for Built-up Shafts

Diameter bore for crank shaft	Diam. shaft + 1½ in.
Diameter bore for crank pin	Diam. pin + 1 in.
Metal around crank shaft	0.5 × diam. shaft
Metal around crank pin	0.4 × diam. pin
Thickness of webs if sections of shaft are interchangeable	0.68 × diam. shaft
Thickness of webs if sections of shaft not interchangeable	Forward cylinder = 0.52 × diam. shaft After cylinder = 0.73 × diam. shaft.

Crank Pins. It is usual to make the crank pin the same **diam.** as the crank shaft, although it is sometimes made larger to reduce the pressure, if it is desired to shorten the pin, or to reduce the stress, if a long pin is required to keep the pressure low. **Ratios of length to diam.** of pin vary in good practice from 0.95 to 1.0 for triple and quadruple expansion engines and from 1.0 to 1.1 for compound engines. The **press. on the crank pin** is figured from the thrust of the connecting rod, P_1, produced by the unbalanced press. on the piston. (See connecting rods, p. 862). In computing the press. the actual length of bearing surface of the connecting rod is used which is about ¼ in. less than the length of the pin. **Allowable pressures** per sq. in. are 300 to 325 lbs. for triple and quadruple expansion engines and 400 to 425 lbs. for compound engines. Some designers give consideration to the heat produced by the friction of the crank pin in proportioning its size, but unless the design is out of the range of ordinary practice this is unnecessary.

The **after crank pin**, through which the power from the forward cylinders all passes, should always be figured for stress. This pin is subjected to a bending moment produced by the torque from the forward cylinders applied at the forward end, and also to a bending moment from the torque produced by the after cylinder, applied at the middle of its length. These bending moments are combined and the resulting bending moment used in figuring the stress as follows:

Force from forward cylinders applied at forward end of pin,

$$F_1 = \frac{\text{I.H.P. of Fd. cyls.} \times 33{,}000 \times 12}{2 \times \pi \times r \times \text{R.P.M.}} \times C$$

Force from after cylinder applied at middle of pin,

$$F_2 = \frac{\text{I.H.P. of aft. cyl.} \times 33{,}000 \times 12}{2 \times S \times \text{R.P.M.}} \times 1.1$$

in which r = rad. of crank in in., S = stroke in in., and C = ratio of max. to mean twisting moment as given in Table 14. Let L = the length of crank pin in in., then max. bending moment on after pin, $B_m = F_1 \times L + F_2 \times \frac{L}{2}$ **Stress in pin** = $B_m + R_b$ where R_b = resistance of the pin to bending = $D^3 \div 10.2$ for solid pins and $(D^4 - d^4) \div 10.2D$ for hollow pins in which D = diam. of pin and d = diam. of hole. Stress in the crank pin computed by this method shou.d not exceed 5000 lbs. per sq. in. for merchant engines.

Thrust Shafts. The diam: is determined by the Classification Society rules, but it should be figured for stress due to combined bending and twisting, the bending moment being produced by its own weight between centres of the end bearings. The max. twisting moment, T, will correspond to the value given under Crank Shafts. Max. bending moment = $(W \times L) \div 8$, where W is the weight and L = length of shaft between centres of bearings in in. The equivalent twisting moment and stress may then be found, as shown under crank shaft, p. 873. Stresses of 5000 to 6000 lbs. per sq. in. are allow-

able in merchant work. For **No. of thrust collars** see "thrust bearings," p. 868. In order to be stiff and not deflect the **diameter of the thrust collars** should not exceed 1.45 to 1.65 times the diam. of the shaft. With collars of these proportions pressures per sq. in. of 65 lbs. for the smaller ratio to 40 lbs. for the larger ratio may be carried safely. For the same reason the **thickness of collars** should not be less than 0.6 to 0.5 times the height. To provide sufficient space for stiff thrust shoes the **space between collars** should be 2.3 to 2.6 times the thickness of the collars for cast iron and 1.4 to 1.6 for cast steel shoes.

Line Shafts. **The diam.** is determined by the Classification Society rules but it should be figured for stress by the method given under "thrust shaft." If the stress exceeds 5000 lbs. per sq. in. for small or 6000 lbs. for large shafts the length between bearings should be reduced. In any event the **overall unsupported length of line shafts** should not exceed 20 ft. to avoid deflection.

Propeller Shafts. The **diam.** is determined by the Classification Society rules but the stresses due to combined twisting and bending should always be calculated. The max. twisting moment is the same as given under crank shafts and the bending moment is produced by the weight of the shaft. The bending moment = $(2W \times L) + 8$, in which W = weight of shaft between bearings, including brass sleeves, if fitted, and L = length in in. between centres of bearings. The factor 2, according to Seaton, is added for rough weather. Stress = equivalent twisting moment ÷ resistance to twisting. The stress in the after or overhung end of the shaft due to combined twisting and bending should also be computed. The bending moment = $W_1 \times L_1$, where W_1 = weight of propeller and shaft aft of the bearing, and L_1 = dist. in in. from centre of propeller to 3 in. inside of bearing. Stress = equivalent twisting moment ÷ resistance to twisting. (The brass sleeve should not be included in figuring the resistance.) Stresses in propeller shafts between bearings or in the overhung end should not exceed 5000 lbs. per sq. in. for small and 6000 lbs. for large shafts. Propeller shafts are fitted with brass sleeves in the stern-tube or strut bearings, the thickness of which should not be less than ¾ in. for all sizes of shafts up to 18 in., and 1 in. for shafts above 18 in. diam.

FIG. 61.—Propeller Shaft.

It is usual to cover the shaft between bearings with **brass sleeves**, the **thickness** of which may be about ⅛ that of the bearing sleeves. When brass sleeves are made in several sections the joints between the sections are scarfed leaving about ⅛ in. open space for burning in or calking full of soft copper as shown in Fig. 61. In twin screw vessels where struts are used and the propeller shafts are accessible, the shafts between bearing sleeves are often protected by wrapping with marlin or painting with anticorrosive paint.

For proposed standard **propeller hubs, nuts** and **keys** see pp. 1066 to 1076.

Flanged Shaft Couplings. The diameter of the coupling flanges on crank, thrust, line and propeller shafts is about 1.8 to 2 times diam. of shaft and the thickness about 0.23 to 0.28 × diam. of shaft but the couplings **should be proportioned from the size of bolts required.** The **bolts** must be of a size to safely resist the shear produced by the maximum twisting moment, T. **Stress** = **T** ÷ **n** × **a** × **r** in which n = number of bolts, a = area of each bolt at face of coupling and r = radius of pitch circle. Allowable stress is 5000 to 6000 lb. **Six bolts are used unless** a greater number is required to avoid excessive size. Coupling bolts are tapered as shown in Fig. 62 except in interchangeable crank shafts in which case they must be straight as shown in Fig.63. If diameter of bolt at face of coupling = A, then **proportions of couplings and bolts** when six bolts are used will be as follows: Thickness of coupling, B = 1.2A; distance from outside of shaft to pitch circle, D = 0.9A; distance to outside of coupling, C = from 2 1/8 in. for a 2 in. bolt to 3 3/8 in. for a 3 1/2 in. bolt; diam. of thread, F = 0.75A; depth of nut, G = 0.75A; depth of head for straight bolts, M = 0.7 to 0.65 A; size of head across corners, N = 1.38A; taper for tapered bolts = 3/4 in. per ft. on diam.

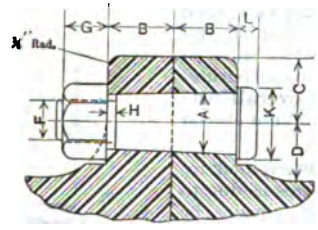


FIG. 62.

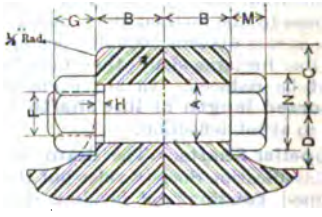


FIG. 63.

Proportions of Shaft Couplings and Bolts.

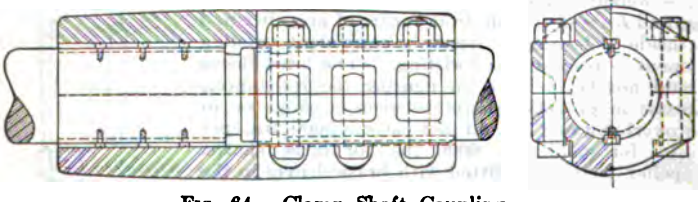


FIG. 64.—Clamp Shaft Coupling.

Clamp Shaft Couplings. When the propeller shaft is too long to be withdrawn inboard or if it is desired for other reasons to withdraw it outboard a removable coupling of the clamp type of cast or forged steel as shown in Fig. 64 is used. The two fore and aft keys in each shaft transmit the torque and should be figured for shear from the maximum twisting moment as shown under Propeller Shafts, p. 875. The diam. of coupling is from 1.7 to 2.0 times the diam. of shaft. The length must be sufficient to keep the key stresses within the allowable limit and is usually from 3 to 4 times the diam. of shaft.

The bolts should be as close to the shaft as possible and must be large enough to cause the coupling halves to grip the shaft tightly. The bolts are sometimes heated through the openings in the sides of the coupling while tightening up, in which case there will be a stress due to contraction in addition to the initial tension. The following formula adapted from Bauer gives the total area of all bolts on both sides at bottom of threads: Area $= C \times l \times (D - d) \div 2$, in which D, d, and l = outside diam., bore and length of coupling respectively and C is a constant = 0.25 to 0.35. When going astern the propeller thrust is carried by projections on the inside of the coupling which fit into grooves turned in the shafts. These projections and the metal between the grooves and the ends of the shafts should be figured for shear and crushing due to the astern propeller thrust. Astern Thrust in pounds = $\frac{\text{Astern I.H.P.} \times 33{,}000}{V \times 101.3}$ where V = astern speed of the ship in knots. The astern I.H.P. may be taken as 80 per cent of the ahead I.H.P. and V as 60 per cent of the ahead full speed.

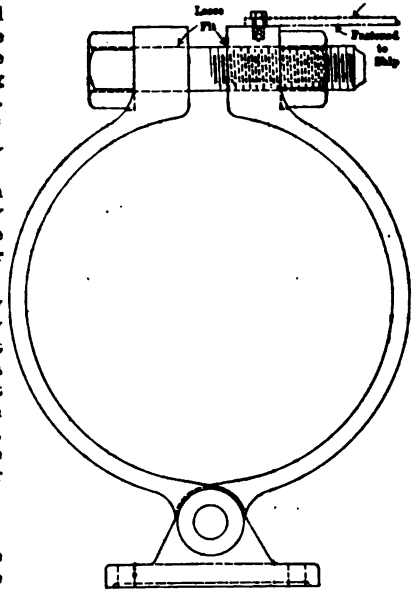


FIG. 65.—Shaft Brake.

Shaft Brake. In twin screw and sometimes in single screw vessels a brake similar to that shown in Fig. 65 is fitted to a shaft coupling aft of the thrust bearing to keep the shaft from turning while making repairs, or, in twin screw vessels, to keep one shaft from turning while under way with the other engine.

VALVE GEAR

The two general classes of valve gears which permit of reversal are (1) link motion with eccentrics and (2) radial gears with or without eccentrics. In America **Stephenson Link Motion** with two eccentrics is the most common type of valve gear used but **radial gears** of the **Marshall type**, with one eccentric and the **Joy type**, worked by links from the connecting rod, are frequently used. With Stephenson link motion the valves are in the fore and aft centreline of the cylinders although sometimes the H.P. valve is placed on the side of the cylinder and the valve motion transmitted from the links through a rocker arm. With the Marshall and Joy types the valves are on the side of the cylinders and hence the fore and aft length of the engine is less. A valve gear to be perfect would have a motion that would instantly open the valve wide at the beginning of the stroke and instantly close it at the point of cut-off without wire drawing; it would also instantly open it wide to exhaust at the end of the stroke and instantly close it when exhaust is completed. To more nearly attain this motion, **double ported and trick**

valves are often introduced. This motion is more nearly approached by radial valve gears than by link motion. With radial valve gears the lead is constant or nearly constant for all cut-offs; this is considered a very desirable feature by some designers.

Stephenson Link Motion. In this type of valve gear there are two eccentrics keyed to the crank shaft each operating an eccentric rod connected to a common link as shown in Fig. 106. The positions of the eccentrics are such that one eccentric rod operates the valve for going ahead and the other for going astern, the common link acting as the means for changing from one to the other. The valve stem is connected to the link block which slides on the link and changes the positions of the eccentric rods relative to the valve stem. When the block is in mid position there is practically no motion imparted to the valve and the engine stops. When it is in extreme position on the "**go ahead**" eccentric rod the link is in "**full gear**" and "**linked up**" when the link block is in any position between extreme and mid positions. Reversal is accomplished by moving the links by the suspension rods so that the sliding block is changed from the "go ahead" to the "**go astern**" eccentric rod which reverses the motion of the valve. Each cylinder may be **linked up independently** by means of the screw in the slotted reverse arm thus moving the links to bring the sliding block into the desired intermediate position. When linked up both eccentrics act on the valve and the motion is equivalent to that of an eccentric with less eccentricity and greater angle of advance resulting in shorter valve travel and a change in lead, cut-off, release, and compression. The change in these functions is shown by the valve diagram. The go astern eccentric is set at the same angle relative to the crank as the go ahead eccentric but on the opposite side of the crank.

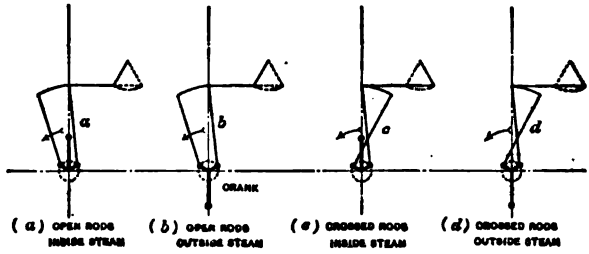


FIG. 66.—Open and Crossed Eccentric Rods.

Open and Crossed Rods. With the eccentricity toward the links the eccentric rods are "**open**" when arranged as shown at "*a*" and "*b*", and "**crossed**" when arranged as shown at "*c*" and "*d*", Fig. 66. In full gear the valve motion is the same for open as for crossed rods but when the gear is linked up the effect on the valve functions is materially different. The curves in Fig. 67 show that with open rods a greater range of cut-off may be obtained with a smaller reduction of port opening than with crossed rods. As the gear with open rods is linked up the lead increases while with crossed rods it decreases and may become negative. Linking up with crossed rods decreases the valve travel very rapidly with consequent reduction of both steam and exhaust port openings and therefore crossed rods are not as well adapted for expansive working as open rods. With crossed rods the links are thrown to-

ward the back of the engine when in ahead gear which leaves the front of the engine open for oiling and inspection. Also the suspension rods are much longer which reduces the slotting action of the link block. Most marine engines are fitted with open rods.

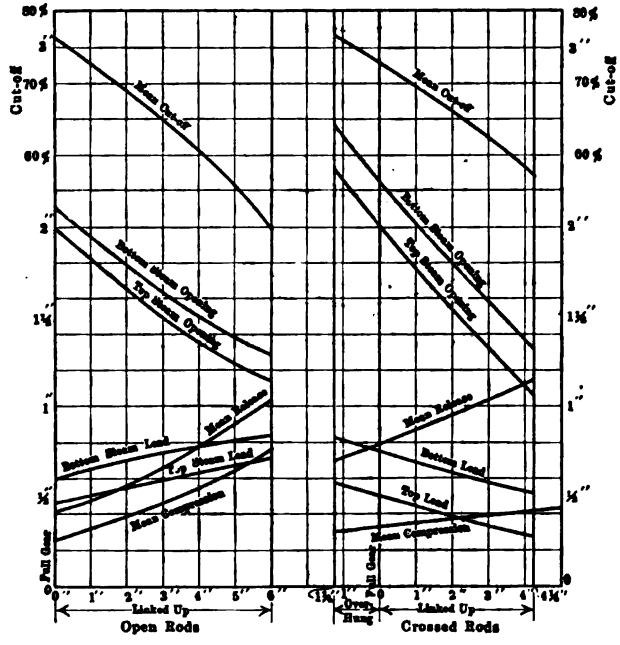


FIG. 67.—Changes in Valve Functions for Open and Crossed Rods.

Slip of the Link Block. In laying out a valve gear the slip or slotting action of the link block on the links should be given consideration. While this slip has no practical effect on the action of the valve, the length of the suspension rods and the point of suspension should be chosen to reduce this action as much as possible on account of wear. The nearer the link block to the point of suspension the smaller will be the slip. If the links are suspended directly over the position of the link block when in full gear ahead the slip may be considerable when the engine is reversed but this is of little importance if it is a minimum for the usual working position of the links. When the probable location of the reverse arm has been determined, the path of the point of suspension for one complete revolution should be plotted for full gear ahead and astern and maximum linked up position ahead and astern and clearances carefully checked. By examination of these loops, examples of which are shown in Fig. 68, it may be possible to choose a position of the reverse arm which will reduce the slotting action.

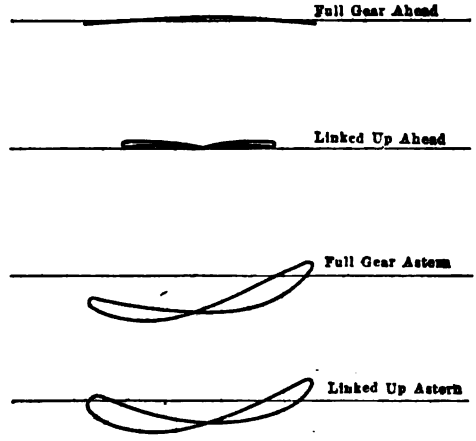


FIG. 68.—Slip of the Link Block.

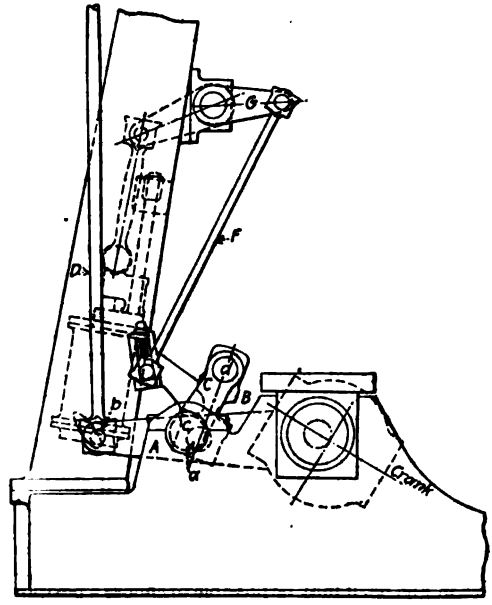


FIG. 69.—Marshall Radial Valve Gear.

Marshall Radial Valve Gear. This type of marine engine valve gear as usually constructed in America has the valve rod connected to the eccentric rod outside of the point of attachment of the guide link while European practice is to make the connection between the guide link and the eccentric. The arrangement of this gear for each cylinder consists of one eccentric mounted on the crank shaft, giving motion to a short and massive eccentric rod *A*, Fig. 69, which is guided at *a* by the link *B* and is connected at *b* to the valve rod *D*. The guide link *B* is supported at *d* by the bell crank lever *C* having its axis at *c* and connected by the rod *F* to the reverse arm *G*. The valve rod *D* may be connected to the valve stem direct or through a rocker arm. The gear is linked up by shortening the valve travel which is done by means of the adjustment block in the bell crank lever *C*. The movement of the point *b* is an ellipse as shown inclined to the horizontal in one direction for ahead motion and in the opposite direction for astern motion, the change being effected by the reversing engine through the reversing rod *F*. The angle between the eccentric and the crank is either 0 deg. or 180 deg. according to whether the valve is driven direct or through a rocker arm or whether it takes steam inside or outside. Lead is constant and a sharp cut-off and good steam distribution are obtained but due to the curvature at the path of the point *a*, the top and bottom cut-offs are unequal. The large number of joints subject to wear result in excessive lost motion and frequent adjustments must be made. The **oval valve diagram** is used for this type of gear and it is usually worked out by means of a simple model.

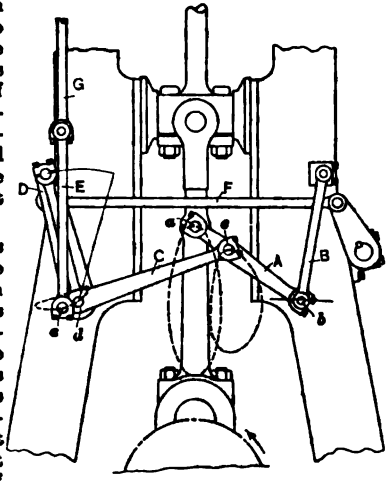


FIG. 70.—Joy Radial Valve Gear.

Joy Radial Valve Gear. In this type of valve gear no eccentrics are used, the movement being taken from the connecting rod by a series of links. By utilizing the reciprocating and vibrating actions of the rod a movement results which gives almost equal cut-off for both sides of the piston for all points of expansion, less compression at short cut-off, and much quicker action at the points of admission and cut-off than is given by Stephenson link motion. Lead is constant. The vibrating link *A*, Fig. 70, is connected to a point on the connecting rod and to the radius rod *B*. The movement of the point *a* is an ellipse and of the point *b* an arc of a circle as shown. Another lever *C* with a fulcrum at *d* is connected at *c* to a point on the vibrating lever and at *e* to the valve link *E* which actuates the valve stem *G*. The fulcrum *d* is connected to the reverse lever *D* or to a link block sliding in a slotted link. The movement of the point *e* is an ellipse, the inclination of which from the horizontal, as effected by the position of the reverse lever *D*, or slotted link, governs the degree of expansion. The engine is reversed by inclining the axis of this ellipse the other way by means of the reverse lever through the link *F*. For this type

of valve gear the **oval valve diagram** is used and it is best worked out by means of a simple model. The defects of the Joy Gear are the large number of joints subject to wear, resulting in excessive lost motion, and the difficulty of inspecting and taking care of the crosshead and crank pin due to the obstruction of the moving parts while the engine is running.

Action of the Valve. The successful operation of the engine depends upon the proper design and setting of the valves. The action of the valve is the same whether piston or flat slide valves are used, and the elementary principles apply to both. In mid position, the valve not only covers the steam ports, but overlaps the edges of the ports, as shown in Fig. 71. The amount of overlap *a*, on the steam side is the **"steam lap"** and on the exhaust side *b*, the **"exhaust lap."** The exhaust lap *b*, is **"positive"** if overlapping the edge of the port and **"negative"** when not covering the port. The other elements are *c*, **steam port width**; *d*, **maximum steam port opening**; *e*, **exhaust port width**; *f*, **minimum exhaust port opening**; and *g*, **half travel.** The complete cycle of distribution of steam in the cylinder consists of admission, expansion, release and compression. Admission begins through *c*, when the outer edge of the valve has travelled a distance equal to the steam lap *a* and uncovered the edge of the steam port (**point of admission**), and ends when the valve has reached this point again, travelling in the opposite direction (**point of cut-off**). Expansion then takes place until the inner edge of the valve has uncovered the edge of the steam port to exhaust (**point of release**). Exhaust continues until the valve closes the port travelling in the opposite direction (**point of compression**). From this point compression takes place until the valve is again opened to admission. The maximum steam port opening *d*, is usually less than the width of port *c*, and is reached when the valve has travelled its maximum distance from mid position. The amount which the steam port is open when the piston is at the end of the stroke is the **"lead,"** or **"steam lead."** Similarly the amount which the exhaust port is open is the **"exhaust lead,"** commonly called **"release."** To bring the edge of the valve into proper position relative to the edge of the steam port when the piston is at the end of the stroke, it is necessary to move the eccentric through an angle which will move the valve from its central position a distance equal to the steam lap plus the steam lead. This angle is called the **"angle of advance."**

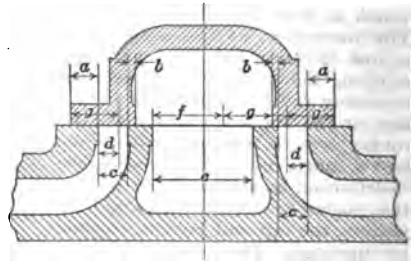


FIG. 71.—Elementary Slide Valve.

Valve Functions. If there were neither Lap nor Lead the valve would be central each time the piston reached the end of the stroke and steam would enter the cylinder throughout one stroke and be exhausted throughout the next. There would be no expansion and no compression. The use of lap makes cut-off and expansion possible. It is customary to make the top **Steam Lap** slightly larger than the bottom lap to lessen the difference between the top and bottom cut-offs. Steam laps of 1 in. to 2½ in. are commonly used in marine engines. Positive **Exhaust Lap** retards the period

of release and makes compression earlier while negative exhaust lap has the opposite effect. The amount of lap and whether positive or negative is determined by the exhaust opening which should be wide open at the end of the stroke. Exhaust lap, if negative, should always be less than the steam lap to avoid a passage for steam from one end of the cylinder to the other at the moments of admission and cut-off. Frequently exhaust lap is zero but it may be as much as ± 1 in. **Steam Lead** is generally settled according to the judgment of the designer. The inertia of the moving parts relative to the area of the piston should be taken into consideration in determining lead as ordinarily compression is not sufficient to overcome the inertia of the moving parts by the time the piston reaches the end of the stroke. Lead therefore **helps to cushion the piston** and should be greater for large than for small cylinders. It may vary from $\frac{1}{8}$-in. in small H.P. cylinders to $1\frac{1}{2}$ in. in large L.P. cylinders. On account of the angularity of the connecting rod the bottom lead is always made more than the top lead. Top lap plus lead must always equal bottom lap plus lead. As linking up increases the lead with open rods and decreases it with crossed rods, as shown in Fig. 67, **normal lead** is made greater with crossed rods. **Lead with Marshall, Joy and other radial valve gears** is constant for all grades of expansion. The **cut-off** for equal power in each cylinder of a multiple expansion engine depends on the relative volumes of the cylinders as discussed on p. 834. With Stephenson Link Motion the bottom cut-off is from 5 to 9 per cent less than the top cut-off due to the angularity of the connecting rod. **Release** (sometimes called Exhaust Lead) should be early enough to give a free passage for exhaust to avoid wire drawing; 7 to 16 per cent is the range for marine engines. **Compression** should not be carried beyond the pressure of admission; it varies from about 6 per cent in H.P. cylinders to about 12 per cent in L.P. cylinders. The **Angle of Advance** is the **Lap Angle** plus **Lead Angle** and may range from 33 deg. for H.P. to 45 deg. for L.P. cylinders. With valves taking outside steam the eccentric precedes the crank 90 deg. plus angle of advance and with valves taking inside steam the eccentric follows the crank 90 deg. minus angle of advance. Although the **angularity of the connecting rod** affects the position of the piston with reference to the crank, the length of the eccentric rod with reference to the throw of the eccentric is so great that the displacement of the valve is considered equivalent to the vertical travel of the centre of the eccentric and the valve travel equal to twice the radius of the eccentric.

Valve Travel. It is important that the travel of the valve be kept as short as practicable to reduce frictional loss. The travel of a piston valve may be reduced by increasing the diameter and of a slide valve by making it double ported, or in case of very large cylinders with long strokes, triple ported. The valve travel of all cylinders is kept the same if possible but it is often found, when working out the valve diagram, that it is better to increase the travel of the L.P. valve. The travel is usually 2 to 3 times the width of the steam port.

Angularity of the Connecting Rod. Inspection of Fig. 72 will show that for the same angular position of the crank relative to the beginning of the stroke, the piston moves a much greater distance on the downstroke than on the upstroke. If Fig. 72 represents the positions of the piston and crank at cut-off, it is evident that with a connecting rod of infinite length, the piston would have traveled equal distances and *ac* and *db* would be equal to *en* and *gk* and the cut-offs would be equal; but the angularity of the connecting rod increases the cut-off on the downstroke and reduces it on the upstroke

making $ac = eo$ and $db = gm$. Actually a movement of the crank of 120 deg., with a connecting rod length equal to 4 times the crank, corresponds to a cut-off or piston travel of 80 per cent on the downstroke and 70 per cent on the upstroke. To partly overcome this difference the top steam lap is made larger than the bottom. However, this difference with Stephenson link motion, can never be reduced below about 5 per cent.

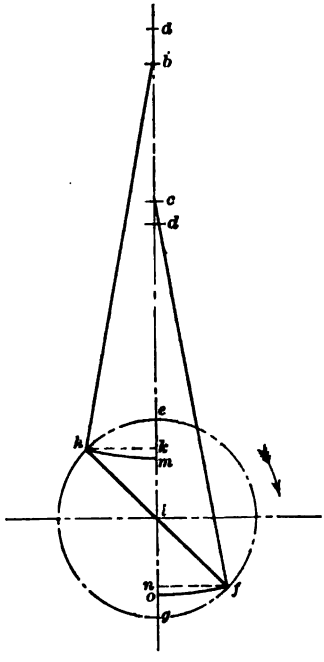


FIG. 72.—Effect of Angularity of the Connecting Rod.

Zeuner Valve Diagram. In laying down a valve diagram the valve travel, cut-off, steam lead and exhaust release are assumed and the other functions determined by trial. The Zeuner valve diagram is generally used for Stephenson Link Motion and is constructed as follows:

In Fig. 73 let the line AB be the direction of travel of the valve. With O as a centre describe the circle DKE with radius equal to the eccentricity (full size) and also the crank pin circle $A3B$ reduced to a convenient scale. Assume A top

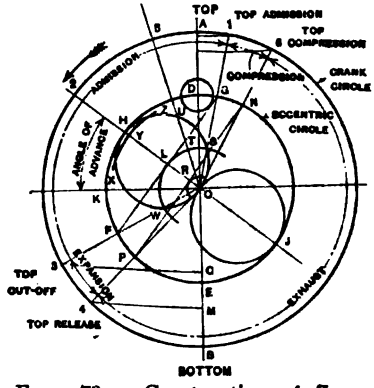


FIG. 73.—Construction of Zeuner Valve Diagram.

and B bottom of stroke. Offset AC from A equal to the top cut-off and draw the corresponding crank position $O3$, using the length of the connecting rod reduced to scale, as a radius. With D as a centre draw a circle with radius equal to the top steam lead and from F a line FG tangent to this circle. Bisect this line with a normal HJ through O. A circle HSO with HO as diameter will represent the movement of the valve from the central position. The angle HOK is the **angle of advance**, LO the **steam lap** and LH the **maximum steam opening** which occurs at crank position $O2$. The **steam opening for any other crank position** $O5$ is TU. Draw $O1$ through G which passes through the intersection of the steam lap circle WLS and the valve circle HSO. The line $O3$ also passes through the intersection of these circles. The points S and W therefore represent the **opening and closing of the valve**. Complete the construction for the bottom of the stroke in a

similar manner. Offset BM from B equal to the top exhaust release and draw the crank position $O4$. Also draw $O6$ making angle $6O2$ equal to angle $4O2$. The point 6 on the crank circle is the crank position when **top compression begins.** With a straight line connect points N and P and, with O as a centre, draw a circle tangent to NP. The radius OR of this circle is the **exhaust lap.** If this circle lies within circle HO the lap is **negative** and if within circle OJ the lap is **positive.** When the width of the port is less than HL an arc XYZ is drawn making the distance YL equal to the port opening. The steam lap, lead, release and valve travel must be adjusted until considered

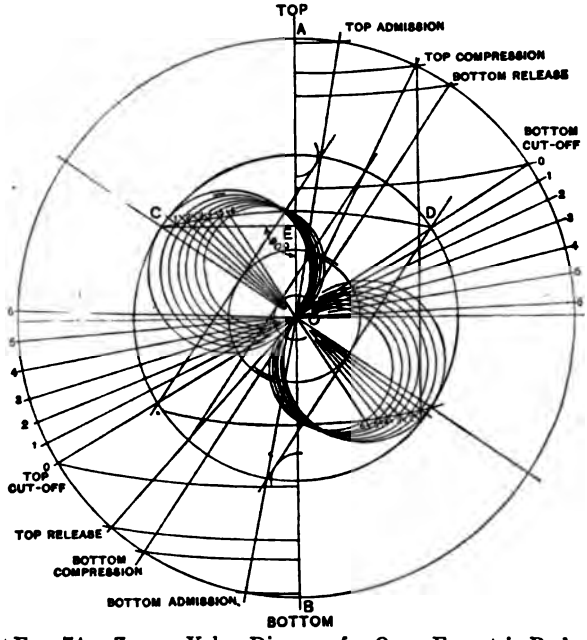


FIG. 74.—Zeuner Valve Diagram for Open Eccentric Rods.

satisfactory for the required cut-off. The diagram is always laid off in the same manner relative to the top of the stroke whether the valve takes steam inside or outside. A separate diagram must be drawn for each cylinder. To find the valve functions for linked up positions, see Fig. 74, which is a **complete diagram for open rods.** The arc CD is described through point C from a centre on OB extended and with a radius equal to

$$\frac{\text{Length of ecc. rod} \times 2 \times EC \text{ (Fig. 74)}}{2 \times \text{dist. bet. ecc. rod pins on link}}$$

Considering the length of the arc CD as equivalent to the length of the link, lay off proportionally from E the points 1, 2, 3, *etc.* to represent linking

up of 1, 2, 3, etc. inches, and describe circles with *O1*, *O2*, *O3*, *etc.* as diameters. From these circles the points of admission, cut-off, release and compression, corresponding to the linked up positions, may be found in the same manner as for the full gear condition first described. **For crossed rods** the arc *CD* is layed off from a centre on *OA* extended instead of on *OB* as shown in Fig. 75 which is a complete diagram for crossed rods.

In some cases the links are arranged for carrying the cut-off beyond the point of attachment of the eccentric rods thus increasing the valve travel

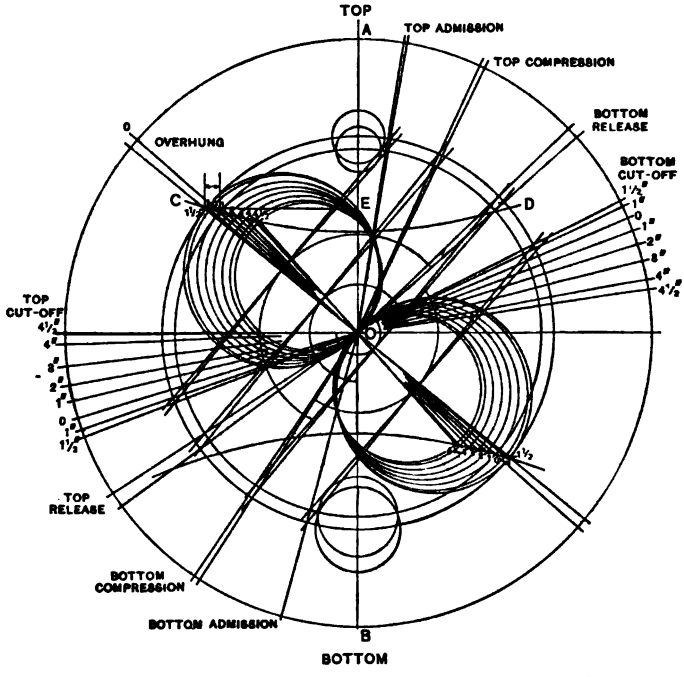


FIG. 75.—Zeuner Valve Diagram for Crossed Eccentric Rods.

and decreasing the angle of advance; this may be represented on the diagram by a proportionate extension of the equivalent link line *CD*, Fig. 75, beyond the point *C* and is generally called "**overhung gear.**" The difference between open and crossed rods can best be understood by comparing the valve functions for different cut-offs as shown by the curves in Fig. 67, which are drawn for valve gears with open and crossed rods suitable for the same engine. For crossed rods in this case it was found advisable to overhang the links 1¼ in. to get the benefit of larger valve travel and the proper range of cut-off without reducing too much the steam opening and steam lead when linked up. The large steam opening and steam lead at maximum linked

out position necessary to obtain good running conditions in the maximum linked up position may be noted.

The Oval Valve Diagram is frequently used in connection with radial valve gears. Its construction for Marshall valve gear is as follows:

The path of the lower valve rod pin *b*, during one revolution, is plotted by laying off to a convenient scale or by use of a simple model, the locations of the centres of the eccentric rod, the suspension pin *d* on the bell crank and the attachment of the swinging link *B* to the eccentric rod at *a*, Fig. 69. On the eccentric circle *ABC*, Fig. 76, with a radius = $\frac{2 \times \text{ecc.} \times \text{length of con. rod}}{\text{stroke}}$, arcs are drawn corresponding to each tenth position of the piston. The intersections of these arcs with the eccentric circle marked 1, 2, 3, *etc.* for the downstroke and 1′, 2′, 3′, *etc.* for the upstroke, are used in plotting the path of the lower valve rod pin, which is done for full gear ahead, maximum linked up, and astern positions as shown. Points on the path of the valve rod pin are marked to correspond to the piston positions. In Fig. 77, draw line *DE* representing the stroke of the piston to a convenient scale and divide into tenths. On the left side of line *DE* plot the positions of the valve for the downstroke of the piston which are found by scaling off the vertical distances of the points on the path of the valve rod pin from the arc *FG*, the radius of which is the length of the valve rod. On the right side of *DE* plot the positions for the upstroke. A line through these points will describe an oval figure representing the

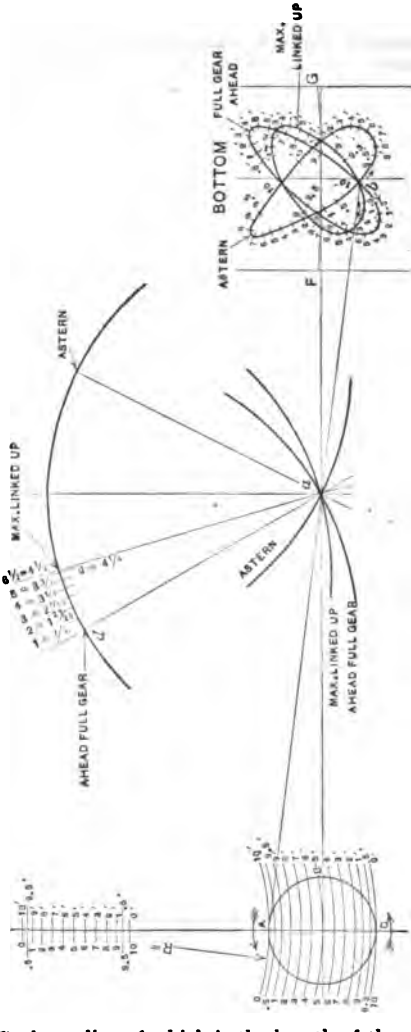


FIG. 76.—Construction of Oval Valve Diagram for Marshall Gear.

positions of the valve for the corresponding positions of the piston. Draw the different lap lines for steam and exhaust parallel to *DE*, the left side representing the downstroke and the right side the upstroke. The intersections of the lap lines and the oval represent the **points of admission, cut-off, exhaust release and compression.** If the exhaust lap is positive the lap line is drawn on the opposite side of its steam lap and if negative on the same side. In the diagram shown in Fig. 77, the top exhaust lap is zero. A separate diagram must be drawn for each cylinder. The diagram shows that increasing the steam lap decreases the lead and the cut-off and increasing the exhaust lap makes release later and compression earlier. Due to the angularity of the connecting rod and the curvature of the path of the point *a*, Fig. 69, the oval is not a true ellipse but one end is fuller than the other which shows that the top and bottom cut-offs are unequal. To partly overcome this the laps are made unequal. As lead is constant for all grades of expansions ovals drawn for linked up positions will all pass through the same top and bottom points of lead.

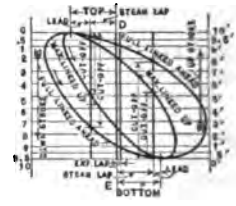


FIG. 77.—Oval Valve Diagram for Marshall Gear.

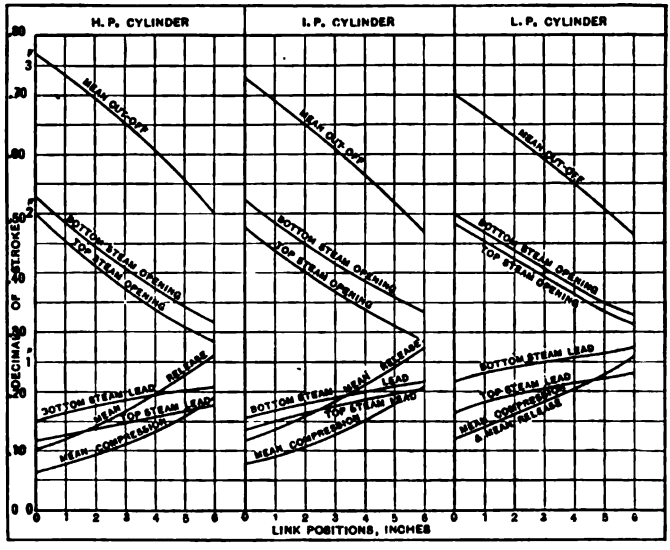


FIG. 78.—Changes in Valve Functions for Different Link Positions.

Valve Diagram Data. When the valve diagrams have been drawn for each cylinder the data is tabulated on the drawing for easy reference as shown in Table 16. It is also well to plot on the drawing, curves similar to Fig. 78 showing the changes in the valve functions for different link positions so that

Table 16. Valve Diagram Data

(See Fig. 78)

Engine 24½-41½-72 × 48 in. stroke I.H.P. = 2200. R.P.M. = 70 Piston speed = 560
Stephenson link motion—open rods Length of connecting rod 9 ft. 0 in.

	H.P.		I.P.		L.P.	
No. and size of valves	One 11" at top 10¼" at bottom		One 22 3/16" at top 22" at bottom		One double ported slide valve	
Side on which steam is taken	Inside		Outside		Outside	
Angle of advance	34°–5°		37°–20′		40°–40′	
Eccentricity and valve travel	3½–7		3½–7		3½–7	
	Top	**Bottom**	**Top**	**Bottom**	**Top**	**Bottom**
Data for full gear						
Steam lap in inches	1½	1⅜	1 9/16	1½	1 13/32	1 13/32
Exhaust lap in inches	−¼	0	−½	+1/16	−5/16	+5/16
Steam lead in inches	15/32	19/32	9/16	5/8	1 1/16	7/8
Cut-off in inches	38 1/16	35 1/16	36¾	32 13/16	35 5/16	32 5/16
Cut-off in dec. of stroke	0.794	0.731	0.7656	0.683	0.735	0.670
Mean cut-off in dec. of stroke	0.7622		0.7243		0.7025	
Exhaust release in inches	5 1/16	5	5 7/8	5 9/16	5 7/8	5 3/8
Exhaust release in dec. of stroke	0.1055	0.1041	0.1224	0.1160	0.1225	0.112
Compression in inches	2 7/16	3 5/16	3 5/8	4 5/16	5 3/4	5 7/8
Compression in dec. of stroke	0.0599	0.069	0.0755	0.0872	0.1120	0.1225
Width of port in liner, inches	3	3	3¼	3¼	2×2½	2×2½
Steam opening in liner, inches	2	2⅛	1 13/16	2	2×1 39/32	2×2 5/32
Exhaust opening in liner	Full port		Full port		Full port	
Area of steam opening in liner, sq. in.	53.8	54.0	103.0	106.0	250.0	274.0
Area of exhaust opening in liner, sq. in.	80.62	76.12	172.25	172.25	327.5	327.5
Area of port in cylinder, sq. in.	58.1	58.1	140.0	140.0	327.5	327.5
Velocity through steam opening in liner	4900	4620	7360	7010	9120	8260
Velocity through exhaust opening in liner	3275	3275	4400	4310	6960	6910
Velocity through port in cylinder	4540	4295	5410	5300	6960	6910
Data for 6 inch linked up position						
Cut-off in dec. of stroke	0.555	0.463	0.522	0.423	0.5025	0.430
Mean cut-off in dec. of stroke	0.509		0.4725		0.466	
Steam opening, inches	1 5/32	1 9/32	1 5/16	1¼	1¼	1 1/16

the cut-off, release, compression, etc., may be easily read for any position of the links. Table 16 and Fig. 78 are from the valve diagram of an actual engine.

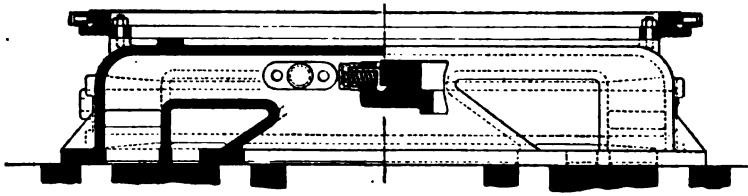


FIG. 79.—Double Ported Slide Valve with Relief Ring.

Flat Slide Valves. On cylinders in which the pressure will not cause excessive wear, it is customary to use a slide valve. If the cylinder is large

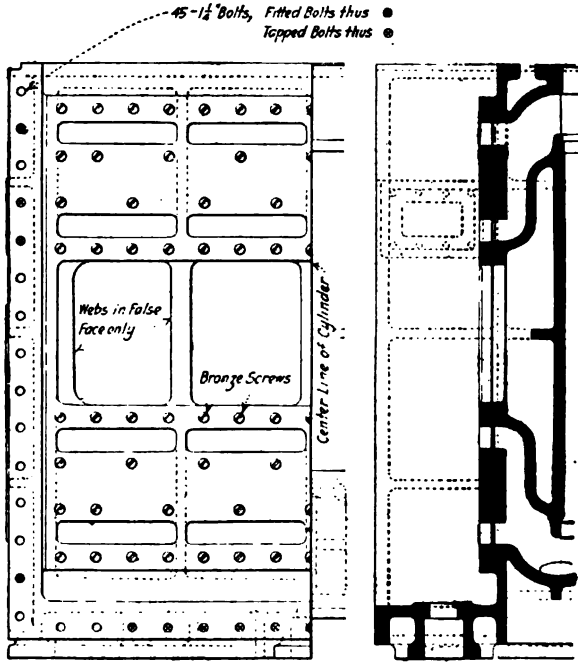


FIG. 80.—Slide Valve False Face.

this valve may be double ported as shown in Fig. 79 or even triple ported to reduce the travel and to give quick closing ports. The faces of the valve and seat are sometimes scraped to a true surface but usually if these faces are

planed at right angles they will soon wear to a good surface. To reduce friction, grooves are often cut in the bearing surface of the valve or seat so steam may enter and act as a lubricant. Care must be taken that the valve has sufficient bearing surface in order to keep the pressure per sq. in. within proper limits. If the ports are very wide one or more central bars must be used. Even though the pressure per square inch on a slide valve may be low, the total pressure, due to the large area of the valve, and consequently the force required to move it is large. For this reason, it is customary to use a relief or balance ring, as shown in Fig. 79, or similar type to relieve the steam pressure on the back of the valve, and thus reduce the wear on the valve face and seat. The ring is held steamtight against a planed surface on the inside of the valve chest cover by springs and the space inside of the ring is connected to the condenser or to the receiver of the next cylinder. These springs also serve to reseat the valve in case it is lifted off its seat by water in the cylinder or any other cause.

Slide Valve False Face. A hard close grained cast iron false face or liner is generally fitted to the cylinder as a working surface for a slide valve as shown in Fig. 80. This liner varies from about 1 to 2 in. thick depending on the size of the engine and is firmly secured to the cylinder by a number of bronze screws set about ⅛ in. below the surface to allow for wear.

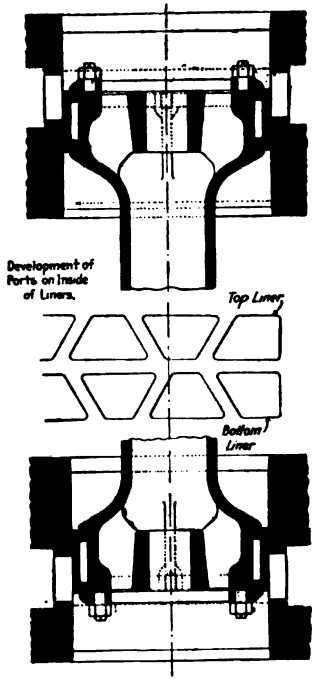


FIG. 81.—Cast Iron Piston Valve.

Piston Valves. It is customary to use a piston valve on the H.P. cylinder as the wear of a slide valve is excessive under high pressure. The volumetric clearance is more than with a slide valve, due to the steam passages extending around the valve but the use of straight ports reduces this difference. A piston valve, if properly fitted, works with little friction but is not as steamtight as a slide valve and as it cannot lift from its seat if dirt or an obstruction enters the port it is liable to stick and throw severe strains on the valve gear. For the same reason water in the cylinder is unable to escape. Piston valves take "**inside steam**" when the steam enters the cylinder past the inner edges of the valve rings and "**outside steam**" when the steam enters past the outer edges. The general type of piston valve is shown in Fig. 81. Large piston valves are often made with cast steel ends riveted to a central section of steel tubing as shown in Fig. 82. The rings are sometimes held against the valve liner by springs but on account of excessive wear it is better to use solid rings or, to provide for adjustment, to have the rings cut and the ends fastened together by screws or bolts as shown in Fig. 82. The diameter of the rings may then be increased by the use of liners. On I.P. cylinders,

either piston or slide valves are used, according to the idea of the designer. To avoid excessive distances from the centre of the cylinder, two piston valves may be used instead of one as discussed under "Laying down the Engine," p. 918.

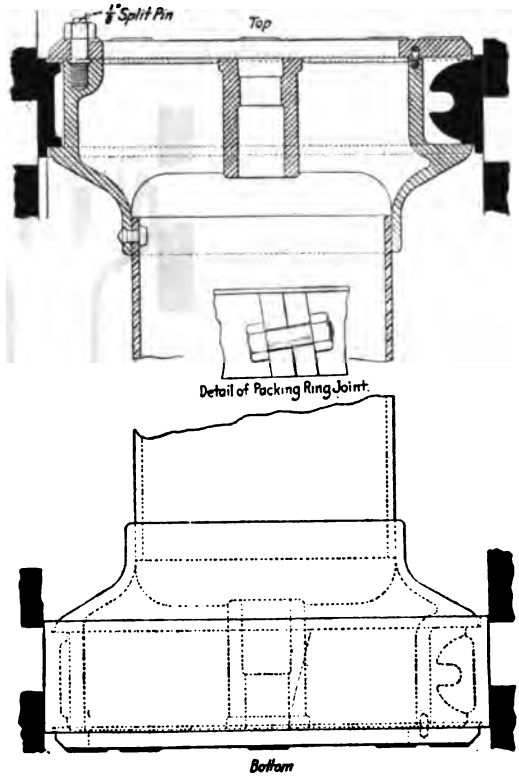


FIG. 82.—Steel Piston Valve.

Piston Valve Liners. A hard close grained cast iron liner is usually fitted in the valve chest as a working surface for a piston valve as shown in Fig. 81. The thickness varies from about $\frac{7}{8}$ in. to $1\frac{1}{4}$ in. depending on the size of the engine. The bridges in the steam port, see Fig. 81, are arranged diagonally to insure equal wear on the valve rings and are usually about 25 per cent of the circumference of the valve. The openings in the valve liners are accurately machined. The ends of the liners are counterbored to allow the valve rings to overrun a short distance at each end.

Expansion Valves. With the ordinary types of valve gear and the usual valve laps a variation of cut-off of 15 per cent to 25 per cent may be obtained. If a greater

range or an earlier cut-off is desired a separate expansion or cut-off valve, consisting of an auxiliary valve, working on top of or within the main valve and driven by a separate eccentric is fitted. This valve is arranged so that the cut-off desired may be obtained by changing its position by means of a screw. To obtain a wide range the main valve is usually designed for a very late cut-off as the auxiliary valve can only effect an earlier cut-off. The other functions of the main valve are not changed. Although seldom fitted at the present time there is a large number of engines with these valves still in service.

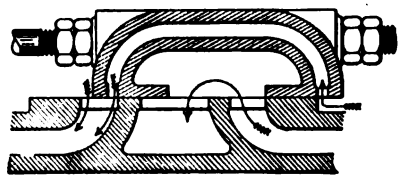


FIG. 83.—Trick Slide Valve.

Trick Valves. To obtain a quicker and fuller opening of the steam port trick valves are sometimes used. Fig. 83 shows a trick slide valve and Fig. 84 a trick piston valve. Passages are cored through the valve and so arranged that their entrances just overlap the end of the valve seat or liner when the valve is at the end of its stroke. As the valve uncovers the steam port at one end the cored passages admit steam to the same port from the other end of the valve. The width of the trick ports is such that they are completely open to steam at the same time as the steam port. Exhaust takes place only through the inner passage of the main valve in the same manner as valves of the ordinary type. The valve diagram for a trick valve is drawn in the same manner as for an ordinary valve keeping in mind that the steam opening is the sum of the openings of the main and the trick valves.

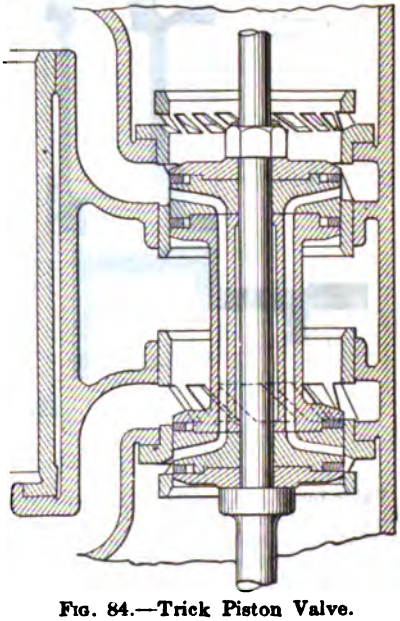


FIG. 84.—Trick Piston Valve.

Balance Cylinders and Pistons. To relieve the eccentrics of a large portion of their load, a balance cylinder with its piston attached to an extension of the valve stem is often fitted in the upper valve chest cover of heavy piston or flat slide valves as shown in Figs. 106 and 107. The under side of the piston is in communication with the valve chest or receiver and the upper side connected to the condenser or a steam chest of lower pressure. The area of the piston is made such as to just support the weight of the valve and gear so that the valve load is made up entirely of friction. In case of the L.P. valve, where the low steam pressure in the chest would necessitate a very large balance piston, the under side of the piston is sometimes connected to the I.P. instead of to the L.P. chest or receiver. The pistons are solid cast iron packed with several small cast iron rings. The balance piston acts as a guide for the upper end of the valve stem.

Lovekin Assistant Cylinders. In this type of balance cylinder, Fig. 85, the admission of steam to the under side of the piston is controlled by the movement of the piston past a steam port. The steam is taken directly from the chest and the action is automatic. Its purpose is to balance the inertia force of the valve gear on the downstroke as well as the weight, by compression of steam on the under side of the piston to a pressure higher than that of the valve chest. In cases where the inertia force is much greater than the weight of the valve gear steam is also admitted on top of the piston to provide compression on the up stroke as well as the down stroke. In this case the steam required is only that necessary to replace the steam which condenses as there are no exhaust ports.

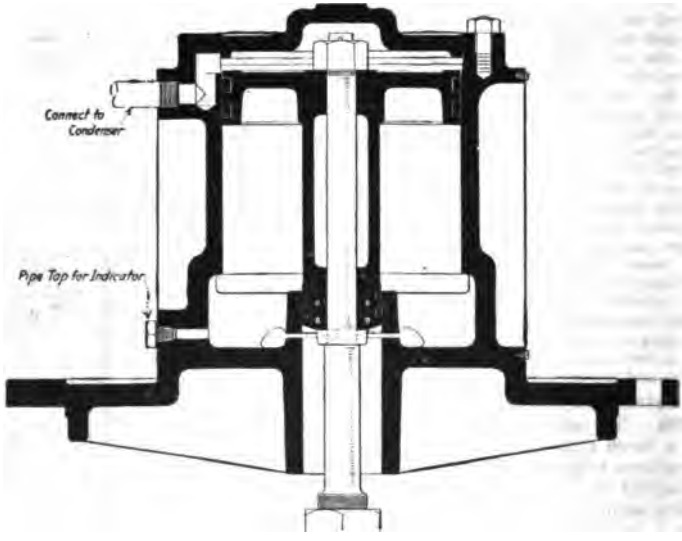


FIG. 85.—Lovekin Assistant Cylinder.

Valve Load. A number of formulæ are in use for obtaining the loads on the valves in order to proportion bearing surfaces and the strength of the various parts of the valve gear. Some formulæ take into account the weight and inertia of the valve gear and others the friction of the valves. Any formula must be more or less empirical and it is well to simplify the calculation by basing the load on the friction of the valves only and keeping the pressures and stresses low enough to cover all other contingencies. The pressures and stresses mentioned under valve gear details have been found by successful practice to be satisfactory, and are based on valve loads found by the following formulæ:

$$\text{Valve load in lbs.} = L = c \times b \times l \times p \text{ for slide valves}$$
$$= c \times \pi d \times l_1 \times p \text{ for piston valves}$$

in which c = coefficient of friction; b and l = width and length of slide valve; d and l_1 = outside diameter and length of top + bottom piston valve rings;

and p = unbalanced absolute pressure between the steam and exhaust sides of the valve. The constant c should be taken 0.15 for I.P. and L.P. slide valves and 0.22 for H.P. and I.P. piston valves which values are high enough to cover almost dry surfaces, a condition which is liable to occur. For L.P. cylinders, p should be taken not less than 30 lbs. for compound, 25 lbs. for triple and 20 lbs. for quadruple engines; for other cylinders p may be taken from Table 2. For slide valves the total area of the back of the valve is used, neglecting the effect of the relief ring if any, as the valve gear must be proportioned to operate the valve should the relief ring fail to work. The load for piston valves depends on the fit of the valve rings. With tight rings and dry surfaces the load may exceed that given by the formula. The valve loads for all cylinders should be determined and, unless the loads are materially different, the valve gear proportioned from the largest load and the parts made the same throughout the engine. Often, however, the valve load for the L.P. cylinder is much greater than for the other cylinders and it is not unusual to increase the size of the parts for this cylinder. The following details of valve gear cover Stephenson Link Motion but the details of other types may be designed in the same manner by using the valve load L as a basis.

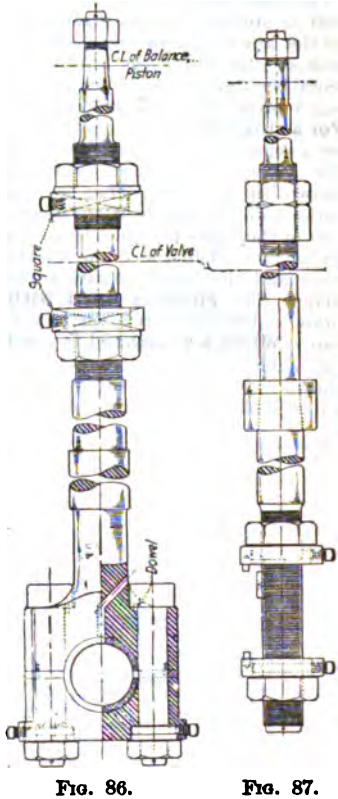


FIG. 86. FIG. 87.
Valve Stems.

Valve Stems. When one valve is used the valve stem is fitted with a **cap and adjustable brass bushing** at the lower end, Fig. 86, which attaches to the link block. When there are two valves the stem, Fig. 87, is **screwed into a crosshead,** which connects to the link block. Piston valves are secured to the stem by a collar and washer below and lock nuts above the valve. **Adjustment of the valve along the stem** is made by changing the thickness of the washer. **To provide for wear** of a slide valve and to allow the steam pressure to hold it firmly against its seat, the hole through which the stem passes is made oval. Nuts above and below the valve are locked against a pipe distance piece over the stem or in some other manner so that the valve is free on the stem. **The diameter** of the valve stem below the valve is determined by the method given under Piston Rods, p. 858, using the valve load L and a factor of safety of about 30. The stress at the bottom of the thread in the valve should not exceed 3000 lbs. The cap and bolts are proportioned in a manner similar to connecting rod caps and bolts using the same allowable stresses. The stems, collars, nuts, caps and bolts are forged steel.

Valve Stem Crossheads. When two piston valves are used a crosshead, usually of cast steel, of a design similar to Fig. 89 is used. The lower end of the crosshead carries the bushing for attaching to the link block.

Valve Stem Guides. The valve stem must be guided at the lower end to prevent the side strain due to the angularity of the eccentric rods or the action of the links, from bending the stem or throwing a pressure on the stuffing box. **For a single valve** this guide may be a separate casting or cast with the lower valve chest cover as shown in Fig. 88. The bearing surface will be ample if the diameter in the guide is made ¼ in. to ½ in. more than the diam. of the stem and the length 3 times the diam. The **guide is lined with** brass and fitted with a removable cap. **When a crosshead is used** the guide is cast iron or steel as shown in Fig. 89 with adjustable brass gibs fitted in the crosshead. As one valve is liable to work much stiffer than the other and throw continual

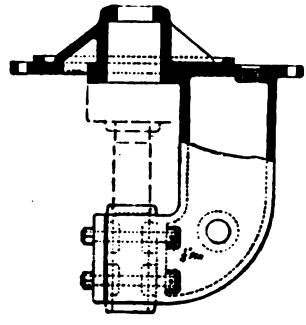


FIG. 88.—Valve Stem Guide.

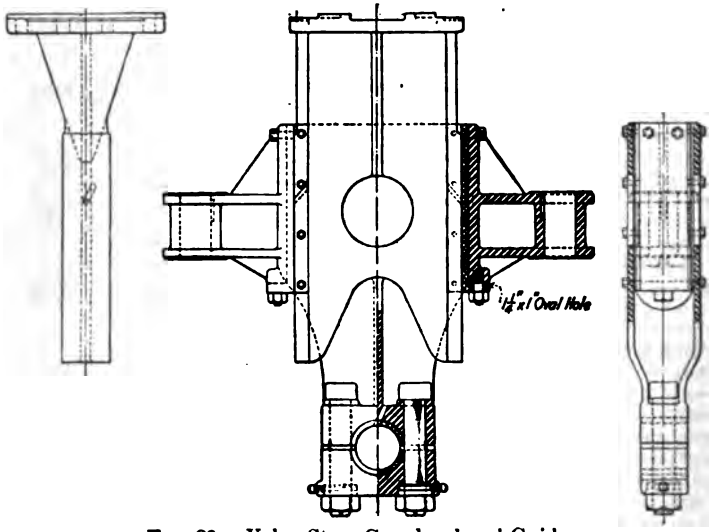


FIG. 89.—Valve Stem Crosshead and Guide.

pressure on the guides, it is advisable to make the bearing surface almost double that used for a single valve. The upper end of the valve stem is guided by the balance piston or by a bronze bushing in a cast iron hood.

Links. The double bar type, Fig. 90, is generally used. The mean radius is the distance between the centre of the eccentric and the centre of the link. The links are forged steel machined all over and held together at the ends by bolts through cast iron distance pieces. The four forged steel eccentric rod pins, two of which are extended to form the suspension rod pins, are riveted through the link bars. The suspension rods, for Merchant engines, are generally attached to the go-ahead eccentric rod pins to reduce the slotting action of the link block in ahead gear. The **depth of the bars** is made about three times the thickness and the **distance between eccentric rod pins** six to seven times the eccentricity. The **eccentric rod pins are proportioned** for a bearing pressure of 600 to 800 lbs. per sq. in. The **area of the suspension rod pins** is about ¾ that of the eccentric rod pins. Each

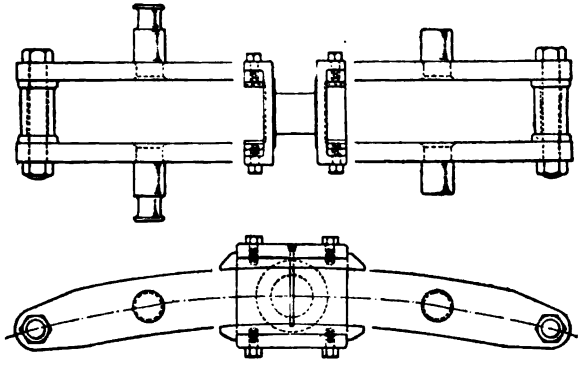


FIG. 90.—Links and Link Block.

link bar is proportioned as a beam, supported at the eccentric rod pins, to carry **half** the valve load, *L*, in the middle; 5000 to 6000 lbs. stress may be allowed.

Link Blocks. Fig. 90 shows a forged steel link block of the usual type. The pin for the attachment of the valve stem is proportioned to carry the valve load with a pressure of 600 to 950 lbs. per sq. in. The **length** is made equal to the diameter. Brass faces or gibs proportioned for a press. of 250 to 350 lbs. per sq. in., are fitted between the link block and the links.

Eccentric Rods. The **diameter** of the eccentric rod is determined in the same manner as a connecting rod using hinged ends, the valve load, *L*, a length equal to the distance between the centres of the eccentric strap and the pins on the links, and a factor of safety of 18 to 22. The **diameter of the lower end** is made about 1.3 times that of the upper end. The rods are forged steel fitted at the upper end with adjustable brass brushings secured by a forged steel or brass cap. The fork of the ahead rod is usually made symmetrical and the astern rod offset, see Fig. 107. The **fork should be proportioned** as explained under connecting rods, allowing a stress of about 4000 lbs. The lower end of the rod is secured to the eccentric strap by two bolts which must be proportioned to carry the valve load. The length of the eccentric rod varies from 8 to 15 times the valve travel.

Eccentrics. The usual type of eccentric, Fig. 91, is of cast iron although the smaller part is sometimes forged or cast steel. The parts are held together by bolts about 1 in. diam. for a 12 in. to 2 in. for a 30 in. eccentric. The keyway, which is always in the large part, is often made wider than the key and side keys fitted so that small changes in the angle of advance may be made. **To secure the eccentric against moving sideways** a set screw is fitted or the ahead and astern eccentrics are bolted together. The joint between the parts is stepped to take side strain off the bolts. The **width of the eccentric** is determined from the valve load, L, the projected area being such that the press. per sq. in. is 60 to 140 lbs.

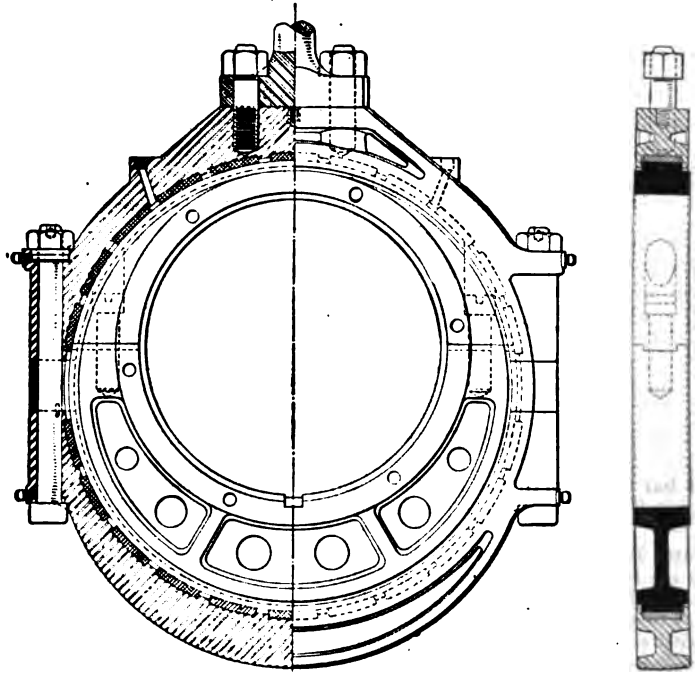


FIG. 91.—Eccentric and Eccentric Straps.

Eccentric Straps. Fig. 91 shows a cast steel eccentric strap lined with white metal of the type generally used. A lip at each side of the bearing surface keeps the strap in place. The strap **should be sufficiently stiff** not to pull away from the eccentric where bolted to the rod nor cause excessive friction by closing in at the sides. The bolts connecting the top and bottom must safely carry the load on the valve gear. As spare eccentric straps are usually carried, the top and bottom parts are often made from the same pattern to reduce the number of spare parts; see Fig. 107, p. 917. The lower

part of the strap is proportioned by considering it as a beam supported at the centres of the bolts and loaded in the middle with the valve load, L, allowing a stress of not over 4000 lbs. for cast steel.

REVERSING GEAR

For small engines reversing is accomplished by a hand lever attached directly to the reverse shaft or by some hand operated gear, either worm and

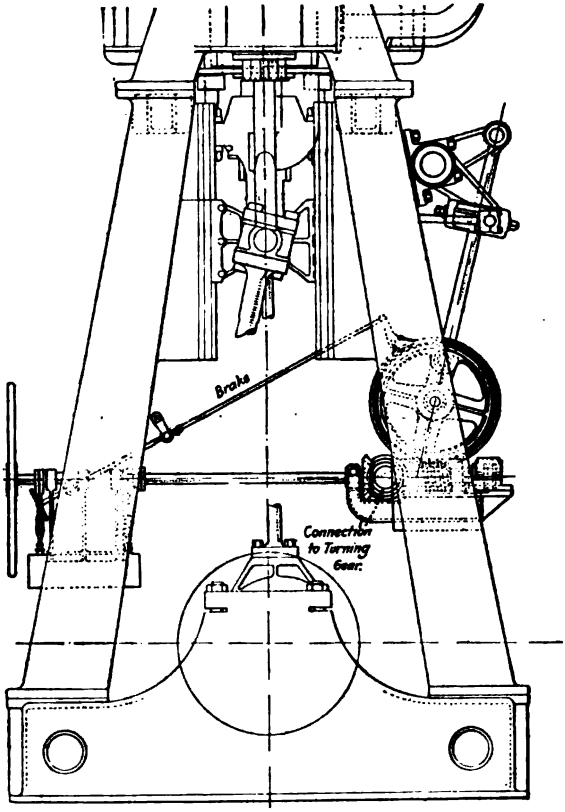


FIG. 92.—All-round Reversing Gear.

wheel, or screw. For large engines, steam reversing gear is fitted. The "all-round" type is almost universally used in European practice, and the "direct-acting" type in American practice.

The "all-round" type, Fig. 92, consists of a small one- or two-cylinder engine driving a worm and worm wheel. The worm wheel is connected to an arm on the reverse shaft by a connecting rod and a crank shaft or a

connecting rod and a pin in the rim of the wheel. The engine is usually fitted with a fly wheel of large diameter with a light smooth rim readily accessible for use as a hand wheel for starting the reversing engine when on dead centre, or in making slight movements of the valves. The fly wheel also forms an emergency means of starting the valve gear in case it should stick. The "all-round" type of gear is also useful in "warming-up" as the reversing engine may be kept running and continuously reversing the valves. It is less liable to be damaged by careless handling, and the engine may also be arranged to operate the turning gear. The valve gear for the reversing engine is sometimes fitted with a "follow-up" gear which stops the reversing engine when the gear of the main engine has reached the desired position. An indicator is usually fitted so that the valve position may be readily seen. Sometimes a brake is fitted so that the engine may be stopped quickly and held at any point.

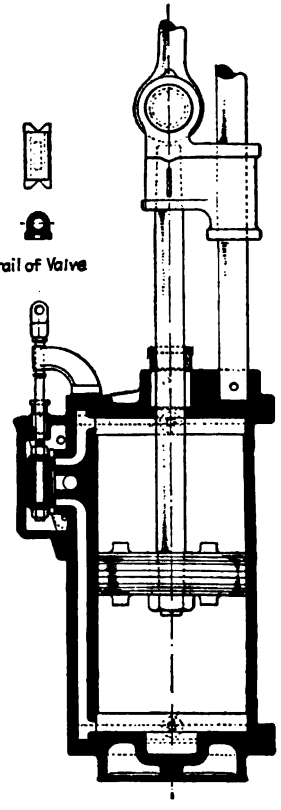


FIG. 93.—Direct Acting Reversing Engine.

The "Direct-acting" Type, Figs. 93 and 106 consists of a cylinder with a connecting rod directly connected to the reverse shaft arm, one stroke of the piston moving the reverse shaft from "ahead" to "astern" position. The valve gear of this engine is operated by a hand lever fitted with a quadrant for securing it. A "follow-up" gear stops the reversing engine when the gear of the main engine has reached the desired position. The piston should be provided with stops at each end of the stroke consisting of springs or of bosses on the piston which strike the cylinder head. The valve of the reversing engine is usually of the "*D*" slide type. A "V" notch is often cut in the steam lap at each end of the valve, Fig. 93, which makes the steam balance maintained by the follow-up gear more sensitive and the movement of the piston more easily controlled by slight movements of the valve. To provide a cushioning effect, the reversing gear of a large installation sometimes has an oil or water cylinder in tandem with the steam cylinder with a valve also in tandem, so that any movement of the steam valve and piston causes a corresponding movement of the hydraulic valve and piston allowing the fluid to flow from one end of the hydraulic cylinder to the other. The fluid may be throttled, so as to regulate the speed of the movement. Sometimes the hydraulic cylinder is also fitted with an auxiliary hand pump which may be used as a hand reversing gear or as an emergency gear in case the main engine valve gear sticks.

The size of reversing engine is determined more or less empirically, but with constants taken from practice, the following formula, adapted from *Bauer*, gives good results: $d^2s = \frac{l \times t}{p \times n} \times c$ where d and s = diam. and stroke of reversing engine, l = maximum load on one main engine valve (see Valve Load), t = half travel of the valves, p = boiler press. (gage), n = rev. of reversing engine required to reverse main engine (= 1 for direct-acting type), and c = constant. All dimensions are in inches. If a double-cylinder all-round engine is used d^2 is the sum of the squares of the two cylinders. For three-cylinder triple expansion engines with cranks at 120 deg., *Bauer's* values of c, modified to agree with the above formula, are 7 for direct-acting, and 17 for all-round type; and for four-cylinder triple and quadruple expansion engines with cranks at 90 deg., 10 for direct-acting, and 23 for all-round type. However, reversing engines are often made of such size that the constants are from 1½ to 2 times these values.

For the "direct-acting" type the diam. of the reversing cylinder may be taken roughly as 0.15 to 0.23 times the diam. of the L.P. cylinder for all types of engines and the stroke 2 to 3 times the valve travel. For the all-round type it is usual to make the stroke about equal to the cylinder diam.

Reversing Gear Load. The various parts of the reversing gear are proportioned from the **reversing engine load, P = Ap**, where A = area of reversing engine cylinder and p = steam press., which for purposes of design is taken as boiler press. As this maximum load comes on the gear only in case the valves stick, stresses figured by this method may run much higher than ordinary working stresses. For simplicity and in view of the margin of safety in using boiler press. the angularity of the reversing engine connecting rod and the suspension rods is neglected.

Reversing Engine Piston Rod, Connecting Rod and Crosshead. The piston rod is forged steel tapered at each end and secured to the piston and the crosshead by nuts. The **crosshead**, Fig. 106, is of cast steel or composition, and carries the forged steel connecting rod pin and the crosshead guide shoe which is composition bushed if the crosshead is of steel. The **connecting rod**, Fig. 106, is a steel forging with a solid brass bushing at the crosshead end and a split brass bushing with steel cap at the other end. The piston rod and connecting rod are proportioned by the method given under main engine piston and connecting rods, using a factor of safety of 8 to 10. The direct tension in the piston rod at the base of the threads may run as high as 10,000 lbs. The stress in the crosshead pin is figured in the same manner as a main crosshead pin of the same type, a stress of about 5000 lbs. being allowable. The bearing pressure on the crosshead pin may be as high as 3500 lbs. per sq. in. The **length of the connecting rod** is usually about 1½ to 2 times the stroke of the reversing engine.

Reverse Shaft. The reverse shaft, Fig. 106, is forged steel and supported by bearings bolted to the engine housings or to the cylinders. The stress in the shaft is figured from the equivalent twisting moment. Usually at one or both ends of the shaft a reversing arm is located outside of the bearing, so that it is overhung. The **reversing engine arm should be located** near the middle of the shaft to keep torsional stresses low, and near a bearing to reduce bending. The equivalent twisting moment is figured at each reversing arm to determine the maximum value between bearings and overhung. The stresses are then found by dividing the largest equivalent twisting moment for each condition by the resistance. If T = twisting moment and B =

bending moment, the equivalent twisting moment $= B + B\sqrt{1 + (T + B)^2}$. The resistance to twisting $= \frac{D^3}{5.1}$ where D is the shaft diameter. The maximum twisting moment occurs at the reversing engine arm, and $= P \times L$, where L = length of the arm. Usually the maximum equivalent twisting moment between bearings will be at the reversing engine arm. If there is also a reversing arm between the same bearings the bending moment used is the resultant of the components of the bending moments. For reversing arms at the ends of the shaft the twisting moment is the product of the load on the reversing arm times the length of the arm. For reversing arms between the ends and the reversing engine arm the twisting moment will be the sum of the twisting moments of all the arms between the end of the shaft and that point. Stresses in reverse shafts figured by this method may safely run from 8000 to 10,000 lbs.

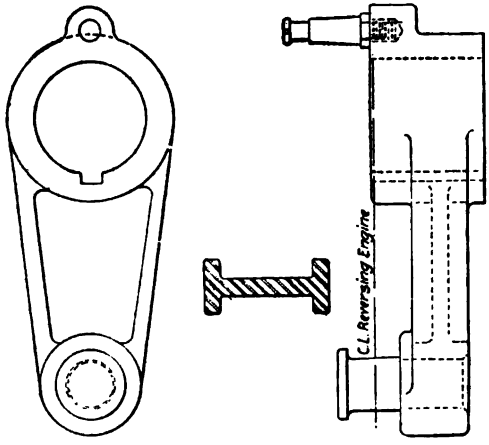


FIG. 94.—Reversing Engine Arm.

The reversing engine arm, Fig. 94, usually of cast steel of I section, is figured for bending stress near the hub due to the reversing engine load P. If b = distance from centre of the pin to the section taken and R = resistance of the section, stress $= \frac{P \times b}{R}$. This stress may safely be 10,000 lbs. The pin in the reversing engine arm is proportioned for a bearing pressure of about 3000 lbs. due to the load P, and also for bending stress. If l = length of pin and d = diam. in the arm, stress $= \frac{P \times l \times 10.2}{2 \times d^3}$. The allowable stress may run as high as 18,000 lbs. Diam of reversing engine arm boss = 1.6 times diam. of the shaft and the length equal to shaft diam. The lengths of the reversing engine arm and the reversing arms are proportioned approximately so that

$$\frac{\text{Distance bet. ecc. rod pins on links}}{\text{Length of reversing arm}} = \frac{\text{Length of reversing arm}}{\text{Length of reversing engine arm}}.$$

The follow-up gear of the reversing engine is often attached to a pin in the boss of the reversing engine arm as shown in Fig. 94.

The reversing arms, Fig. 95, are usually of cast steel of *I* section and should be placed near a bearing to reduce bending moments in the shaft. Each reversing arm is fitted with a screw working in a crosshead having a pin at each end, to which the suspension rods are attached so that the linkage of any cylinder may be varied by adjusting the screw. **The slot in the reversing arm** should be approximately parallel with the suspension rods in ahead gear and perpendicular to them in astern gear so that linking up has no effect when going astern. The reversing arms are also figured for bending stress near the hub. Let Q = the load on the arm, b_1 = distance from centre of crosshead to the section taken and R_1 = resistance of the section. As the reversing arms for all cylinders are usually made the same, R_1 and b_1 will be the same for all cylinders, but Q will vary in proportion to the valve loads.

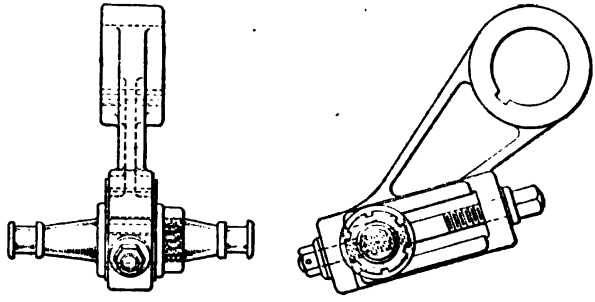


FIG. 95.—Reversing Arm.

If L = length of reversing engine arm from centre of pin to centre of shaft, L_1 = corresponding length of the reversing arm, then the **total load on all reversing arms,** $P_1 = P \times \frac{L}{L_1}$. The load P_1 is divided among the reversing arms in proportion to the valve loads, that is, the load Q on any reversing arm is equal to $P_1 \times$ the ratio of the valve load for that cylinder to the sum of the valve loads. Stresses may safely run as high as 10,000 lbs. The diam. of the reversing arm boss = 1.6 times and the length 0.8 times diam. of shaft. The adjusting screw is figured for direct tension at the bottom of the threads, the load being Q and the allowable stress about 5000 lbs.

The reversing arm crosshead, Fig. 95, is figured for bending stress in a section through the middle, through which the adjusting screw passes. If L = distance between centres of suspension rod pins, B = width and D = depth of middle section, and d = diam. of hole for the adjusting screw, stress $= \frac{Q \times L \times 6}{4 \times D^2(B-d)}$ which may run to 6000 lbs. The crosshead is also figured as a cantilever at the collar, where the tapered portion joins the central block. This section is usually circular and if l = distance from centre of pin to section taken and d = diameter of section, stress $= \frac{Q \times l \times 10.2}{2 \times d^3}$. This

stress is usually higher than the stress in the central block, and runs about 8000 lbs. The pins are proportioned for a bearing pressure of about 1000 lbs.

The suspension rod, Fig. 96, is figured as a column with hinged ends, loaded with half the reversing arm load. The lowest factor of safety is usually in the L.P. rods and runs about 30 to 40. The caps and bolts are figured in the usual manner.

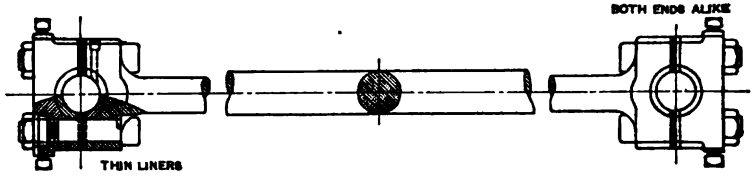


FIG. 96.—Suspension Rod.

TURNING GEAR

Provision is always made for turning the engine for the purpose of setting the valves, making repairs, or for keeping the bearings and working parts in condition when in port by turning the engine several revolutions when considered necessary. A simple form of turning gear used on **small engines** consists of recesses in the coupling, or a notched wheel attached to the shaft to which a bar or jack may be applied. For **larger engines,** a worm wheel is fitted on the shaft or coupling and operated by a worm which may be disengaged by having the worm-shaft pivoted or the worm working on a feather on the shaft thus allowing it to be moved axially out of gear when the turning gear is not in use. This worm is usually actuated by a small single or double-cylinder engine although a ratchet lever is often used on engines of small size. If hand operation is used, the ratio of revolutions of the worm to the revolutions of the engine is about 50 to 1. If an engine is used, a second worm and wheel are introduced, making the ratio of gearing such that the revolutions of the turning engine required for one revolution of the main engine is from 1200 to 2000 or even more for very large engines. The turning engine operates at 200 to 400 R.P.M. and it required 5 to 8 min. to turn the main engine one revolution. The engine should be of sufficient size to turn over the main engine in port with steam press. of 80 lbs. or less. A flywheel is usually fitted and the engine has a change valve, links or shifting eccentric for reversing so the main engine may be turned in either direction. Holes in the rim of the flywheel for inserting a bar provide means for turning the engine by hand. The engine is generally located on the engine housing or bedplate, a typical arrangement being shown in Fig. 97. If an all-round reversing engine is used it is generally also arranged for turning through a belt, chain or shafting and a separate engine is not required.

The following formula adapted from *Bragg*, may be used for determining the **size of turning engine:** $d^2s = \frac{D^2 \times S}{R} \times C$ where d and s = diam. and stroke of turning engine, D and S = diam. of L.P. cylinder and stroke of main engine, R = ratio of gearing and C = constant. All dimensions are in inches. If a double-cylinder turning engine is used d^2 is the sum of the squares of the two cylinders, and if there are two L.P. cylinders D^2 is the sum of the squares of the two. For single-cylinder turning engines a safe value of the

constant C is 1.75 for three-cylinder main engines and 2 for four-cylinder main engines. The constants may be reduced somewhat for double-cylinder turning engines. The stroke of the turning engine may be equal to the diam. of the cylinder or 1 or 2 inches less. A range of strokes of from 5 in. to 8 in. is sufficient for all sizes of main engines.

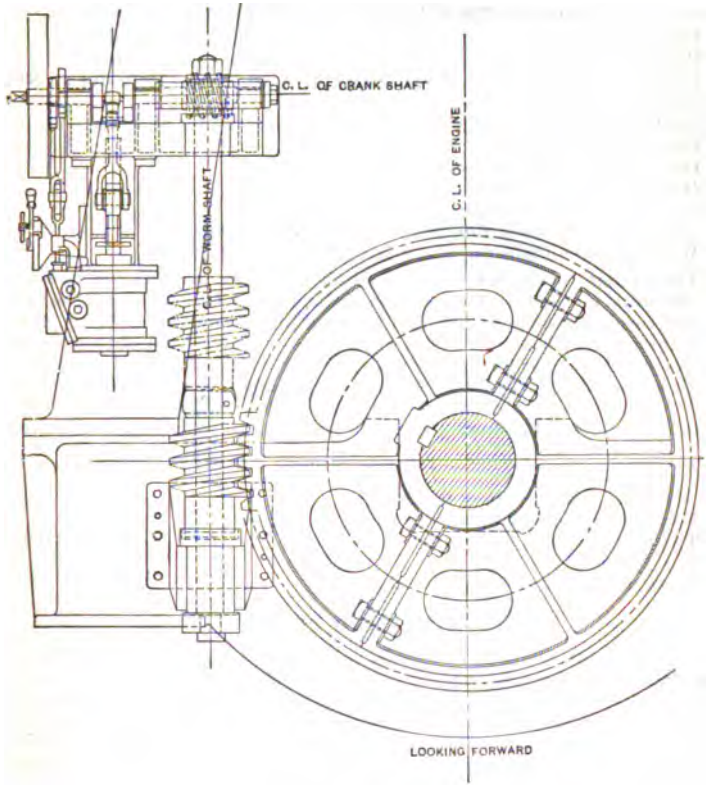


FIG. 97.—Arrangement of Turning Engine and Gear.

The size of turning engine may also be determined from the turning moment required at the main engine crankshaft, for which *Bauer* gives the formula: M (ft. lbs.) $= C \times d \times$ I.H.P. $\div$ R.P.M., in which d is the diam. of the main engine crankshaft and C a constant which is about 280 for very light engines and 560 to 670 for heavy engines. If this formula is used the turning engine should be made large enough to give at least 1.5 times the twisting moment required, with 30 to 50 lbs. M.E.P., taking into account the gear ratio and the efficiency of the gearing which will be from 13 per cent to 15 per cent.

For merchant engines the main and auxiliary worm wheels have cast teeth and are generally of cast iron. The main and auxiliary worms are single thread and generally of cast iron although sometimes forged steel or hard bronze is used. For very large engines the auxiliary wheel may be bronze with cut teeth. The main wheel is usually split which requires that an even number of teeth be used. The pitch is made in even quarter or half inches. The following **proportions of the worms and wheels** may be used:

Per cent of total gear ratio in main and auxiliary gears: 75 and 25.
Pitch diam. of main wheel: 4 × diam. main engine crankshaft.
Pitch of teeth in main wheel: 2 in. for a 40 in. to 3½ in. for an 80 in. wheel.
Pitch of teeth in auxiliary wheel: 1½ in. to 2¼ in. for all sizes.
Pitch diam. of worms: 3 × pitch.
Length of worms over teeth: 3.5 to 4.5 × pitch.
Height of teeth in wheels: 0.7 × pitch.
Height of teeth above pitch circle: 0.3 × pitch.
Thickness of teeth at pitch circle: 0.46 × pitch.
Width of face of wheels: 2 to 2.5 × pitch.

The strength of the wheel teeth, shafts and other parts should be figured by the usual methods. The loads on the teeth may be found from the power of the turning engine, assuming a mean effective pressure of 30 to 50 lbs., and taking the efficiency of each worm 40 per cent, each set of thrust collars 95 per cent, and each set of bearings 95 per cent. If n_1 and n_2 = No. of teeth in the auxiliary and main worm wheels, r_1 and r_2 = radius of the auxiliary and main worm wheels, and C = 2.0 for single and 1.5 for double-cylinder turning engines: Maximum twisting moment on turning engine crankshaft, $T = \frac{63{,}000 \times I.H.P. \times C}{R.P.M.}$. Twisting moment on auxiliary shaft, $T_1 = T \times n_1 \times 0.40 \times 0.95 \times 0.95 = 0.36T \times n_1$.

Force on auxiliary worm wheel teeth, $F_1 = T_1 \div r_1$.

Twisting moment on main engine shaft, $T_2 = T_1 \times n_2 \times 0.40 \times 0.95 \times 0.95 = 0.36T_1 \times n_2$.

Force on main worm wheel teeth, $F_2 = T_2 \div r_2$.

ATTACHED PUMPS

On Merchant Ships, with engines of about 5000 I.H.P. and under, it is customary to drive the main air and bilge pumps through links and beams from one of the main engine crossheads, usually the L.P. Frequently the feed and sanitary pumps are also driven in this manner. The pumps are vertical, single acting and the stroke is about 45 per cent of the stroke of the main engine. The arrangement of attached pumps and gear is shown in Fig. 107.

Main Air Pumps. The bucket type of (attached) air pump in which there are foot, bucket and head valves is shown in Fig. 98. The bucket, which is fitted with soft packing, has a depth of face of ¼ to ⅛ of the diam. The **pump is located so** that the condensate flows from the condenser by gravity into the bottom of the pump. The downstroke of the bucket closes the foot valves, and forces the trapped air and water through the bucket valves. On the upstroke the bucket valves are closed and the air and water pass through the head valves into the pump discharge. The upstroke forms a partial vacuum below the bucket which causes air and water to flow through the foot valves. **The Edwards type** of (attached) air pump in which there

are only head valves, Fig. 99, is now generally used. The absence of foot and bucket valves reduces the liability to breakdown and increases the efficiency. The face of the bucket is packed by water grooves as shown and is about ¼ of the diam. in depth. The condensate flows by gravity into the conical base of the pump and the bucket, which is also conical, on its down stroke, forces the water through the ports to the upper side of the

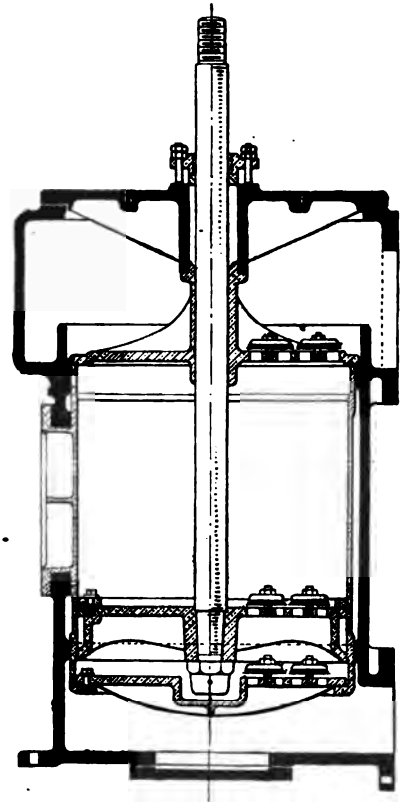


FIG. 98.—Bucket Type of Attached Air Pump.

bucket. On the upstroke the bucket closes the ports and the trapped air and water are forced through the head valves into the pump discharge. The clearances at top and bottom of the stroke are made much smaller than for the bucket type, ¼ in. to ¾ in. at the top and ¾ in. at the bottom being sufficient. The angle of the bottom and the bucket is about 130 deg. and that of the direction of discharge through the ports about 35 deg. from the

horizontal. The ports occupy about 75 per cent of the circumference of the liner and their area is 35 to 50 per cent of the area of the bucket. In this type of pump a relief valve, 1/16 to 1/8 the diam. of the pump, is always fitted at the bottom, Fig. 99, to avoid damage in starting up if an excessive amount of water has accumulated. With surface condensers the **ratio of displacement of the L.P. piston** on a double stroke **to the displacement of the air pump bucket** on a single stroke is about 28 for the bucket type and about 35 for the Edwards type. For either type the **ratio of the area through the valves to the area of the bucket** is about 0.27 and the **ratio of the area of the suction and discharge openings to the area of the bucket**

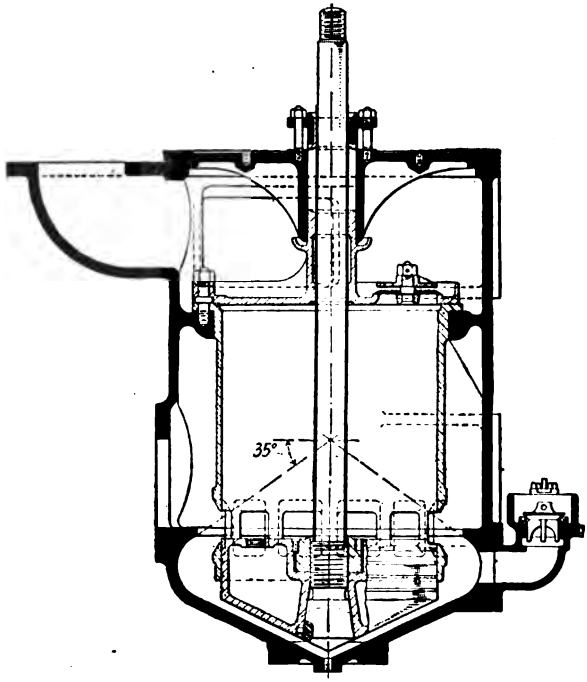


FIG. 99.—Edward's Type of Attached Air Pump.

is about 0.18. The air pump barrel is usually of cast iron fitted with a removable brass liner. In large sizes a manhole, which forms part of the cylinder wall, is fitted for access to the valves as shown in Fig. 98. The bucket in small sizes is of brass and in large sizes of cast iron with a brass casing. The pump rod is either steel with a brass sleeve or solid tobin bronze. An extension attached to the upper end of the piston rod works in a guide as shown in Fig. 107. The thickness of the cast iron barrel, bottom and cover varies from about 7/8 in. for a 20 in. to 1 in. for a 30 in. pump and the brass liner from about 1/2 in. to 3/4 in. The **valve decks** are made about 1.2 times the thickness of the liner.

Unless the pump discharge is taken off in such a manner that there is a head of water on the discharge valves, these valves are kept submerged by casting a rim around the valve deck or the pump cylinder as shown in Figs. 98 and 99. This head of water keeps the valves closed on the downstroke. A "stand pipe," open at the top, connected to and usually the same size as the discharge pipe, extends about 15 feet above the pump discharge and serves as a relief in case an abnormal amount of water enters the pump when starting up or from any other cause.

Air pump valves are now usually made of three sheet bronze discs about $\frac{3}{64}$ in. thick, Fig. 100. Rubber is liable to be affected by the oil which finds its way into the steam from the lubrication of the stuffing boxes. Lightness, which is essential to good efficiency, is obtained by using brass discs. Holes in the middle and lower discs as shown, allow the suction to act on all the discs, which assists in seating the valve and prevents the edges from curling. The valve seats, which are also brass, are in the form of grids to reduce the unsupported area of the discs. A brass guard on a stud regulates the **lift of the valve** which must be such that the circumferential discharge area is equal to the clear area through the seat. There may be one row of large or two rows of small valves.

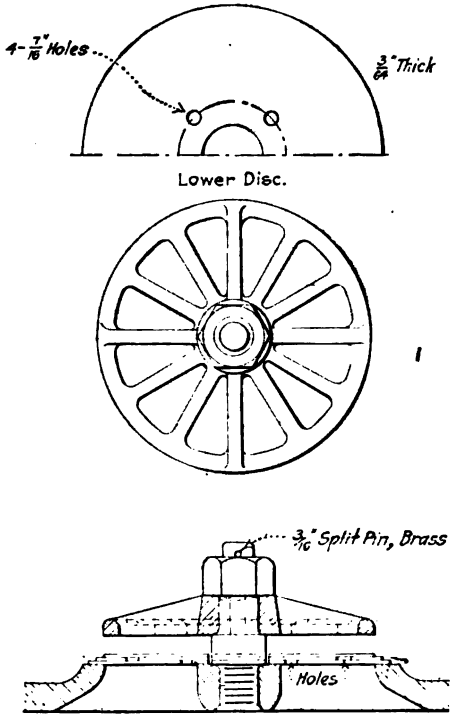


FIG. 100.—Bronze Air Pump Valve.

Feed pumps are of the plunger type, single acting and are worked from one or both ends of the air pump crosshead. The plunger works through a stuffing box fitted with metallic or soft packing. The inlet and outlet valves are carried in a valve box which is cast separate from the pump barrel to reduce the cost of replacement in case one or the other is damaged. The valves are of brass with springs to assist in seating. An air vessel and a relief valve are fitted on the pump or in the discharge piping. Feed pumps are designed for a **theoretical capacity** of about 2.25 times the boiler evaporation except for small engines where the ratio is increased. The evaporation may be assumed as 20 lbs. per I.H.P. for compound engines, 16 to 17 lbs. for triple and 15 lbs. for quadruple engines. Feed and other attached

pumps usually have cast iron barrels and valve boxes and brass plungers, but steel plungers are often used for feed pumps to avoid excessive wear. No definite ratios can be given for the sizes of bilge and sanitary pumps, as their capacities depend upon the type of vessel.

Attached Pump Gear. The beams are made up of two pairs of steel plates connected by a rocker shaft which works in bearings on the engine housings, Fig. 101. The link pins at each end are carried between the plates. Each link consists of two steel bars with bushings at the ends. The working parts are designed for the load on the pumps which is taken as 30 lbs. per sq. in. on the air pump bucket and 20 lbs. per sq. in. on the plungers of the other pumps. These pressures are high enough to cover the weight of the reciprocating parts as well as the working pressure (or vacuum). The up-

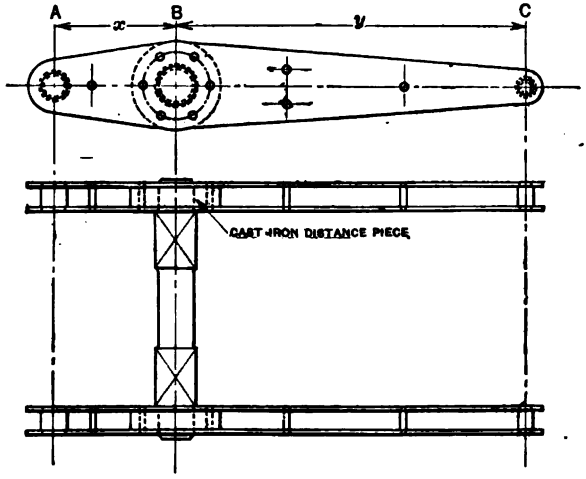


FIG. 101.—Attached Pump Beams.

stroke load is used. The pump beams are proportioned for a bending stress of about 4500 lbs. at the section B, Fig. 101. If P is the total load at the pump end, A, b and t, the width and thickness of one beam plate at B, and d the diameter of the rocker shaft in the beam plate, the **stress in the beam at** $\mathbf{B} = \frac{6P \times b}{4t(b^3 - d^3)}$. The pins are proportioned for working pressures in the link bushings of about 400 lbs. at A and 300 lbs. at B. The load at C is found by multiplying the load at A by the ratio $\frac{x}{y}$. The load on the rocker shaft at B is equal to the sum of the loads at A and C. The working press. on the rocker shaft bearings is about 150 lbs. The air pump load only is used when figuring the pump rod and the bending stress in the middle of the pump crosshead. If the breadth and depth of an average section in the middle of the pump crosshead are b and h, the average diam. of the hole d, the distance be-

tween the centres of the link bearings l and the load on the pump rod P_1, the **stress in the central block** $= \frac{P_1 l}{4h^2(b - d)}$. The air pump rod is figured as a column fixed at the ends, by the usual method, using a factor of safety of about 20, and also for tensile strength at the bottom of the threads, the allowable stress being about 3500 lbs. The link rods and caps are proportioned by the usual methods.

PIPING AND FITTINGS

Throttle Valves. The ability to maneuver quickly requires that the throttle valve be balanced for easy handling and arranged for convenient operation from the working platform either by a lever or a handwheel. The double seated valve, sometimes called the double beat poppet, shown in Fig. 102, is the type most commonly used. One disk is made slightly larger than the other so that the unbalanced pressure will act to close the valve yet not be so great as to cause difficulty in opening it by hand. To reduce the uneven expansion between the valve disks and the valve body the distance piece between the disks is generally made of the same material as the body. If the valve is operated by a hand lever it is well to fit a **gag screw** with a handwheel to limit the stroke or to lock the valve tight when in port. **Large single seated valves should be avoided** as they are slow in closing and require special balancing arrangements which are delicate and liable to give trouble.

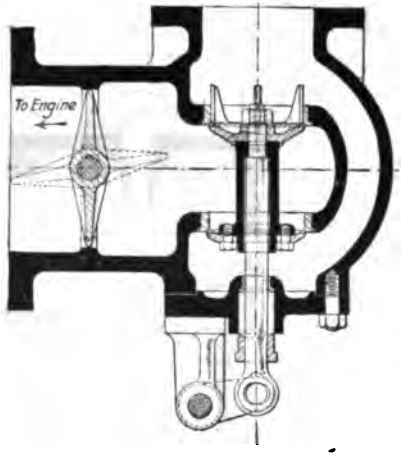


FIG. 102.—Double Beat Poppet Throttle Valve and Butterfly Valve.

A throttle valve **fitted with balance piston and pilot valve** is shown in Fig. 103. Unless this valve is carefully designed it will not be balanced sufficiently to operate with a hand lever. The balance piston should be on the engine side of the valve where it is less apt to chatter than if on the steam side. The **combined disc and cylindrical valve**, fitted with a pilot valve as shown in Fig. 104 (from *Bauer*) is easy to balance and gives good adjustment but is rather slow to open and shut. Steam leaking through the valve liner presses the disc firmly on its seat. As the spindle rises the pilot valve is opened, the steam above the disk escapes and the main valve is easily operated. The flow of steam is regulated by the uncovering of the openings in the valve liner.

A butterfly valve, Fig. 102, is often fitted in the steam line in addition to the throttle valve for quick handling when the engine is racing. The angle of travel of the valve is generally from 60 deg. to 90 deg. The valve is controlled by a hand lever from the working platform and is attached to the governor gear if fitted.

Governors. To prevent racing in bad weather or if the engine is suddenly relieved of its load a governor is often fitted. The **governor may be actuated** by the pressure under the stern of the vessel or by the revolutions of the engine above a predetermined speed. In the first type a pipe is lead from a chamber connected to the water at the stern to a pressure chamber at the engine fitted with a diaphragm. Any change in draft at the stern due to pitching of the vessel will vary the pressure on the diaphragm which, through suitable gear, operates the reversing engine which in turn controls the speed of

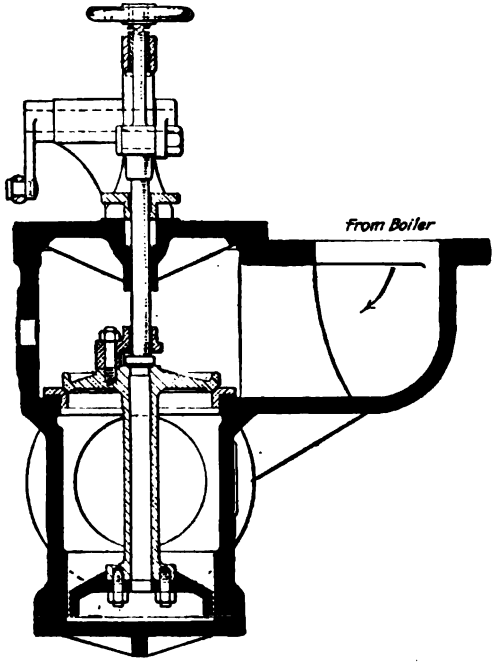


FIG. 103.—Throttle Valve with Balance Piston and Pilot Valve.

the engine by moving the links. Governors of the second type may make use of the centrifugal force of revolving weights or of the force of inertia. Governors operating on the latter principle are more frequently used, one of the most popular being the **Aspinall** shown in Fig. 105. This device is carried on the air pump beam or on a rocker arm supported on one of the housings and actuated by a link attached to the crosshead. The mechanism consists of a spring retarded weight, *W*, whose inertia at speeds of about 5 per cent in excess of a predetermined speed, is sufficient to overcome the spring and set the pawls, *PP*, so that the lower pawl on the next upstroke, picks up the lever, *H*, which closes the throttle valve. When the speed of the engine has returned to normal the pivoted weight is returned to its original position

by the springs, and the detent, *D*, alters the position of the pawls, *PP*, so that the upper pawl picks up the lever, *H*, and opens the throttle. Should the speed continue excessive the repeated movement of the weight causes the first pawl to remain set, and the throttle remains closed until the speed has become normal. This governor is also fitted with an emergency weight, *A*, which comes into use in the case of very excessive speed such as would result from losing a propeller or breaking a shaft. The action of the weight, *A*, causes the weight, *W*, to be locked which results in the throttle remaining shut until the engine stops and the throttle can be unlocked. This governor may be arranged to operate the reversing gear instead of the throttle.

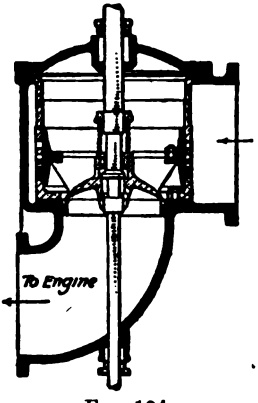


FIG. 104.

Handling Gear. All gear necessary for the control and maneuvering of the engine should be located at one point and arranged for quick and easy handling under emergency conditions. The levers for operating the throttle, butterfly valve and reversing engine are usually located together on a bracket fitted to the side of the forward or second housing at a convenient height from the engine room floor. The receiver steam and cylinder drain controls are located on the same side of the engine as the throttle control but not necessarily at the same point. The butterfly valve, used for quick control of the engine when racing due to pitching of the vessel in a heavy sea, is also connected to the governor, if fitted, for automatic control under such conditions. The various water service and lubricating oil cocks are placed where most convenient.

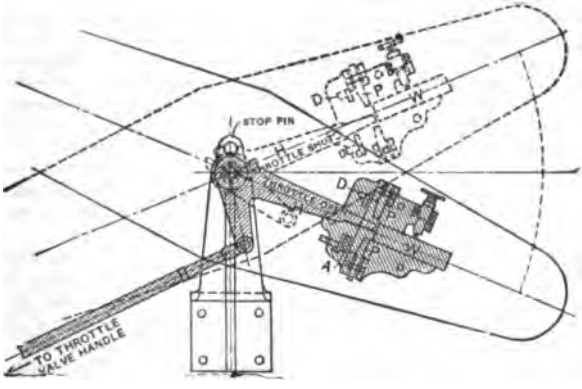


FIG. 105.—Aspinall Governor.

Working Platform and Gratings. The engine controls are handled from the engine room floor but gratings should be placed wherever necessary to provide access for oiling, indicating, and making repairs on the main engine as well as for access to other equipment about the engine room. The usual

arrangement consists of a grating all around the engine below the bottom of the cylinders with cross gratings between the housings for access to the crossheads and stuffing boxes. Another grating is fitted all around the engine near the top of the cylinders for use when indicating or making repairs on cylinders or pistons.

Lubrication. Merchant engines are fitted with gravity lubrication to all bearings, crosshead guides and thrust and line shaft bearings. Oil cups or boxes may be fitted directly on the part to be lubricated but it is usual to provide a box or reservoir, of a capacity to last several hours, on the side of each cylinder from which pipes are lead to the cups or boxes on the engine. These boxes or reservoirs have small tubes fitted with wicks leading up through the oil, which by capillary attraction, and siphon effect, feed the oil through a sight feed to the point to be lubricated. The size and number of wicks control the flow. Receptacles mounted on rapidly moving parts into which oil is dropped from the distributing system should be packed with curled hair held in place by a perforated cover or wires to prevent the oil being thrown out and to keep dirt from plugging the pipes. An oil box is mounted on each main bearing cap with one or more tubes feeding the bearing. A cup is fitted on each eccentric rod with a pipe leading down to the eccentric strap. For all valve gear, air pump, and reversing and turning engine parts, where the movement is intermittent, an oil cup of commercial make is usually fitted. The crosshead guides have oil piped to both the forward and after sides and lead by suitable drillings to the face of the guide, on which are oil retaining grooves. Each stroke a wiper on the crosshead dips into a box fitted at the bottom of the guide and distributes on the face of the guide the oil that runs down into the cup. Oil boxes on top of the crosshead pin caps, lubricate the crosshead pins through drillings in the caps. The same boxes lubricate the crank pin through one or two pipes lead down the connecting rod. The thrust bearing is lubricated by a cup with capillary feeders on each shoe with oil channels lead to the face of each collar. The base of the thrust bearing is usually filled with oil in which the lower part of the collars are immersed. The line shaft bearings are lubricated either by oil cups with capillary feeders mounted on the caps or by a block of grease placed in an opening in the cap which lubricates by rubbing on the shaft. The piston rods and valve stems are lubricated by oil swabs while the engine is running, the oil thus carried into the cylinder usually being all that is required for internal lubrication with saturated steam. With superheated steam, oil must be forced into the cylinder by a lubricator. The oil is distributed around the bearings by grooves cut in the white metal so arranged that oil will not leak out the ends of the bearings. The grooves are about $\frac{1}{4}$ in. to $\frac{1}{2}$ in. wide and $\frac{1}{8}$ in. to $\frac{1}{4}$ in. deep.

Water Service. Connections for circulating sea water are lead to the parts of the main engine, thrust, line shaft and stern tube bearings which are liable to heat. These connections may circulate water in cored passages in the part to be cooled, thereby providing "indirect" cooling or the water may be allowed to flow on the part to be cooled thus provided "direct" cooling. The main bearings and crosshead guides are generally fitted with indirect service. Direct service through $\frac{1}{2}$-in. to $\frac{3}{4}$-in. pipes taken off a main supply pipe, is supplied for almost every bearing, in addition to hose connections for emergency use in case of excessive local heating. These pipes are on adjustable fittings to direct the flow of water within certain limits and are fitted with valves so that they may be operated individually. The crank pins are often served by a transverse perforated pipe on each side distributing

water thereby from above during a complete revolution. The thrust shoes are cored for indirect water circulation and sometimes cooling coils are fitted in the oil chamber in the base. Direct water service is also provided by branch pipes similar to those on the main engine and so arranged that water may be played on each collar. The bases of the line shaft bearings are usually piped for water circulation, while the cap may be fitted with direct service from above. The supply of water is taken from the discharge side of the main circulating or other pump, and in the case of indirect service, which is usually in use when the main engine is running, this water is either returned to some pump suction or to the bilge. The circulation for the guides is often taken from the sanitary pump in which case the water is returned to the sanitary system. The water used in direct service drains to the bilge. The water service system for the main engine is sometimes kept separate from the system supplying the thrust bearing and line shaft bearings on large installations, but usually one system suffices.

Gages and Gage Board. The engine room gage board is usually located on the forward engine room bulkhead near the engine control gear and within easy sight of the engineer handling the throttle. A clock and the following gages are fitted on the gage board: Main Steam or Boiler Steam; H.P. Chest; I.P. Receiver; L.P. Receiver; Vacuum.

Tools and Wrenches. The engine room tool equipment should include a full set of mechanics hand tools such as vise, wrenches, hammers, chisels, files, taps, dies, screw jack, chain falls, soldering copper, ratchet drill, etc. Also a full set of special wrenches for the main engine and large wrenches for the stern tube nut and propeller nut.

Lifting Gear. Suitable lifting gear should be provided over the engine for lifting the cylinder covers and drawing the pistons and valves. This gear consists of one or more chain blocks of suitable capacity hung from travellers fitted to a strong I beam placed fore and aft above the centreline of the engine at a height sufficient to lift out the piston with rod attached. In the bottoms of the cylinders are tapped holes for eye bolts for attaching chain falls for handling the main bearing caps and other heavy parts.

Arrangement of Cylinders and Cranks. The order in which the cylinders are arranged beginning with the forward cylinder is called the order of cylinders. Different orders of cylinders is discussed under Balancing, p. 932. The order in which the cranks pass the top centre when the engine is going ahead is called the sequence of cranks. The H.P. is commonly called the leading crank when the sequence is H.P., I.P., L.P. Similarly the L.P. is the leading crank when the sequence is L.P., I.P., H.P. A right-hand engine has the working platform on the starboard side and the cranks in passing their top position turn to the starboard. In other words a right hand engine turns clockwise when viewed from the after end.

Engine Designs. Fig. 106 shows a triple expansion engine in which the ports are short but not as short as if the valve chests were extended above and below the cylinders. The crankshaft is in three interchangeable and reversible sections, so that one section of spare shaft may be used for any cylinder with either end forward. The top and bottom of the eccentric straps are made from the same pattern to reduce the number of spare parts carried. Fig. 107 shows a quadruple expansion engine with the ports as short as it is possible to make them. The crankshaft is in four interchangeable but not reversible sections. The position of the links and reversing arms indicates that the engine shown in Fig. 106 has open eccentric rods and the engine in

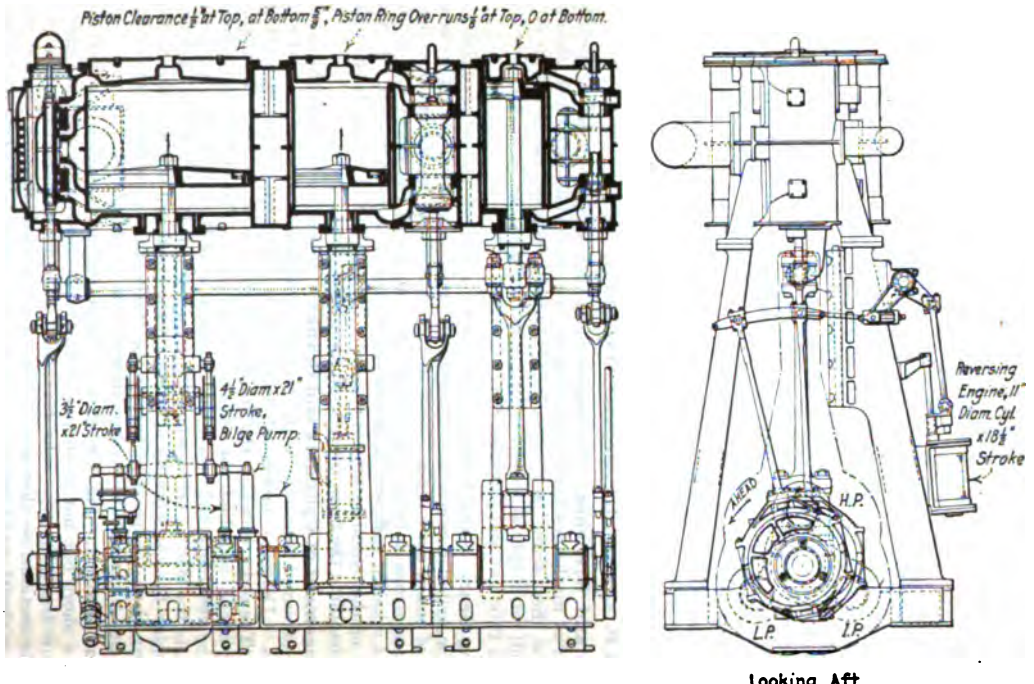

Piston Clearance 1/8" at Top, at Bottom 5/8", Piston Ring Over runs 1/8" at Top, 0 at Bottom.
3½" Diam. x21" Stroke
4½" Diam x21" Stroke, Bilge Pump.
Reversing Engine, 11" Diam. Cyl. x18" Stroke
AHEAD
H.P.
L.P.
L.P.
Looking Aft.

FIG. 106.—Triple Expansion Engine. Designed and Built by the Newport News Shipbuilding and Dry Dock Company.

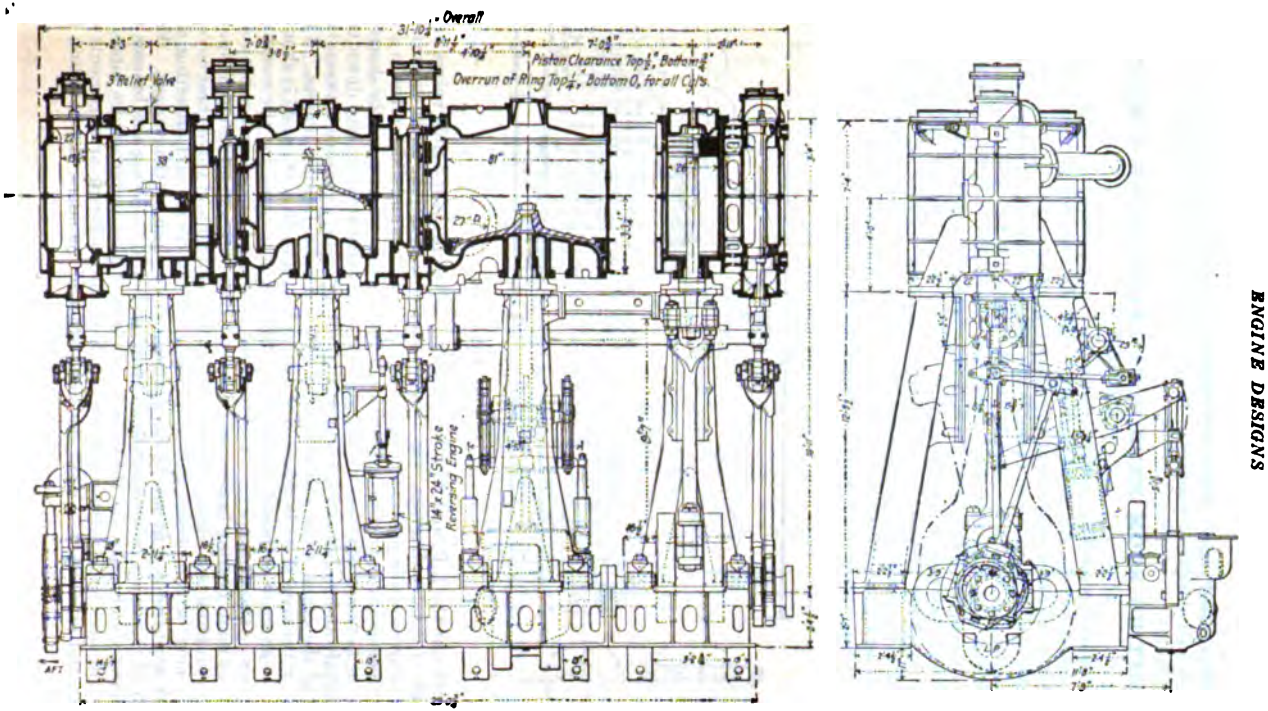

Overall
3" Relief Valve
Piston Clearance Top 1/8", Bottom 3/4"
Overrun of Ring Top 1/4", Bottom 0, for all Cyls.
W. 24" Stroke Reversing Engine

FIG. 107.—Quadruple Expansion Engine. Designed and Built by the Newport News Shipbuilding and Dry Dock Company.

Fig. 107, crossed rods. These are representative Merchant engines, the proportions and details of which are in accordance with the methods given in this section.

DRAFTING ROOM METHOD OF LAYING DOWN A MARINE ENGINE

The following method used in the design of a quadruple expansion engine is presented by *Mr. B. Meurk formerly of the Engineering Department of the Newport News Shipbuilding and Dry Dock Company:*

In the engine taken as an example the crankshaft is in four interchangeable sections; piston valves are used in the H.P. and 1st I.P. and slide valves in the 2d I.P. and L.P. cylinders; liners are fitted in the H.P., 1st I.P. and 2d I.P. cylinders. The size of cylinders, crankshaft, coupling bolts, connecting rod and bolts, crosshead and main bearings are all determined according to the methods previously described. The best balance of a quadruple expansion engine is obtained when the order of cylinders is H.P., L.P., 2d I.P. and 1st I.P. and when the sequence of cranks is H.P., 1st. I.P., L.P. and 2d I.P. and this arrangement only is discussed.

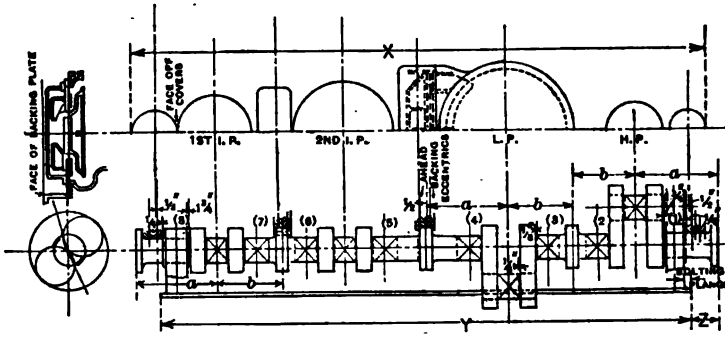


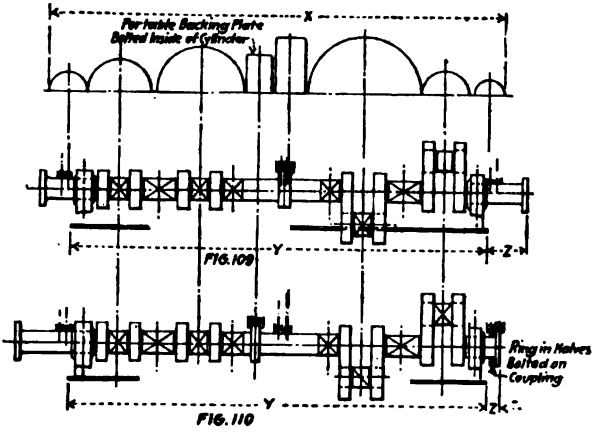
Fig. 108.

Block out the L.P. and 2d I.P. cylinders, valve chests and slide valves and enough of a valve diagram, Fig. 108, to determine the stroke of the eccentrics and the areas of the passages necessary in the cylinders and slide valves to give the proper steam and exhaust velocities. This is done to determine the shortest distance between centre-lines of cylinders and valve stems. Lay down the crankshaft and main bearings, Fig. 108, allowing $\frac{1}{8}$-in. clearance between the sides of the crank webs and the crank pin and main bearing brasses. The distance between main bearings Nos. (2) and (3) and Nos. (6) and (7) should be made as small as possible, clearance only being allowed for the coupling bolt nuts. The distance "a" is settled by allowing $\frac{1}{4}$-in. clearance between the after main bearing brasses and the 1st I.P. eccentrics and sufficient space for the coupling bolt nuts. The outline of the cylinder and valve chest covers should next be completed, Fig. 108, facing off the covers and offsetting the ahead eccentric rod about $\frac{1}{2}$ in. if necessary to close up the centres. The width of the main bearing girders is determined by allowing the brasses to overhang about $1\frac{3}{4}$ in. at the sides to allow for fitting water service pipes.

Before advancing further it is well to look into different arrangements of valves and crankshaft. With the same order of cylinders, the variation in

the dimensions *X*, *Y* and *Z* for different arrangements of valves and crankshaft may be noted in the following table, which is made up for an engine with cylinders 26 in., 38 in., 55 in., 81 in., by 51 in. stroke.

	X	*Y*	*Z*	
Fig. 108	31 ft. 10¼ in.	29 ft.–0¾ in.	19¼ in.	Crankshaft in four interchangeable sections.
Fig. 109	28 ft. 11¾ in.	27 ft. 2¼ in.	2 ft. 7½ in.	Crankshaft in two interchangeable sections.
Fig. 109	28 ft. 11¾ in.	27 ft. 2¼ in.	11½ in.	Crankshaft in two interchangeable sections. Forward end of shaft cut off (one spare section only required).
Fig. 110	28 ft. 11¾ in.	27 ft. 2¼ in.	8 in.	Crankshaft in two interchangeable sections.



After deciding on a suitable arrangement the end view is worked out Fig. 111. Draw the crank circle and the connecting rod on the top centre. Sweep the connecting rod on the crank circle a full stroke and spot the clearance line, being sure that the line covers all projecting parts such as set screws. This line should clear the bedplate and housings by about 2½ in. or 3 in. Next work out the conditions surrounding the upper end of the connecting rod including the crosshead, piston rod stuffing boxes, pistons, and bottom and top of H.P. and L.P. cylinders. Sufficient space should be allowed for the piston rod packing and gland to accommodate any type of metallic packing commonly used as the type of packing is often changed.

The length of the eccentric rod is determined by laying down the valve in its bottom position and locating the bottom valve chest cover with its stuffing box and gland. The clearance between the gland and the guide face on the

valve stem should be sufficient to allow repacking the gland with the valve in its top position. The length of the guide face should be such as to allow the valve stem facing to overrun at top and bottom in maximum linked up gear. The valve stem guide and the main links in their top position may then be located allowing ample space between the links and the guide bracket for oil cups on the link block and eccentric rod. The length of eccentric rod is then the distance between the centre of the links and the top of the eccentric circle drawn on the crank centre.

To locate the reverse shaft centre, lengths of reversing engine arm and reversing arms, angle of slot in reversing arms and outer end of the suspension rod, it is necessary to plot the paths of the eccentric rod pins on the links for full gear ahead and astern and maximum linked up astern positions. The motion of the main links is determined by the suspension rod and the two eccentric rods. The point where the suspension rod attaches to the link, moves in a circular arc with a radius equal to the length of the rod. The point where the astern eccentric rod is attached to the link has a compound motion due to the ahead eccentric rod and the suspension rod which causes the point to travel through a figure eight. This figure should be plotted for the link positions mentioned above and the reverse shaft, reversing arms and outer end of suspension rod located to give the smallest possible slotting action of the link block. It is also advisable to check the clearances between the link block and the connecting pins at the ends of the links for the corresponding link positions. Fig. 68 shows the slotting action of the link block for full gear ahead, linked up ahead, full gear astern and linked up astern positions.

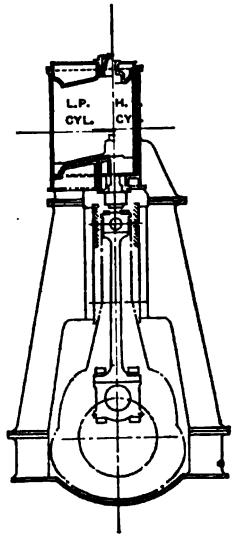


FIG. 111.

BALANCING MARINE ENGINES

The following text on "*Balancing*" was prepared by *Mr J. B. Woodward, Jr., of the Engineering Department of the Newport News Shipbuilding and Dry Dock Company:*

The object in balancing marine engines is to avoid vibrations in the ship's structure. The tremors produced by pounding in the engine due to slack bearings or improper valve setting are also objectionable but should not be confused with the vibrations caused by the unbalanced forces. The **magnitude of the unbalanced forces is dependent upon** the weight and size of the moving parts and the speed of rotation of the engine, but **is independent of** the steam pressure. When the piston with its attached weights is being accelerated, that portion of the steam pressure necessary to accelerate it is absorbed in giving energy to the moving parts and is not transmitted to the engine framing. It follows that the pressure on the cylinder head is only partially balanced by that portion of the steam pressure upon the piston which is transmitted to the engine framing by way of the piston rod, connecting rod, and crank shaft. It is obvious then that the unbalanced forces developed when an engine is revolving are equal and opposite to the forces necessary to accelerate or retard the moving parts.

Analytical Method

The most serious unbalanced forces are those which act in a vertical direction. The horizontal forces in a vertical engine are much smaller, also the ship itself is usually stiffer in the horizontal plane. In determining forces the moving parts are divided into two groups, reciprocating and rotating, and the cylinder and valve gear parts are treated separately, as follows:

1. **Cylinder Reciprocating Parts:** Pistons, piston rods, crossheads and top half of connecting rods. The connecting rod is divided into two parts at the centre of gravity, one assumed to move with the crosshead and the other to revolve with the crank pin. The location of the centre of gravity of the connecting rod may be assumed with sufficient accuracy.

2. **Valve Gear Reciprocating Parts:** Valves, valve stems, balance pistons, links and blocks, ahead eccentric rods, one-half the astern eccentric rods and one-half the suspension rods.

3. **Cylinder Rotating Parts:** Crank webs, crank pins, and bottom half of connecting rods.

4. **Valve Gear Rotating Parts:** Ahead and astern eccentrics and straps.

Formulæ for Vertical Forces. The rotating parts of an engine give rise to a force which is constant in amount but variable in direction while the reciprocating parts set up a force constant in direction but variable in amount. In dealing with the rotating parts only the vertical component of the force will be considered.

Assume A = length of crank arm, B = length of connecting rod, $U = B + A$, b = length of eccentric rod, a = throw of eccentric, $u = b + a$, l = sum of the mean of top and bottom steam laps and the mean of top and bottom steam leads, N = revolutions per sec., and θ = angle of the crank from the vertical axis. All dimensions are in ft. Then for a single cylinder engine, or for each cylinder of a multi-cylinder engine:

$$\text{Cylinder Reciprocating Parts: } F_1 = \frac{4\pi^2}{32.2} N^2 A W_1 \left(\cos\theta + \frac{1}{U}\cos 2\theta\right)$$

$$= 1.226 N^2 A W_1 \left(\cos\theta + \frac{1}{U}\cos 2\theta\right) \quad (1)$$

where W_1 is the weight of the parts.

$$\text{Valve Gear Reciprocating Parts: } F_2 = 1.226 N^2 a W_2 \cos\theta \quad (2)$$

where W_2 is the weight of the parts. The second term $\frac{1}{u}\cos 2\theta$ has been neglected as the weight of the valve gear is small compared with the cylinder parts and as the ratio of eccentric rod length to eccentricity is relatively large.

$$\text{Cylinder Rotating Parts: } F_3 = 1.226 N^2 A W_3 \cos\theta \quad (3)$$

where W_3 is the weight of the parts.

$$\text{Valve Gear Rotating Parts: } F_4 = 1.226 N^2 l W_4 \cos\theta \quad (4)$$

where W_4 is the weight of the parts.

Application of Formulæ. To show the application of the formulæ the vertical forces will be computed for a triple expansion engine of the usual marine type with cranks at 120 deg.

Example: Engine size is 20½ in.–35 in.–60 in. × 42 in. stroke with attached air and bilge pumps driven by a beam from the L.P. crosshead. The order of cylinders from the forward end is H.P., I.P., and L.P. Designed I.H.P. is 1500 at 85 R.P.M. As the crank webs, crank pins and connecting rods are the same for the three cylinders there is no unbalanced force arising from the cylinder rotating parts. When these parts are

not of the same size, the unbalanced force for each crank may be calculated by formula 3. Except for the pump lever pins on the L.P. crosshead (which are included with the pump gear in Table 18) the piston rods and crossheads are also alike for the three cylinders. Their resultant force for this crank arrangement is zero and they may be omitted from the computation. The connecting rods are omitted for the same reason. The L.P. eccentrics are on the shaft while the other eccentrics are on the shaft couplings. The data required is given in Table 17.

Table 17

Designed R.P.S. (N) (85 ÷ 60)	1.42
Length of Crank Arm (A), ft	1.75
Length of Connecting Rod (B), ft	7.50
Connecting Rod Ratio (U)	4.28
1.226 N^2A	4.31

Size and type valve	H.P. 8-in. piston	I.P. 14-in. piston	L.P. double-ported slide
Valve takes steam	Inside	Inside	Outside
Throw of Eccentric (a)—ft	0.33	0.33	0.27
Angle of advance	36°-38′	28°-22′	42°-30′
Sum of mean steam lap and mean steam lead (l)—ft	0.20	0.21	0.18
Weight in lbs.			
Piston	560	1400	2850
Valve gear reciprocating parts	930	1200	2500
Valve gear rotating parts	2100	2100	1200

The attached pumps, Fig. 112, may be combined with the cylinder reciprocating parts

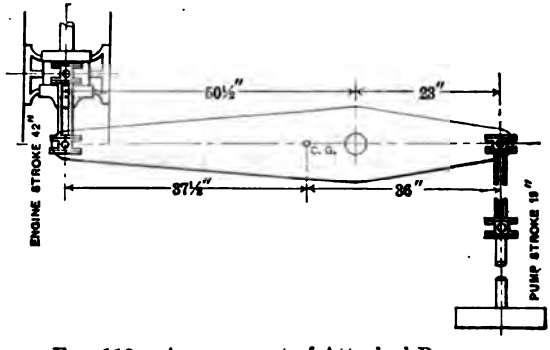


FIG. 112.—Arrangement of Attached Pumps.

by determining the force which would have to be applied at the L.P. crosshead to balance the combined moment of the whole gear, Table 18. The obliquity of the links may be neglected as its effect is small.

Table 18

Item	Weight, lbs.	Arm about beam axis, in.	Moment about beam axis, in.-lbs.
Pump plungers, bucket and rod	350	23.0	8050
Pump crosshead	250	23.0	5750
Pump links and link pins	230	23.0	5290
Crosshead links and link pins	150	−50.5	−7570
Crosshead pins	20	−50.5	−1010
Pump beam	405	−13.0	−5265
Resultant	1405	+3.73	5245

A weight of 104 (= 5245 + 50.5) lbs. applied at the crosshead end would completely balance 1405 lbs. with a moment of 5245 in.-lbs. The effect of the attached pumps is then virtually to reduce the L.P. cylinder reciprocating weight by 104 lbs. It is better however, to treat this weight separately as in Table 19 rather than to combine it with cylinder weight.

The arrangement of the cranks and valve gear is shown diagrammatically in Fig. 113. The position shown for the eccentric in each case is the mean position of the ahead and astern eccentrics.

Calling the angle of the H.P. crank θ, the formulæ become:

Cylinder Reciprocating Parts:

$$\text{H.P.}, F_1 = 2420\left(\cos\theta + \frac{\cos 2\theta}{4.28}\right)$$

$$\text{I.P.}, F_1 = 6040\left[\cos(\theta - 120°) + \frac{\cos 2(\theta - 120°)}{4.28}\right]$$

$$\text{L.P.}, F_1 = 12{,}300\left[\cos(\theta + 120°) + \frac{\cos 2(\theta + 120°)}{4.28}\right]$$

$$\text{Attached pumps}, F_1 = 450\left[\cos(\theta - 60°) + \frac{\cos 2(\theta - 60°)}{4.28}\right]$$

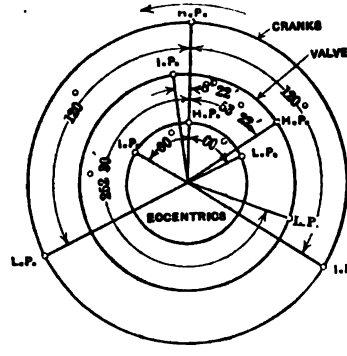


FIG. 113.—Arrangement of Cranks and Eccentrics Looking Aft.

Valve Gear Parts:	H.P.	I.P.	L.P.
Angle of valve from H.P. crank =	306° − 38′	8° − 22′	252° − 30′
$1.226\,N^2a\,W_2$ =	762	985	1670
$1.226\,N^2W_4$ =	1035	1090	535

Then the resultant force of the valve gear *reciprocating* parts, $F_3 = 762\cos(\theta + 306° - 38') + 985\cos(\theta + 8° - 22') + 1670\cos(\theta + 252° - 30') = (762\cos 306° - 38' + 985\cos 8° - 22' + 1670\cos 252° - 30')\cos\theta - (762\sin 306° - 38' + 985\sin 8° - 22' + 1670\sin 252° - 30')\sin\theta = 925\cos\theta + 2060\sin\theta$. The resultant force of the valve gear *rotating* parts, $F_4 = 1035\cos\theta + 1090\cos(\theta + 60°) + 535\cos(\theta - 60°) = 1847\cos\theta - 482\sin\theta$.

Table 19. Vertical Forces

H.P. crank angle	Cylinder reciprocating parts						Valve gear			Total resultant
	H.P.	L.P.	I.P.	Cylinder resultant	Attached pump	Resultant including attached pumps	2772 × cos θ	1578 × sin θ	Resultant	
0	2,985	− 7,600	−3,730	−8,345	172	−8,173	2,772	0	2,772	−5401
20	2,710	− 8,920	−2,370	−8,580	362	−8,218	2,605	540	3,145	−5,073
40	1,950	− 9,370	− 272	−7,692	502	−7,190	2,125	1,012	3,137	−4,053
60	928	− 9,430	2,310	−6,192	553	−5,639	1,385	1,365	2,750	−2,889
80	− 109	− 9,370	4,875	−4,604	502	−4,102	482	1,555	2,037	−2,065
100	− 952	− 8,920	6,760	−3,112	362	−2,750	− 482	1,555	1,073	−1,677
120	−1,495	− 7,600	7,450	−1,645	172	−1,473	−1,385	1,365	− 20	−1,493
140	−1,755	− 4,830	6,760	175	− 20	155	−2,125	1,012	−1,113	− 958
160	−1,840	− 553	4,875	2,482	−176	2,306	−2,605	540	−2,065	241
180	−1,855	4,720	2,310	5,175	−276	4,899	−2,772	0	−2,772	2,127
200	−1,840	9,930	− 372	7,818	−325	7,493	−2,605	− 540	−3,145	4,348
220	−1,755	13,780	−2,370	9,655	−341	9,314	−2,125	−1,012	−3,237	6,077
240	−1,495	15,160	−3,730	9,935	−344	9,591	−1,385	−1,365	−2,750	6,841
260	− 952	13,780	−4,380	8,448	−341	8,107	− 482	−1,555	−2,037	6,070
280	109	9,930	−4,600	5,221	−325	4,896	482	−1,555	−1,073	3,823
300	928	4,720	−4,640	1,008	−276	732	1,385	−1,365	20	752
320	1,950	− 553	−4,600	−3,203	−176	−3,379	2,125	−1,012	1,113	−2,266
340	2,710	− 4,830	−4,830	−6,500	− 20	−6,520	2,605	− 540	2,065	−4,455
360	2,985	− 7,600	−3,730	−8,345	172	−8,173	2,772	0	2,772	−5,401

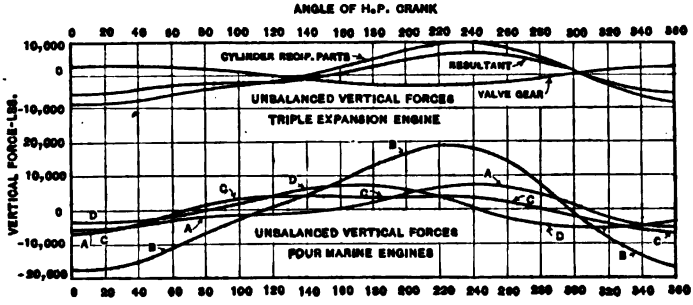


FIG. 114.—Curves of Unbalanced Vertical Forces.

The total force of the valve gear is then = 2772 cos θ + 1578 sin θ.

The vertical forces of the cylinder parts and the valve gear and the resultant vertical force are given in Table 19 and are shown graphically on a crank angle base in Fig. 114. The resultant vertical forces for the four engines below are shown in Fig. 114.

Curve	Size in.	R.P.M.	I.H.P.	Order of cylinders	Sequence of cranks
A	20½–35–60×42 stroke	85	1500	H.P., I.P., L.P.	H.P., I.P., L.P.
B	31–50–84×54 stroke	82	4000	H.P., I.P., L.P.	H.P., I.P., L.P.
C	35–58–69–69×60 stroke	75	5500	F.L.P., H.P., I.P., A.L.P.	F.L.P., A.L.P., H.P., I.P.
D	24–35–51–75×51 stroke	70	2600	H.P., L.P., 2 I.P., 1 I.P.	H.P., 1 I.P., L.P., 2 I.P.

In all cases the cranks are equally spaced and the cylinder rotating parts are balanced in all cases except *B*, where the crank webs are of unequal thickness. In large three cylinder triple-expansion engines (*B*) the vertical forces are necessarily great unless a counterweight is used opposite the l.-p. crank to offset the effect of the heavy piston, which however is not often done. No counterweights have been used in any of the above engines.

Rocking Moments. As the unbalanced forces of the various cylinders act at different points along the axis of the engine, unbalanced moments arise which tend to tilt, or rock, the engine in a fore and aft direction. It is convenient to refer these moments to a horizontal axis at right angles to the axis of the crank shaft and passing through the intersection of the crank shaft axis with the centre line of the forward cylinder. The arms are then measured from the point of intersection to the axis of the various cylinders and valves. Where there is a valve forward of the point of intersection, its arm is considered negative. The formulæ for the rocking moments may be derived from formulæ (1), (2), (3) and (4) by inserting a factor in each to represent the arm about the given axis expressed in feet.

Primary and Secondary Forces and Moments. It is useful, particularly in representing the forces graphically, to separate the force due to the reciprocating members into two parts, one part varying as cos θ and called the **primary force**, and the other varying as cos 2θ and called the **secondary force.** Corresponding to the primary and secondary forces there are also primary and secondary moments. The weights for the secondary forces, as

may be seen in formula (1) are only $\frac{1}{U}$ times the weights for the primary forces. However, it should be borne in mind that the forces or moments thus separated for convenience into primary and secondary parts actually appear combined as a single force or moment.

Graphical Method

The method given here is based on Rear-Admiral *D. W. Taylor's* paper *"Balancing Marine Engines"* (Trans. Soc. of Nav. Arch. and Mar. Eng., Vol. 9).

Polygons are drawn for the primary and for the secondary forces due to the reciprocating parts, and for the forces due to the rotating parts. The corre-

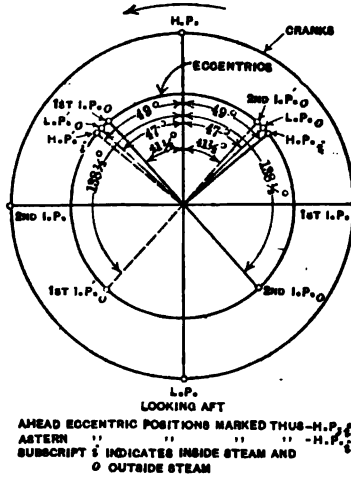


FIG. 115.—Arrangement of Cranks and Eccentrics.

sponding moment polygons are also drawn. In order to combine the valve gear in the same polygon with the main gear, the valve weights are reduced to virtual weights, having the travel of the piston for reciprocating weights and revolving with the crank radius for rotating weights. The reduction factor for the primary polygons is the ratio of valve travel to engine stroke $(a \div A)$; and for the secondary polygons, the factor is the product of the ratio of valve travel to stroke $(a \div A)$ by the ratio of the throw of the eccentric to the length of the eccentric rod $(a \div b)$.

The valve gear is here separated into (1) ahead reciprocating parts, consisting of valves, valve stems, balance pistons, link blocks, and one-half the links, suspension rods and ahead eccentric rods; (2) astern reciprocating parts, consisting of one-half the links and astern eccentric rods; (3) ahead rotating parts, consisting of the ahead eccentrics and straps and one-half the ahead

eccentric rods; and (4) astern rotating parts, consisting of astern eccentrics and straps and one-half the astern eccentric rods.

Example. The method will be applied to a quadruple expansion engine 26 in.–38 in.–55 in.–81 in. by 51 in. stroke designed for 3000 I.H.P. at 70 R.P.M. The particulars of the engine and the data required to construct the polygons are given in Tables 20 and 21. The arrangement of cranks and eccentrics is shown in Fig. 115 and the arrangement of cylinders and valves in Fig. 116. The air, bilge and sanitary pumps attached to the L.P. crosshead have an upward resultant of 560 lbs., which is subtracted from the L.P. reciprocating weights.

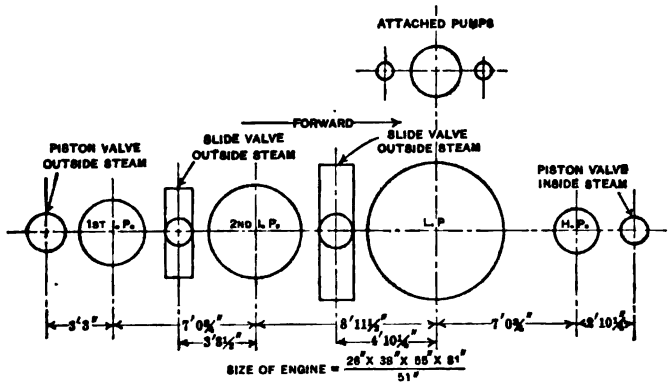


FIG. 116.—Arrangement of Cylinders and Valves.

Table 20.

Designed R.P.S. (N) (70 ÷ 60)	1.167
Length of Crank Arm (A), ft	2.125
Length of Connecting Rod (B), ft	9.580
Connecting Rod Ratio (U)	4.500
Length of Eccentric Rods (b), ft	9.250
$1.226N^2A$	3.540

	H.P.	1st I.P.	2d I.P.	L.P.
Travel of valves, in	8.0	8½	8½	9.0
Primary reduction factor for valve gear weights	0.1570	0.1670	0.1670	0.1760
Secondary reduction factor for valve gear weights	0.0057	0.0064	0.0064	0.0071

Table 21. Weights and Moments

(Moments are expressed in foot-pounds)

	Rotating				Reciprocating			
	H.P.	1st. I.P.	2d I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.
Actual weights:								
Cylinder parts	10,400	10,400	11,000	11,000	5,500	6,500	7,000	8,980
Valve gear parts ahead	1,580	1,580	2,220	2,220	1,540	2,230	4,230	5,580
Valve gear parts astern	1,600	1,600	2,250	1,250	500	500	500	500
Virtual primary weights:								
Valve gear parts ahead	250	260	370	390	240	370	710	980
Valve gear parts astern	250	270	380	400	80	80	80	90
Virtual secondary weights:								
Valve gear parts ahead	9	10	14	16	9	14	27	40
Valve gear parts astern	9	10	14	16	3	3	3	4
Moment arms about H.P.:								
Cylinders	0	23.08	16.02	7.06	0	23.08	16.02	7.06
Valves	−2.85	26.33	19.73	11.92	−2.85	26.33	19.73	11.92
Moments about H.P.:								
Cylinder parts	0	240,080	176,000	78,000	0	150,000	112,000	63,000
Valve gear parts ahead	−700	6,800	7,300	4,700	−700	9,700	14,000	11,700
Valve gear parts astern	−700	7,100	7,500	4,800	−200	2,100	1,600	1,100
Moment arms about 1st I.P.:								
Cylinders	23.08	0	7.06	16.02	23.08	0	7.06	16.02
Valves	25.93	−3.23	3.35	11.16	25.93	−3.23	3.35	11.16
Moments about 1st I.P.								
Cylinder parts	240,000	0	77,500	176,000	127,000	0	49,500	142,500
Valve gear parts ahead	6,500	−800	1,200	4,400	6,200	−1,200	2,400	11,000
Valve gear parts astern	6,500	−900	1,300	4,500	2,100	−300	300	1,000

Primary Force Polygon, Fig. 117. From O as a centre to a convenient scale construct the polygon $OABGI$ of which each side represents by its length the weight, Table 21, of the cylinder reciprocating parts for one of the four cranks and parallels the crank position. The letter at the end of each line is the same as that marking the corresponding crank position. In similar manner construct the polygon $OBCDFHJKL$ for the valve gear reciprocating parts, the sides being equal in length to the virtual primary weights, Table 21, of the various valve gear reciprocating parts and parallel to the corresponding eccentric positions. On account of the relatively small weight of the valve gear it is convenient to use a larger scale for the valve gear polygons. The closing line LO to the scale of the main polygon $OABGI$ reduces to MO. IO and MO are combined by means of the parallelogram $OMNI$ into NO, which then represents the weight and position of an imaginary weight which revolving on the crank radius would completely balance the primary forces of the main and valve gear reciprocating parts. To obtain the absolute value of the vertical force, ON is multiplied in length by $1.226N^2A$ ($=3.54$) and the new line OP laid off to the scale which it is desired to use when plotting the forces on a crank angle base, Fig. 122. The projection OQ of OP on the vertical axis represents the magnitude of the vertical force due to the reciprocating parts when the H.P. crank is passing the top centre. By revolving OP about O and finding its projection on the vertical axis

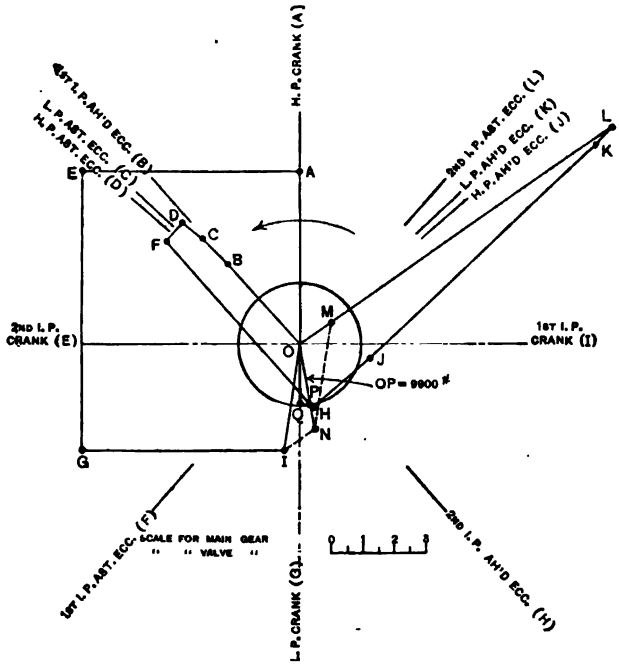

H. P. CRANK (A)
1ST I. P. AH'D ECC. (B)
L. P. AST. ECC. (C)
H. P. AST. ECC. (D)
2ND I. P. AST. ECC. (L)
L. P. AH'D ECC. (K)
H. P. AH'D ECC. (J)
2ND I. P. CRANK (E)
1ST I. P. CRANK (I)
OP = 9900 #
1ST I. P. AST. ECC. (F)
SCALE FOR MAIN GEAR
" " VALVE "
2ND I. P. AH'D ECC. (H)
L. P. CRANK (G)

FIG. 117.—Primary Force Polygon.

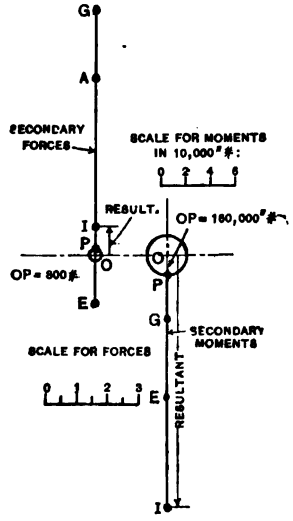


FIG. 118.—Secondary Force and Moment Polygons.

for any position, the vertical force of the reciprocating parts may be found for the corresponding H.P. crank angle. Thus if it is desired to find the force when the H.P. crank has moved 20 deg. from the vertical position it is only necessary to revolve *OP* 20 deg. in the direction of rotation and obtain its projection on the vertical axis.

The Secondary Force Polygon, Fig. 118, is constructed in a manner similar to the primary polygon except that each weight is laid off in a position corresponding to double the angle in Fig. 115. As a rule the virtual secondary weights of the valve gear may be neglected as is done in this case. The closing line (*IO*) of the polygon is multiplied by $1.226N^2A + (B/A) = 0.79$ and laid off (*OP*) to the scale of forces in Fig. 122. In finding the vertical forces *OP* is revolved at twice the rate of the H.P. crank and projected on the vertical axis.

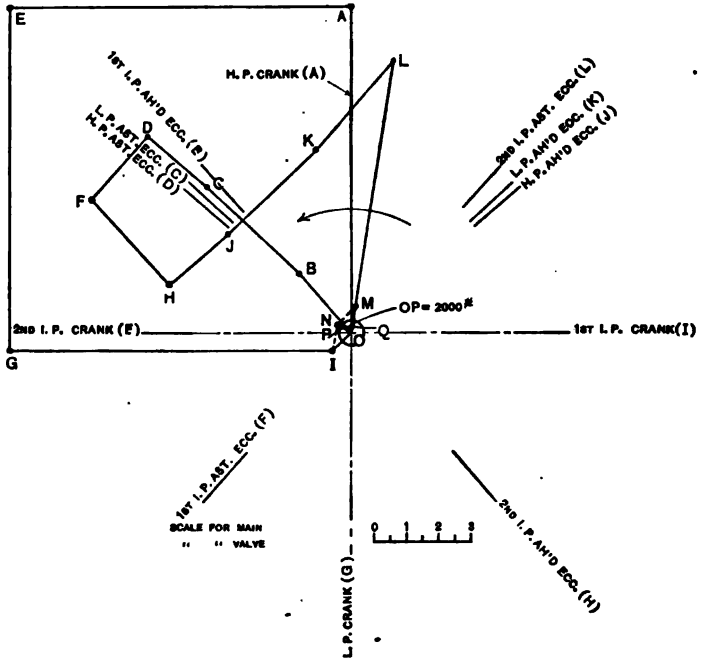


FIG. 119.—Rotating Force Polygon.

The Rotating Force Polygon, Fig. 119, is drawn in a manner similar to the primary polygon for the reciprocating parts and the same factor 3.54 is applied to the closing line. The horizontal forces if desired, may readily be obtained by projecting *OP* on the horizontal axis.

Moment Polygons, Figs. 118, 120 and 121, are plotted in the same manner as the force polygons, using moments instead of weights, and applying the same factors to the closing lines as in the force polygons.

Force and Moment Curves, Fig. 122. The reciprocating and rotating forces are plotted on a crank angle base and combined into a single resultant vertical force. The

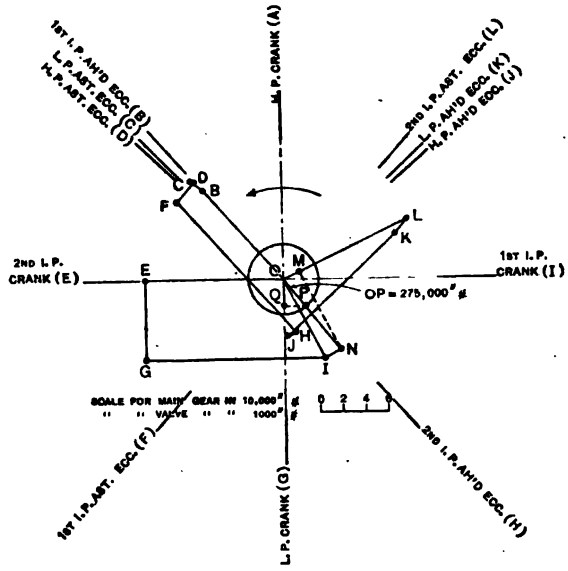

H. P. CRANK (A)
1ST I. P. ANT'D ECC. (B)
L. P. AST. ECC. (C)
H. P. AST. ECC. (D)
2ND I. P. AST. ECC. (L)
L. P. ANT'D ECC. (K)
H. P. ANT'D ECC. (J)
2ND I. P. CRANK (E)
1ST I. P. CRANK (I)
OP = 275,000 #
1ST I. P. AST. ECC. (F)
2ND I. P. ANT'D ECC. (H)
L. P. CRANK (G)
SCALE FOR MAIN GEAR IN 10,000 #
" " VALVE " " 1000 #
0 2 4 6

FIG. 120.—Primary Moment Polygon.

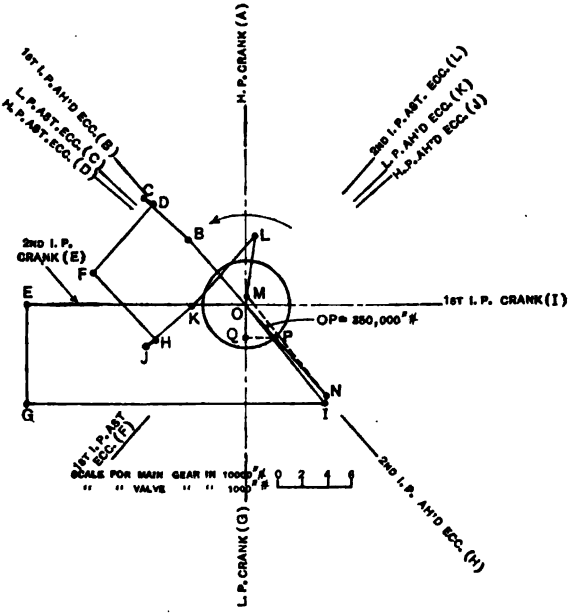

H. P. CRANK (A)
1ST I. P. ANT'D ECC. (B)
L. P. AST. ECC. (C)
H. P. AST. ECC. (D)
2ND I. P. AST. ECC. (L)
L. P. ANT'D ECC. (K)
H. P. ANT'D ECC. (J)
2ND I. P. CRANK (E)
1ST I. P. CRANK (I)
OP = 350,000 #
1ST I. P. AST. ECC. (F)
2ND I. P. ANT'D ECC. (H)
L. P. CRANK (G)
SCALE FOR MAIN GEAR IN 10000 #
" " VALVE " " 1000 #
0 2 4 6

FIG. 121.—Rotating Moment Polygon.

moments about the forward crank may be plotted and combined in the same manner. The resultant moment only is shown in Fig. 122. The resultant moment about the after crank is also shown.

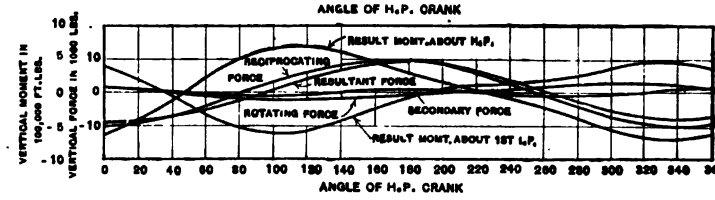


FIG. 122.—Unbalanced Vertical Forces and Rocking Moments, Quadruple Expansion Engine.

Means of Improving Balance

In a single-crank engine the unbalanced force of the rotating parts can be compensated for by a counterweight on the crank equal in weight to the rotating parts. If the counterweight be made equal to the rotating and reciprocating parts, the unbalanced force of the latter also disappears, but a new horizontal force is introduced equal to the radial acceleration of the reciprocating parts.

In a two-crank engine revolving balance may be secured by the use of counterweights opposite each crank, and if balance of the reciprocating parts is necessary, it is probably best to use lever balance weights for each cylinder. Perfect balance may be secured only by special arrangements in which the cranks are no longer at 90 deg.

The arrangement of cylinders generally used **in three-crank triple-expansion engines** counting from the forward end is H.P., I.P. and L.P. with the cranks at 120 deg. The crank pins and webs and also the piston rods, crossheads, and connecting rods are usually made the same for all three cylinders. With such an arrangement the balance may be improved by increasing the weight of the H.P. piston and lightening the L.P. so as to make the weights of the reciprocating parts as nearly equal as possible. Even when the crank webs are of unequal thickness it is seldom advisable to add weight and complication to the engine by using counterweights to balance the rotating masses.

In four-crank engines *Bauer and Robertson* state that balancing is most effectually carried out when (1) the cylinders with the heaviest connecting rods are in the centre, and (2) the distance between the two middle cylinders is as large as possible, as compared with the distance between the two cylinders at either end of the engine. "If these conditions cannot be secured in practice, the outer connecting rods must be artificially lightened by means of counterweights and the connecting rods of the middle cylinders artificially weighted by the use of heavy pistons. In triple expansion engines good balancing is most easily obtained by having two L.P. cylinders, each doing purposely less work, and having lighter connecting rods than the H.P. and I.P. cylinders, outside which they are placed. An engine with four cranks, in which the forward pair of cranks is at right angles to the after pair, and the cranks of each pair are at an angle of 180 deg. to each other, gives an almost perfect balance, if the distance between the two pairs of cranks is relatively large in proportion to the distance between the individual cranks constituting each

pair, and when the masses of the forward pair are equal to the masses of the after pair." Four-cylinder quadruple expansion engines should be arranged with the H.P. and 1st I.P. cylinders on the ends and the 2d I.P. and L.P. inside, with the attached pumps, if any, driven by a beam from the L.P. crosshead. The L.P. piston should be made as light as possible. The pair of cylinders on each end should be as close together as practicable, which is facilitated by placing the valves for the H.P. and the 1st I.P. cylinders at the ends of the engine and the valves for the other two cylinders in the middle.

Revolving balance may be obtained in four-crank engines in a number of ways, simplicity and cheapness usually being the determining factors. Instead of fitting balance weights opposite each crank two revolving counterweights are often used, one forward of the engine framing and the other aft. The turning wheel may generally be used for the attachment of one weight, and the other may be made in the form of a disk attached to the forward end of the crankshaft, extended a few inches if necessary. The amount and angular position of such weights may be readily determined from the force and moment polygons. Whether or not counterweights are used it is well to consider the advantage to be gained by drilling holes in one or more crank pins if solid, or filling one or more with lead if hollow.

Yarrow-Schlick-Tweedy System. Complete primary balance can be obtained in four-crank engines on the assumption of infinite connecting rods, thus eliminating the secondary forces from consideration. The balance is complete if: (1) the sum of the moments of the horizontal forces and the sum of the moments of the vertical forces about the forward crank = 0; and (2) the sum of the moments of the horizontal forces and the sum of the moments of the vertical forces about the after crank = 0. By proper choice of crank angles, and by changing weights of moving parts and cylinder spacing, in so far as the design of the engine permits, it is possible to approximate very closely the above conditions.

The work is best done graphically. The vertical moment about the forward crank is found for each cylinder and valve, using for the cylinder weights the sum of the actual weights of the cylinder reciprocating and rotating parts, and for the valve gear weights the sum of the virtual weights of the valve gear reciprocating and rotating parts. The horizontal moment about the forward crank is found for each cylinder and valve, using the actual weights of the cylinder rotating parts and the virtual weights of the valve gear rotating parts. The moments about the after crank are obtained in the same manner. The four moment polygons are then plotted by laying off each moment parallel to its corresponding crank or eccentric position. (See Graphical Method.) The sum of the moments represented by a particular polygon is zero when the polygon is a closed figure. All the polygons may usually be made very nearly or entirely to close by properly choosing the angles, weights, and cylinder spacing. If necessary, counterweights may be used at the forward and after ends of the engine, the amount and position of such weight being determined in each case from the moment polygons. If the ratio $L \div l$, in which L is the length between centre lines of the pair of cylinders at either end of the engine and l the distance between centre lines of the inner pair of cylinders, is less than $\frac{4}{5}$ the angles of the two outer cranks are apt to be small, which has an injurious effect upon the maneuvering qualities and the regularity of the turning moment. If the cylinders with the heaviest moving parts are placed in the middle and $L \div l$ made about 2, the best crank angles will be obtained, the smallest being about 65 deg. to 70 deg., and complete balance can be obtained without the use of counterweights.

A full description of the Yarrow-Schlick-Tweedy system of balancing may be found in *Bauer and Robertson.*

Critical Revolutions and Vibrations of Hull. It is important that the revolutions per minute of the engine at its working speed do not coincide with the natural period of vibration of the ship. In cases where heavy vibrations have been set up they have been relieved by changing the working revolutions of the engine. The number of revolutions which produces the maximum vibratory effect is given by *Bauer and Robertson* from the formula derived by Schlick (*Trans. of the Inst. of Naval Architects*, 1894) as follows:

$$N = K\sqrt{T \div DL^3}$$

in which N is the revolutions per min.; T, the moment of inertia of the midship section in foot units; D, the displacement in tons; L, the water line length in feet; and K, a constant having the following values:

For large fast ships with fine lines, $K = 1.872 \times 10^6$

For cargo vessels with full lines, $K = 1.668 \times 10^6$.

Investigations by *Mr. William Gatewood* on an oil tank steamer, *Trans. Soc. Nav. Arch. and Mar. Eng.*, 1915, gave results closely in agreement with the above formula.

TURNING EFFORT

While it is important that the resultant of the tangential forces acting upon the crankpins, frequently called the turning effort, be as uniform as possible throughout the revolution to insure steadiness of running and to reduce the stresses in the shafting, it is not as a rule necessary to construct the turning effort diagram, unless the design departs from usual practice, in which case it is advisable to construct a set of theoretical indicator cards, and proceed as outlined below. It is sufficient for the purpose of calculating shafting stresses to know the ratio of maximum to mean twisting moment, which has the same value as the ratio of maximum to mean turning effort. The value of this ratio is given for various types of engines under *Crankshafts*, p. 873. **Variation in the turning effort is due to** (1) unequal distribution of power in the different cylinders, (2) variation in steam pressure on the pistons, (3) weight and inertia of the moving parts, and (4) the angularity of the connecting rod. In general **the greater the number of cranks the more uniform the turning effort.**

The turning effort on any one crank in pounds multiplied by the crank radius in inches gives the twisting moment on the shaft in pound-inches for that particular crank. The same result is more conveniently obtained if the net vertical force on the piston due to steam pressure and to inertia and dead-weight of the moving parts is multiplied by OR, Fig. 123, the distance from the shaft centre O to the intersection of PQ, the centreline of the connecting rod, with the axis XOX. The turning effort may be readily obtained from the twisting moment by dividing by the crank radius, or from the twisting moment diagram, Fig. 123, by applying the proper scale.

For all practical purposes the indicator cards may be assumed to be taken on the same revolution of the engine. On the crank pin circle, Fig. 123, lay off the points 1′, 2′, 3′, 4′, etc. 30 deg. apart and locate the corresponding points 1, 2, 3, 4, etc. in the travel of the piston. Locate these piston positions for each set of cards, and at each position erect an ordinate. At any position of the downstroke the intersection of the ordinate between the downstroke line of the top card and downstroke line of the bottom card represents the difference in steam pressure on the top and the bottom of the piston; and in similar manner for any upstroke position. The ordinates

from the base line *AA* to *EFGH* show the pressure difference on the pistons in lbs. per sq. in. for a complete revolution, ordinates above *AA* representing downward and ordinates below, upward pressure.

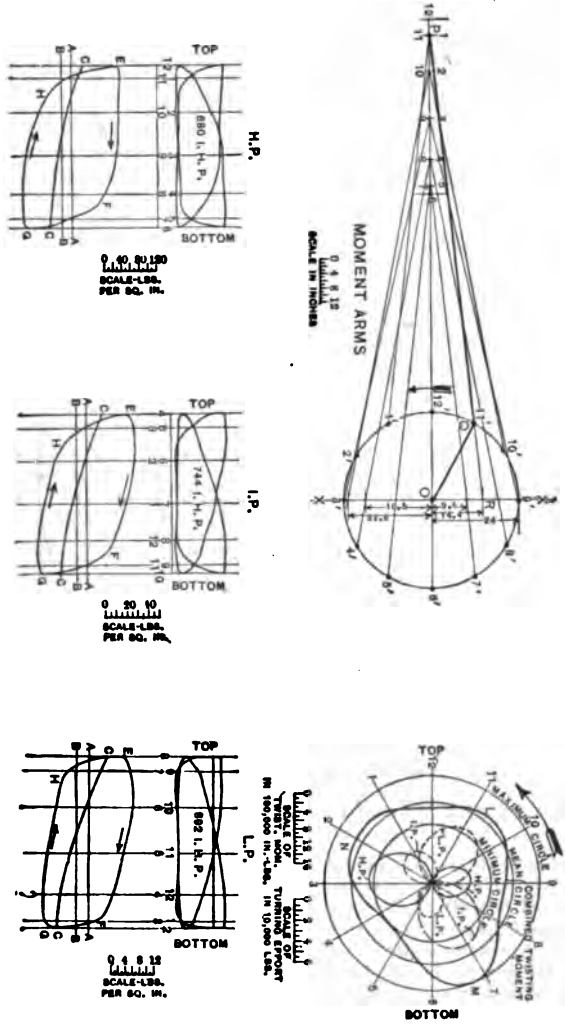


FIG. 123.—Construction of Twisting Moment Diagram for Triple Expansion Engine.

The moving parts are being accelerated during the early portion of the stroke and their inertia must be subtracted from the steam pressure on the

piston; during the later portion the parts are being retarded and their inertia must be added to the steam pressure. The **inertia force** (F) may be calculated by the formula $\mathbf{F = 1.226N^2AW\left(\cos\theta + \frac{\cos 2\theta}{U}\right)}$, in which N = revolutions per second, A = length of crank arm in ft., U = ratio of connecting rod length to crank arm, θ = angle of the crank, and W = weight in lbs. of the piston, piston rod, crosshead, and top half of connecting rod. Attached pumps should be treated as described under *Balancing*, p. 922. The vertical component of the centrifugal force of the rotating parts is not considered, as when combined with the horizontal component the resultant is a radial force. The friction, weight and inertia of the valve gear has little effect upon the turning effort and need not be considered.

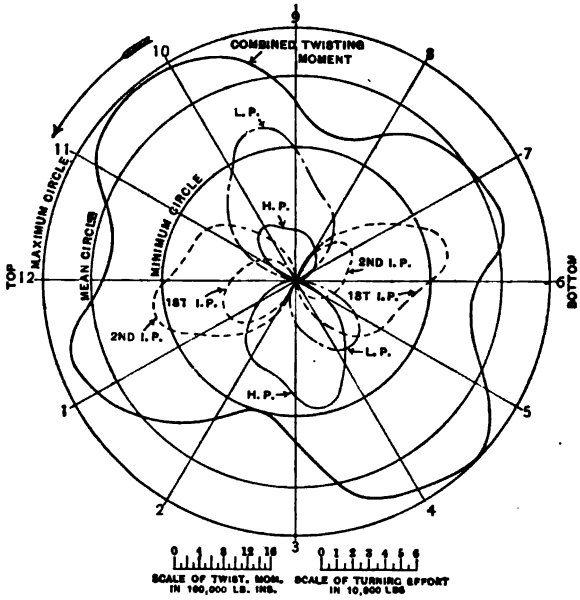


FIG. 124.—Twisting Moment Diagram for Quadruple Expansion Engine.

The dead-weight w, which includes all of W and in addition the bottom half of the connecting rod, the crank pin, and the eccentric portion of the webs, must be added directly to the pressure on the piston during the downstroke and subtracted during the upstroke. In Fig. 123 if BB be laid off from AA as a base to represent w per sq. in. of piston area and CC laid off from BB to represent F per sq. in. of piston area, then the ordinates between CC and $EFGH$ will represent the vertical pressure per sq. in. on the piston due to the steam and to the weight and inertia of the moving parts.

In Tables 22, 23 and 24 the data and calculations and in Fig. 123 the twisting moment diagram are shown for a 24½ in.-41½ in.-72 in. × 48 in.

stroke triple expansion engine. The length of the connecting rod is 9 ft. and the R.P.M. = 72. The twisting moment in lbs.-in. is plotted radially on the corresponding crank position for each crank separately and the three smaller figures then combined into the resultant *LMN*. The ratio of the maximum to the mean twisting moment as shown by the figure is 1.32. The mean twisting moment is 1,800,000 lbs.-in. against 1,850,000 lbs.-in. by the formula (63,000 I.H.P. + R.P.M.).

In Fig. 124 is shown the twisting moment diagram for a 27 in.-39 in.-58 in.-87 in. × 54 in. stroke quadruple expansion engine turning at 85 R.P.M. The ratio of maximum to mean twisting moment in this case is 1.233.

Table 22

Cylinder	H.P.	I.P.	L.P.
Area of cylinder	471.44	1352.7	4071.5
1.226 N^2A + cylinder area	0.0075	0.00261	0.00087
Dead-weight on piston (w)	10,350	11,750	13,400
Dead-weight + cyl. area	22.0	8.7	3.3
Weights for inertia forces (W)	4,450	6,950	7,500

Table 23. Inertia Forces

Crank position θ deg.	Cos θ + $\frac{\cos 2\theta}{U}$	Inertia force—lbs. per sq. in. cyl. area		
		H.P.	I.P.	L.P.
0	1.2222	40.5	22.1	7.9
30	0.9771	32.5	17.6	6.4
60	0.3889	13.0	7.0	2.5
90	−0.2222	− 7.4	− 4.0	−1.5
120	−0.6111	−20.4	−11.1	−4.0
150	−0.7549	−25.2	−13.7	−4.9
180	−0.7778	−25.9	−14.0	−5.1
210	−0.7549	−25.2	−13.7	−4.9
240	−0.6111	−20.4	−11.1	−4.0
270	−0.2222	− 7.4	− 4.0	−1.5
300	0.3889	13.0	7.0	2.5
330	0.9771	32.5	17.6	6.4
360	1.2222	40.5	22.1	7.9

Table 24. Twisting Moment

Crank position	Arms—Fig. 123			Press. in lbs. per sq. in. from Fig. 123*			Twisting moment pound-inches			
	H.P.	I.P.	L.P.	H.P.	I.P.	L.P.	H.P.	I.P.	L.P.	Total
1	14.4	24.0	9.6	98	25	11.5	665,000	810,000	449,000	1,924,000
2	22.8	22.8	0.0	115	26	− 2.0	1,236,000	802,000	0	2,038,000
3	24.0	14.4	9.6	128	13	3.7	1,450,000	253,000	145,000	1,848,000
4	18.5	0.0	18.5	118	−23	5.2	1,030,000	0	391,000	1,421,000
5	9.6	14.4	24.0	40	34	6.5	181,000	662,000	635,000	1,478,000
6	0.0	22.8	22.8	56	45	8.5	0	1,385,000	788,000	2,173,000
7	9.6	24.0	14.4	66	50	9.3	298,000	1,623,000	545,000	2,466,000
8	18.5	18.5	0.0	72	46	− 4.5	628,000	1,150,000	0	1,778,000
9	24.0	9.6	14.4	80	23	8.5	906,000	298,000	498,000	1,702,000
10	22.8	0.0	22.8	70	−16	13.0	752,000	0	1,205,000	1,957,000
11	14.4	9.6	24.0	34	22	14.8	231,000	285,000	1,445,000	1,961,000
12	0.0	18.5	18.5	80	25	14.5	0	625,000	1,090,000	1,715,000

* Including steam pressure, inertia force, and dead-weight.

Revolutions for Different Percentages of Full Power. The revolutions of reciprocating engines corresponding to different indicated horsepowers may be taken from the curve in Fig. 125 which is plotted from data from a number of engines. If the revolutions for full power are known the revolutions for other powers will agree closely with values taken from the curve.

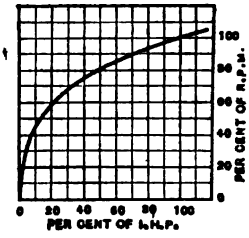


FIG. 125.—Relation of I.H.P. and R.P.M.

Backing Power of Marine Engines. In reversing a reciprocating engine a much greater proportion of ahead full power can be obtained than in a turbine installation in which the backing turbine is a separate element, the power of which is kept as low as consistent with safety in handling the ship in order to reduce the turbine size. *Commander S. M. Robinson* in the Jour. *A.S.N.E.* for February, 1916, gives the results of the backing trials of the Battleship "Delaware" when suddenly reversed from full speed ahead. At the beginning, 44 per cent of full ahead power (28,500 I.H.P.) was developed at 40 per cent of ahead revolutions (130 R.P.M.), which had increased to 55 per cent at 54 per cent of revolutions when the vessel stopped. The maximum power developed as the backing continued was 81 per cent at 74 per cent of revolutions. The Coast Guard Cutter "Tallapoosa" when reversed developed 53 per cent of full ahead power (1000 I.H.P.) at 62 per cent of ahead revolutions (129 R.P.M.), 60 per cent at 78 per cent of revolutions and 90 per cent at 109 per cent of revolutions. The torque developed while backing often exceeds the full power ahead torque, the backing trials of the "Delaware" showing about 9 per cent excess.

WEIGHTS AND DIMENSIONS

Weights of Marine Engines. There are many methods, some of which are very complicated, for estimating the weights of marine engines. For most purposes, the total weight of the engine is all that is desired and that may be obtained in several ways, the most direct of which is based on I.H.P. The following **weights per I.H.P. of vertical marine engines will** give a general idea of the weights of different types: Compounds 65 to 75 lbs.; three cylinder triples 85 to 110 lbs.; quadruples 110 to 135 lbs. However, on account of the different ratings of I.H.P. which the same size engine may have, the total weight may beestimated much more closely if based on cylinder volumes; or, for the sake of simplicity, the sum of the squares of the cylinder diameters multiplied by the stroke, all in inches, which may be designated as Σ. The product of $\Sigma \times$ constant C (which varies with the number of cylinders and the length of stroke) closely approximates the weight of the engine. C for vertical marine engines, exclusive of attached pumps and condensers, for engines with 24 in. to 40 in. stroke may be taken as follows: 0.9 to 0.75 for 3 cylinder triples, 1.10 to 0.9 for 4 cylinder triples. For engines with 40 in. to 66 in. stroke: 0.75 to 0.6 for 3 cylinder triples, 0.9 to 0.7 for 4 cylinder triples, and 0.9 to 0.7 for quadruples. For compound engines with 20 in. to 30 in. stroke, C may be taken as 1 to 0.85. These constants hold for engines of usual proportions but for very light or very heavy engines, the constants may come outside of the range given.

Centre of Gravity of Engines. In computing trim and stability of a vessel the centre of gravity of machinery must be known. In calculating this it is necessary to approximate the location of the centre of gravity of the main

engine. The **vertical centre of gravity** of a vertical marine engine above the centre of the crankshaft is approximately 1.75 times the length of stroke. The **horizontal centre of gravity** aft of the centre of the forward cylinder depends on the order of cylinders, but the following constants, in which D is the sum of the cylinder diameters, will be found sufficiently accurate: compounds with the H.P. cylinder forward, about $0.5D$; three-cylinder triples with H.P. cylinder forward and L.P. aft, about $0.65D$; four-cylinder triples with the L.P. cylinders on the ends, about $0.65D$; quadruples with the H.P. cylinder forward, about $0.75D$.

Overall Dimensions of Engines. In laying out an engine room arrangement it is necessary to know approximately the overall dimensions of the engine. The design of the engine affects these dimensions to a certain extent but in general the length of each type of engine bears a fairly constant relation to the sum of the cylinder diameters, which may be designated as D, and the height bears a similarly constant relation to the length of stroke. The constants given in Table 25 represent a fair average for vertical marine engines

Table 25. Overall Dimensions of Vertical Marine Engines

(D = sum of cylinder diameters)

Type of engine	Compound	Three-cylinder triple valves on centre-line	Three-cylinder triple valves at side	Four-cylinder triple	Quadruple
$\frac{\text{Length over cylinders}}{D}$	2.00	1.80	1.65	1.85	1.90
$\frac{\text{Length over bedplate}}{D}$	1.70	1.70	1.70	1.60	1.75
$\frac{\text{Height, centre of shaft to top of cylinders}}{\text{Stroke}}$	5.25	4.75	4.75	4.75	4.75
$\frac{\text{Width of bedplate}}{\text{Stroke}}$	2.7 to 3.0 for all types.				

Length of Engine Room. Proceeding along the same lines as above, there are certain ratios than can be used to approximate the length of the engine room for average conditions. When the machinery is amidships the length of the engine room is approximately from 2.25 to 2.75 times the sum of the cylinder diameters and when in the stern the length from the forward engine room bulkhead to the after peak bulkhead is approximately from 2.5 to 3.5 times the sum of the cylinder diameters for triple and quadruple engines. With compound engines the length of engine room is about five times the sum of the cylinder diameters.

MACHINING AND TOLERANCES

The allowances and limitations given below are based on good American practice for merchant marine engines.

Interior of Cylinders. The bores of cylinders and piston valve chests should be cylindrical within the following tolerances:

From 20 in. to 40 in. diameter.....0.003 in. to 0.005 in.
Larger than 40 in. diameter.....0.005 in. to 0.008 in.
Difference in diam. at top and bottom.....0.010 in.

Cylinder Bore and Piston Clearances. Between the bore of the cylinder and the diameter of the piston and follower the clearances in in. should not be less than the following:

Diam. of cylinder in in.	H.P.	I.P.	L.P.
21 to 30	0.035	0.0275	0.025
31 to 40	0.045	0.035	0.030
41 to 50	.....	0.05	0.0375
51 to 60	.....	0.06	0.050
61 to 80	.....	.....	0.065
81 to 100	.....	.....	0.085

Working Fits of Piston and Piston Valve Rings. The following allowances have been found satisfactory:

Diameter in in.	Min. in in.	Max. in in.
12 to 15	0.004	0.007
16 to 20	0.006	0.009
21 to 30	0.007	0.015
31 to 40	0.010	0.020

Slide Valves and Seats. Working surfaces should be true planes, free from blemish and thoroughly smooth. It is advisable to plane the working surfaces of slide valves at right angles to the direction of planing of the seats.

Piston rods and valve stems should be accurately ground or machined within the following tolerances:

Tolerances on diam. plus 0.012 in. minus 0.005 in.
Deviation from exact roundness, 0.003 in.

Bearings should be accurately machined and fitted, and special care taken to secure hardness and smoothness. The following tolerances in in. are permissible:

Diam. of shaft in in.	Plus or minus in diam.	Deviation from exact roundness
6	0.005	0.0025
10	0.006	0.003
16	0.007	0.0035
Larger	0.008	0.004

The following clearances for **pins and journals** are considered satisfactory:

Clearance in in. for pins		Diameter in in.	Clearance in in. for journals	
Max.	Min.		Max.	Min.
0.005	0.002	5	0.005	0.0025
0.008	0.004	10	0.010	0.005
0.012	0.006	15	0.015	0.0075
0.015	0.008	20	0.020	0.010
0.018	0.010	25	0.025	0.0125

Bearing clearances in in. as follows are recommended for large engines in good alignment:

Cross-head pin brasses..... 0.006 to 0.007
Crank pin brasses..... 0.008 to 0.009
Main bearings..... 0.009 to 0.010
Eccentric straps..... 0.010 to 0.015
Cross-head slippers..... 0.010 to 0.012
Thrust shoes..... 0.005 to 0.007

Stern Bearings. Lignum-vitæ must be well water soaked before being bored out. The following clearances are permissible on **lignum-vitæ bearings:**

Diameter of journal in in.	Minimum clearance in in. for new bearings	Wear in in. at which bearings should be renewed
8	0.050	1/4
10	0.060	1/4
12	0.075	9/16
16	0.100	3/8
20	0.125	3/8

Bearing Brasses. Sufficient clearance to permit of expansion should be allowed at the ends of the brasses where the two halves meet. Cooling water should not leak onto the bearing surface. Hollow brasses should be tested to demonstrate their ability to withstand pressure of water service.

Thrust bearings must be so fitted that all the rings perform equal duty.

Allowance for Expansion in Castings. Large iron castings subjected to steam of high temperature will expand or grow and take a permanent set. Where large cylinders are used ample allowance for fore and aft expansion of the cylinders must be made in locating valve rod guides and eccentrics.

SECTION 7

MARINE DIESEL ENGINES

BY

MAX ROTTER, Vice-President in charge of Engineering of the Busch-Sulzer Bros. Diesel Engine Co., Mem. A.S.M.E. and A.S.N.E.

CONTENTS

	Page
Classification	945
The Diesel Engine:	
Operating Processes	947
Pressures	948
Ratio of Delivered to Indicated Power	948
Internal Power Consumption	949
Fuel Supply and Injection	949
Scavenging	950
Lubrication	950
Cooling	950
Fuel Consumption	951
Overload Capacity	954
Lubricating Oil Consumption	954
Four Cycle vs. Two Cycle	954
Steam vs. Diesel	956
Types of Diesel Engines	957
Semi-Diesel Engines	962
Fuel Oils	963

MARINE DIESEL ENGINES*

BY

MAX ROTTER

Acknowledgment is made of information furnished by:
Bolinder's Company, New York.
Busch-Sulzer Bros.-Diesel Engine Co., St. Louis, Mo.
New London Ship and Engine Co., Groton, Conn.
Schneider et Cie., Crensot, France.
Weiss Engine Co., Chicago, Ill.

Internal-combustion Engines, for the propulsion of ships, may be divided into three general classes: (1) The Diesel engine; (2) the semi-Diesel engine; (3) the explosion engine; the distinguishing characteristics of each are briefly as follows:

In the **Diesel engine** a charge of pure air, contained in the working cylinder, is heated by mechanical compression by the inward-traveling piston, to a temperature considerably in excess of the igniting temperature of the fuel; the fuel being then introduced into this hot air, at or near the completion of the compression, gradually and in a finely atomized condition, being gasified and ignited, and burning while performing work upon the outward-traveling piston, the rate of fuel introduction corresponding to the rate of travel of the piston so that the combustion is not accompanied by any substantial increase of pressure in the cylinder. On account of this last-mentioned characteristic, this class of engine is sometimes referred to as "**constant pressure**" engines. Engines of this class use, for fuel, heavy liquid hydro-carbons, *i.e.*, oils, having a high vaporizing temperature, see pp. 272 and 963.

In the **semi-Diesel** engine the process is similar to that in the true Diesel, excepting that the compression of the air is not carried to the height which would heat it to a temperature sufficient to gasify and ignite the fuel without supplementary means. These supplementary means usually consist of a hot chamber, or similar heat-retaining element, directly communicating with or forming part of the working cylinder; into this element, or against this hot surface the fuel is sprayed for gasification and ignition, a characteristic which has given this class of engine the name of "**surface-ignition**" engines. The described method of operation naturally results in a more rapid combustion of the fuel than in the Diesel, accompanied by a sudden and substantial increase of pressure in the cylinder. Engines of this class use fuels similar to those used by Diesel engines.

In **explosion engines,** for marine use, the fuel is a liquid hydro-carbon having a relatively low vaporizing temperature, as, for example, gasoline, and enters the cylinder as a vapor combined with the charge of air, the combina-

*CAUTION.—Many of the structures and processes herein described are protected by U. S. and foreign patents. It would unnecessarily lengthen the text to state this fact with reference to each case to which it would apply; but designers and users are cautioned against utilizing structures or processes without having previously assured themselves that such use is not subject to patent restrictions.

tion being compressed in the cylinder, by the piston, to a pressure sufficiently low to ensure its non-ignition without the aid of special means, such as an electric spark. This method results in the sudden, explosion-like combustion of the greater part of the fuel before the piston has started its outward travel, and, therefore, before the volume of the combustion space has been appreciably increased by such outward travel. This last-mentioned characteristic has given this class of engine its name of **"constant volume"** engines. For gasoline, see p. 277.

The comparative processes of compression, ignition, and combustion, typical of these three classes of engine are illustrated in Fig. 1, showing the forms of the indicator diagrams at about the completion of the compression, and during the combustion and the partial expansion of the resultant gases: (*a*) in a Diesel engine, (*b*) in a semi-Diesel or surface-ignition engine, and (*c*) in an explosion engine. These diagrams are not to scale, being intended to show the processes and not the comparative pressures.

The explosion engine is not within the scope of this section, and no further reference will be made to it herein.

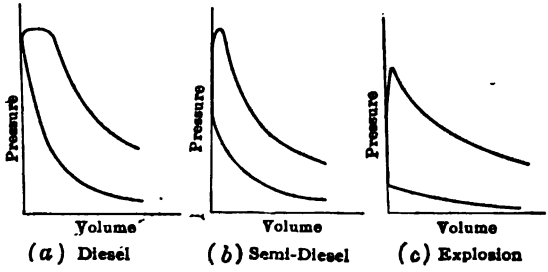


FIG. 1.

Diesel and Semi-Diesel. Even at this comparatively early stage in the use of heavy oil internal-combustion engines for ship propulsion, the **fields of application** of the Diesel and semi-Diesel, respectively, are becoming fairly clearly defined and delimited. Although there still is, and probably will continue to be, an overlapping of sizes in the capacities in which the class of service and commercial considerations may throw the advantage to one or the other, the Diesel unquestionably dominates the field for units of over 500 B.H.P., while the semi-Diesel is similarly favorably situated as regards units below 200 B.H.P. In the intermediate field both compete.

Excepting in the case of engines for very special uses, such as the propulsion of submarine boats, the marine engines of these classes have rarely been developed as types entirely distinct from their stationary prototypes. With comparatively few or no important modifications in their construction, the vertical stationary engines have been adapted to marine uses merely by providing them with means for reversing the direction of their rotation (in the small sizes even these are not required, reversing clutches or reversing propellers being used instead), with simple manually controlled speed regulating mechanisms, and with a sufficient number of cylinders to enable their direct and quick starting without reference to the positions of their cranks. In some designs concessions are made to marine steam engine practice, these

being then generally carried over by the same builders into their stationary designs; but there are no obvious service requirements, other than those above indicated, which would necessitate distinctly different constructions for the two fields of use.

On account of the speed requirements of the propellers, and, perhaps also, to conform somewhat to the preferences of marine engineers, these marine engines are generally operated at normal speeds 10 to 20 per cent. lower than their equivalent stationary engines.

To date marine Diesel and semi-Diesel engines have been built single-acting only. Double-acting engines of these classes have been attempted for stationary purposes, but, so far, with such uncertain success as has excluded this type from a service in which absolute reliability is the most important consideration.

Diesel and semi-Diesel engines are being built in both the four-stroke cycle (briefly "four-cycle"), and the two-stroke cycle (briefly "two-cycle") types. In a **four-cycle engine**, four strokes of the piston, or two revolutions of the crankshaft, are required to complete the cycle of processes in the working cylinder. During the first out-stroke of the piston atmospheric air is drawn into the cylinder, filling it completely at about atmospheric pressure; during the second stroke (the first in-stroke) this air is compressed; at or near the inner dead-centre the fuel is admitted and ignited; during the third stroke (the second out-stroke) the combustion and expansion take place, performing work on the piston; and during the fourth stroke (the second in-stroke) the products of combustion are expelled from the cylinder. These strokes are generally referred to, respectively, as admission stroke, compression stroke, working stroke, and exhaust stroke. Fig. 2 shows a schematic indicator diagram of a four-cycle Diesel engine; admission of air takes place from *A* to *B*, compression of the same from *B* to *C*, fuel injection commences and ignition takes place at *C*, fuel injection and combustion proceed from *C* to *D*, expansion of the gases from *D* to *E* (the combined line from *C* to *E* representing the working stroke), and exhaust from *E* to *A*.

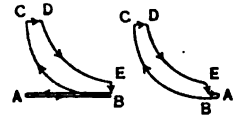


FIG. 2.

FIG. 3.

In a **two-cycle engine** these processes are accomplished during two strokes of the piston, or one revolution of the crankshaft. To do this it is necessary to provide air under slight pressure for clearing the cylinder of burnt gases (known as "**scavenging**") and for recharging it with fresh air, only a brief period, near the end of the out-stroke of the piston and the commencement of its in-stroke, being available for this purpose. Fig. 3 shows a schematic indicator diagram of a two-cycle Diesel engine; charging with fresh air takes place from *A* to *B*, compression of the same from *B* to *C*, fuel injection commences and ignition takes place at *C*, fuel injection and combustion proceed from *C* to *D*, the expansion of the hot gases from *D* to *E*, and exhaust and scavenging from *E* to *A*. As a matter of fact, there is no exact point at which the exhaust and scavenging cease and the charging commences, these processes taking place simultaneously during the greater part of the period between *E* and *B*.

THE DIESEL ENGINE

Operating Processes. The air contained in the cylinder is compressed to 450 to 500 lbs. per sq. in., usually 475 to 485 lbs. This is the **compression pressure** attained when the engine has become warmed up by running; when

cold, the pressure is from 30 to 75 lbs. lower, a fact which must be taken into account when adjustments are made. The **temperature of the air** at the completion of compression is about 850 deg. F. when the engine is cold, and about 1000 deg. F. when the engine is in full operation.

Into this volume of hot compressed air the fuel is sprayed, usually by means of compressed air furnished by a compressor provided for this purpose. The **pressure of the injection or spray air** varies between 550 and 1000 lbs. per sq. in., depending upon the type of the atomizer, the character of the fuel, and the speed and load. Naturally, the injection pressure must exceed the compression pressure. It is important that the fuel be not injected too rapidly, and that it be thoroughly atomized and diffused as it enters the cylinder. Injection of the fuel is so adjusted that it proceeds during the first 8 to 12 per cent of the working stroke, and that the fuel is promptly ignited and burns without any substantial increase in pressure. To this end it is important that the spray valve and its accessories be so designed, and the spray air pressure so adjusted, that, during the period between the completion of one injection and the commencement of the next, a small quantity of fuel remaining in the atomizer collects around the tip of the valve, to enter the cylinder and ignite quickly ahead of the main charge of air and fuel. Thus, the cooling effect of the injection air expanding into the combustion air is counteracted, and the prompt gasification and ignition and steady non-explosion-like combustion of the main charge of fuel is ensured.

Following the completion of the injection a short "**after-burning**" takes place, accompanied by falling pressure, and the hot gases then expand, continuing to perform work on the piston, until the end of the working stroke, where, at full load, their pressure has fallen to about 35 lbs. per sq. in. The gases leave the cylinder at an average temperature around 800 deg. F.

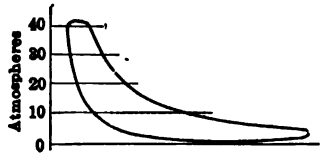


FIG. 4.—Four-cycle.

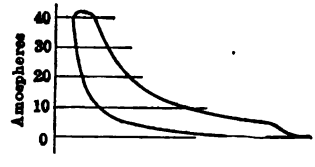


FIG. 5.—Two-cycle.

Pressures. When the engine is loaded to its maximum, the **peak pressure** on the working piston will be around 575 lbs. per sq. in.; although the engine parts should be designed to safely carry 5 to 10 per cent more than this, to allow for irregularities in operation or adjustments. **Indicated mean pressures** as high as 130 lbs. per sq. in. can readily and safely be maintained in four-cycle Diesel engine cylinders, and 120 lbs. in two-cycle; but, to allow ample margin for emergencies, and to avoid the necessity of having to immediately correct every minor dis-adjustment, it is customary to rate marine engines on a considerably lower basis, namely, between 90 and 95 lbs. indicated mean pressure. Figs. 4 and 5 are copies of normal full load indicator diagrams of a four-cycle and a two-cycle marine Diesel engine respectively.

Ratio of Delivered to Indicated Power. There is a considerable difference between the ratio that the actually delivered power bears to the indicated power of a Diesel engine and that of a steam engine. This is misleading, more especially as this ratio, in the case of the Diesel engine, is often referred to as its "**mechanical efficiency.**" In determining the ratio in the case

of a steam engine, it is usual to consider the engine alone, the power consumption of such auxiliary apparatus as condenser pumps, etc., not being charged against the engine output. In the Diesel engine, on the contrary, the air compressor is almost invariably built integral with the engine, and, in the case of two-cycle engines, generally the scavenging pump also; the power consumption of these auxiliary machines is treated as though it were part of the internal losses of the engine. As understood at the present time, the ratios of full load delivered horsepower (B.H.P. or S.H.P.) to the I.H.P., compare approximately as follows:

In steam engines	90 to 94 per cent.
In four-cycle Diesel engines	72 to 76 per cent.
In two-cycle Diesel engines	67 to 71 per cent.

The foregoing is important in determining the size of Diesel engines required to propel a ship which was previously equipped with steam engines.

Internal Power Consumption. Distinct from the thermodynamic losses, the difference between the power actually developed in the cylinder (I.H.P.) and the power actually delivered at the coupling end of the crankshaft (B.H.P.), referred to in the previous paragraph, is made up of internal engine friction and the power required to drive the integral or directly coupled auxiliaries. The distribution of the I.H.P., at full load, is approximately:

In a **four-cycle** Diesel engine—	
Internal engine resistances	15 to 17 per cent.
Air compressor	9 to 11 per cent.
B.H.P.	72 to 76 per cent.
In a **two-cycle** Deisel engine—	
Internal engine resistances	12 to 14 per cent.
Air compressor	8 to 10 per cent.
Scavenging pump	8 to 10 per cent.
B.H.P.	67 to 71 per cent.

Included in the internal resistances are the small powers consumed by the lubricating and cooling water pumps, amounting to about 1 per cent.

Fuel Supply. The speed and power capacity of the engine being solely controlled by the quantity of fuel delivered into its working cylinders, up to the maximum which can in actual operation combine with the available air in these cylinders, the regulation of this quantity is of the greatest importance. The volume of fuel consumed per cylinder per working stroke is exceedingly small—only about one cubic inch in a 1500 horsepower, six-cylinder, four-cycle engine at full load—the variation of this must be accomplished by means which must be delicately responsive to adjustment, yet rugged enough to stand up to the work of delivering against a pressure equal to that of the spray air. For this purpose a special fuel-feed pump is used, with a separate plunger to serve each separate working cylinder. The fuel oil flows to this pump by gravity, and the quantity delivered by the pump is regulated either by varying the plunger stroke, or by varying the plunger chamber volume during the stroke, or by allowing a regulable quantity of the oil to be by-passed or to be forced back into the supply line by preventing the closing of the pump suction valve during a longer or shorter portion of the pump delivery stroke.

It is essential that means be provided to free the pump and all piping between it and the spray valve of all air, and to prevent the passage of spray air from the spray valve into this piping.

Fuel Injection. The fuel is generally sprayed (atomized and injected) into the working cylinder, by means of compressed air at a pressure between 900 and 1100 lbs. per sq. in. at full load. **Mechanical spraying,** sometimes

referred to as "**solid injection,**" was originally used by Dr. Diesel, and abandoned by him, and since then has been variously attempted; but, to date, the vast majority of builders of Diesel engines adhere to the **air spraying.**

The fuel enters the combustion space of the working cylinder in a finely divided or atomised condition so that the particles may be readily gasified, and in such intimate contact with the air as will ensure perfect combustion within the short period available for this. At the same time, the injection proceeds at a definite rate which will maintain a substantially constant pressure during the period of injection.

The **air pressure** is so adjusted that it will accomplish the above-stated purpose, yet not entirely empty the atomizer of fuel, or unnecessarily cool the contents of the cylinder. This renders it necessary to either vary the air volume and pressure according to the quantity of fuel to be injected, or to vary the volume by varying the period of injection. The former is the more usual method, and is simply accomplished by throttling the low-pressure suction of the compressor. A more efficient and economical arrangement is to add to this throttling a coöperating control of the amount and period of the opening of the spray valve. In stationary engines these air regulations are frequently automatic and, like the fuel delivery, under governor control; but in marine engines they, as also the fuel, are manually controlled.

The spray air must be cool, to avoid the vaporization of the fuel or carbonising in the atomizer. The air must also be free from entrained moisture, which would retard the combustion, and, if present in appreciable quantity, would injure the spray valve.

The volume of spray air required to properly atomise and inject the fuel, is about 10 per cent. of the combustion air compressed in the cylinders, varying but slightly on account of differences in the construction of the atomizers.

Scavenging. The air for scavenging and charging the working cylinders of two-cycle engines enters the cylinders **through valve-controlled ports** in the cylinder heads, or **through piston-controlled ports** in the cylinder walls. In one system the wall ports are in two tiers, the lower being piston-controlled, the upper valve-controlled. The **scavenging air pressure** depends upon the admission means, and the dimensions and arrangement of the exhaust piping and muffler; it is rarely less than 3 or more than 7 lbs. per sq. in. The **volume of scavenging air** is always greater by 30 to 60 per cent, than that necessary to fill the working cylinders at atmospheric pressure.

Lubrication. Excepting in smaller engines, the **main bearings, crank pins, and wrist pins,** are usually lubricated by a continuous pressure system, the oil discharged from the bearings being collected in the engine base, filtered, and cooled, and re-circulated. In some cases the system works as a gravity pressure system; in others it is under direct pressure from pumps attached to the engine. It is important that suitable oils be used and that the filtering be efficient. For the lubrication of the **working and compressor cylinders,** a regular multi-plunger cylinder lubricator is used.

Cooling. The cooling of the working and compressor cylinders and their heads, as also the portions of the exhaust piping adjacent to the engine, is absolutely necessary. Intercoolers and aftercoolers for the spray air, and coolers for the lubricating oil must also be provided. In the larger sizes the working piston heads must be cooled.

For the above purposes water is used, excepting in special cases in which the same oil as is used for lubrication is also used for cooling the piston heads. In commercial marine engines the direct use of sea water is becoming general,

causing no difficulties if circulated in quantity sufficient to avoid discharge temperatures in excess of 110 deg. F.

The quantity of heat which the cooling water should be capable of carrying off from an ordinary commercial marine Diesel engine, including the cooling of the lubricating oil and the exhaust connections, is about 35 per cent of the total heat in the fuel consumed. Therefore, an engine consuming per B.H.P.-hr. 0.40 lb. of fuel oil of 18,500 B.t.u. per lb., or 7400 B.t.u. per B.H.P.-hr., will require cooling water sufficient to carry off 2600 B.t.u. per B.H.P.-hr. Allowing for operation in the tropics, where sea water may have a temperature of 85 deg. F., and limiting the discharge temperature to 110 deg. F., the **quantity of cooling water** will be, say, 104 lbs. or 12½ gal. per hr. per B.H.P. **The pressure** of this will depend upon the arrangements in the ship, but will ordinarily not exceed 30 lbs. per sq. in.

Fuel Consumption. The following general data, relating to fuel consumption, are limited to petroleum fuels (crude petroleum, distillates, and fuel oil); tars, tar oils, and similar liquid fuels, now being increasingly utilized in stationary Diesel engines, are not herein considered, as their adoption for marine purposes is not imminent.

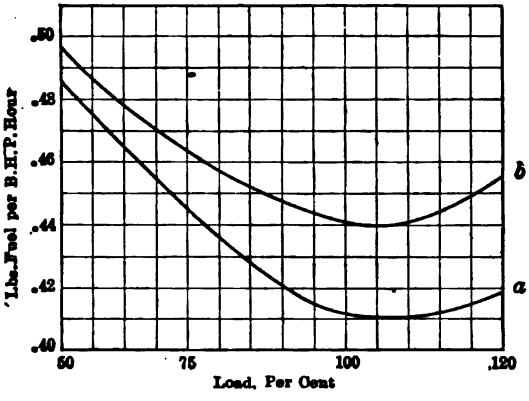


FIG. 6.—Fuel consumption; (*a*) four-cycle, (*b*) two-cycle.

Fig. 6 shows two curves of the fuel consumption, in pounds per B.H.P. per hour, of (*a*) a **four-cycle** Diesel engine of about 1500 B.H.P., and (*b*) an equivalent **two-cycle** Diesel engine, at constant speeds, both engines having their auxiliaries integral or directly coupled.

When propelling a ship, the variation in the engine power results from a variation in the ship's speed, and the power diminishes more rapidly than the speed. For ordinary purposes it may be assumed that, within the limits which are of practical consideration, the engine-speed will be directly proportional to the ship-speed, and the engine load proportional to the cube of the ship-speed. These relations, expressed in percentages, are shown, respectively, by curves (*a*) and (*b*) of Fig. 7, in which 100 per cent is the rated speed and load point.

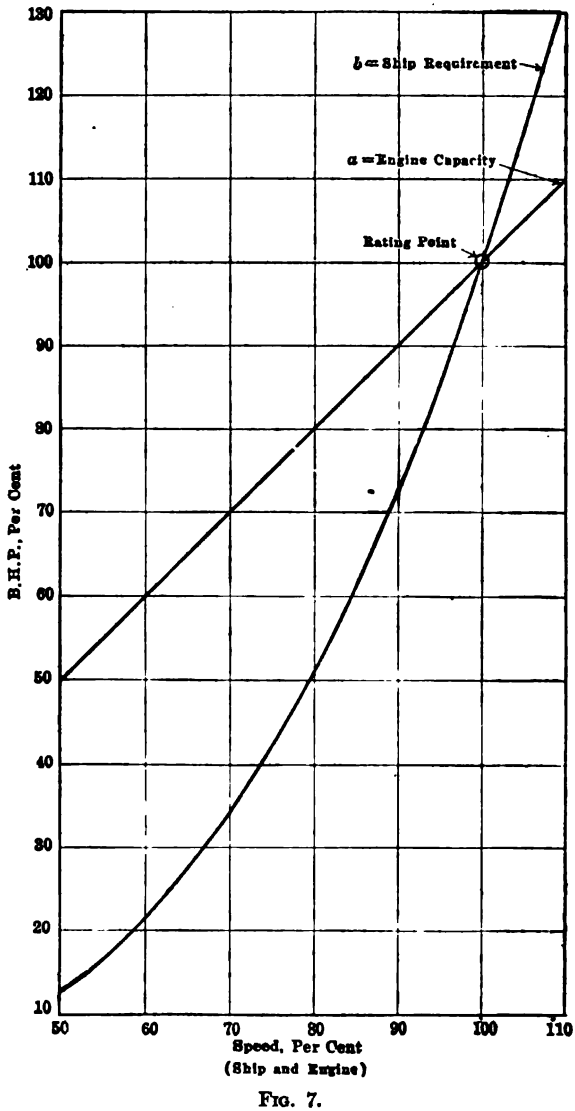

b = Ship Requirement
a = Engine Capacity
Rating Point
B.H.P., Per Cent
130
120
110
100
90
80
70
60
50
40
30
20
10
50
60
70
80
90
100
110
Speed, Per Cent
(Ship and Engine)

FIG. 7.

Furthermore, within the ordinary running limits, the reduction in engine-speed will be accompanied by a slight decrease in the specific fuel consumption; while, within the same limits, the normal rating of the engine, on which its full and fractional load fuel consumptions are based, will be proportional to

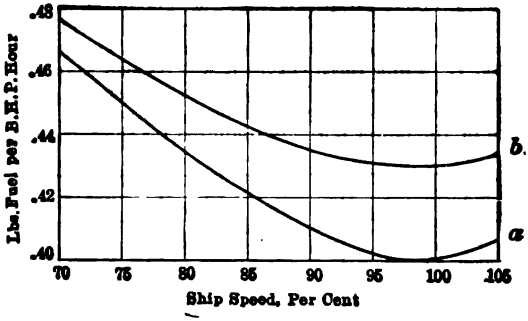


FIG. 8.—(*a*) Four-cycle; (*b*) two-cycle.

the engine-speed. The combination of these variables produce the curves in Fig. 8, which show the fuel consumption per B.H.P.-hr. at various percentages of the rated or trial ship-speed, (*a*) of the four-cycle, and (*b*) of the two-cycle engines previously referred to.

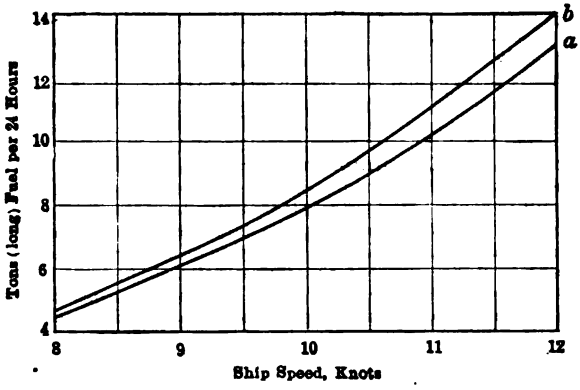


FIG. 9.

The curves in Fig. 9, prepared from Figs. 7 and 8, show the total long tons of fuel consumed per 24 hours, at various ship speeds, by the propelling engines of a ship having a trial speed of 12 knots with a power requirement of 3000 B.H.P. (or S.H.P.).

Overload Capacity. The ability of the Diesel engine to respond to the demand for excess power in heavy weather may be deduced from Fig. 7. For marine use, Diesel engines are rated on a lower basis than are the same engines when used for stationary purposes; at their rated normal speed and load they, therefore, already have a considerable capacity for carrying an overload, usually about 25 per cent. The maximum power of a Diesel engine at various speeds, and the power required to propel the ship, are shown in percentages by curves (*a*) and (*b*), respectively, of Fig. 10, which is a modification of Fig. 7. From these curves it will be seen that the percentage of overload capacity increases as the engine and ship speeds are reduced under heavy weather conditions; curve (*c*) shows this available overload capacity in percentage of the power normally required for the respective ship-speeds.

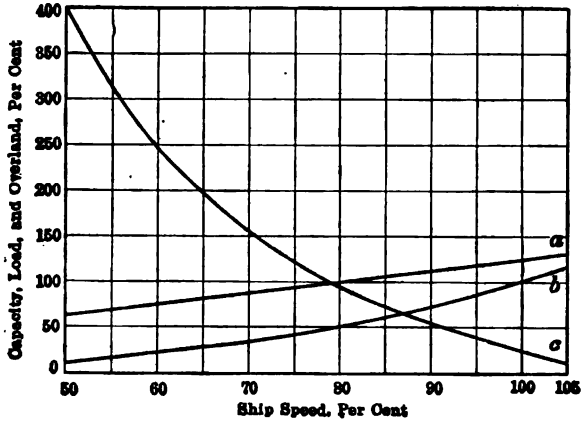


FIG. 10.

Lubricating Oil Consumption. The consumption of lubricating oil cannot be set down with anything like the definiteness of that of fuel, as it is more difficult to control, and wastefulness in its use is less promptly and easily noticed by the ship's officers. In the larger sizes of commercial marine Diesel engines, the consumption of lubricating oil may range all the way from 0.4 per cent to 2 per cent of the consumption of fuel; and in high-speed engines, such as the propelling engines of submarines, especially such as use lubricating oil for cooling their pistons, the consumption may be twice the above-mentioned figures.

Four-cycle vs. Two-cycle. Briefly stated, the following claims are advanced by the advocates of four-cycle and two-cycle Diesel engines, for marine propulsion, respectively, and are here repeated without comment:

For the four-cycle:

1. Lower fuel consumption, due to the absence of the power consumption of the scavenging pump; also, to the more thorough clearing of the cylinders of the products of combustion, attainable by virtue of the exhaust valves in the cylinder heads.

2. Simpler construction, the special scavenging pump, with its mechanism, being omitted, and its functions being performed by the working pistons themselves, all of which, together with their mechanisms, are alike.

3. The absence of a space-consuming and heat-radiating scavenging air receiver.

4. Cams and camshaft run at half the number of revolutions per minute of the crankshaft.

5. Less liability of accident to the fuel pump, its speed being half that of a two-cycle engine of the same number of revolutions.

6. Greater possible speed reduction, without irregularities in running, as a result of the more complete purging of its cylinders, and the maintenance of the compression in the cylinders more nearly at its full speed pressure than is possible in a two-cycle engine in which the pressure of the scavenging air falls off rapidly at reduced speeds.

7. Reduced risk of heat failures of cylinders, cylinder heads, and pistons, as the quantity of fuel consumed per minute per square inch of piston and head area is about half of that in a two-cycle; also, because these parts are exposed to the hot gases during a shorter relative period of the cycle; also, because less hot gases are liable to remain in the cylinder after the completion of the exhaust.

8. The ability to carry greater overloads, because it is possible, for the reasons stated under (7), to increase the quantity of fuel burned without danger of excessive heat stresses.

9. Reduction in the variety of spare parts, on account of the elimination of the scavenging pump and its mechanism, and the scavenging valves with their mechanism.

For the two-cycle:

1. Only slightly higher fuel consumption, as such engines run most of the time at loads below their rating, and the fuel consumption of the two-cycle approaches that of the four-cycle at underloads of normal operation.

2. Reduction in weight by about one-third, as it can develop the same power with much smaller or fewer cylinders, having a power stroke for each revolution instead of for each two revolutions.

3. Reduction in space requirements, for the same reason as (2).

4. Reduction in foundation requirements, due to its lighter weight, smaller dimensions, and steadier running.

5. The possibility of being located farther aft in the boat, due to its reduced space requirements; thus shortening the lineshaft and tunnel, and thus still farther increasing the net space and weight capacity available for cargo.

6. Steadier running, the variation in turning moment in a four-cylinder two-cycle engine being only two-thirds of that in a six-cylinder four-cycle, and in a six-cylinder two-cycle less than one-fifth of that in a six-cylinder four-cycle engine.

7. Reduced diameter and weight of line and tail shafts, for the same reason as (6).

8. Easier maneuvering, due to the very greatly reduced number and weight of the valves, valve gears, and cams.

9. Operation not affected by the wear of crosshead pin and crank pin bearings, because the working forces on these parts are always in one direction, the inertia of the moving parts never reversing the forces as is unavoidably the case in each cycle of a four-cycle engine.

10. No tension stresses on the caps and bolts of main bearings and crank pin boxes, the piston forces being constantly downward as stated in (9).

11. Safer construction of cylinder heads; on account of the absence of the large admission and exhaust valves required in the heads of four-cycle engines, it is possible to make the heads of two-cycle engines of exceedingly simple symmetrical form, thus avoiding casting and heat stresses and permitting far more efficient water cooling.

12. Absence of exhaust valves, with their unavoidable troubles, such as leaks, their liability to break and fall into the cylinder, and the constant attention they therefore demand.

13. Ability to operate successfully with cheaper fuels, and fuels readily obtainable in any part of the world, there being no exhaust-valves to be affected by sulphur and asphaltum.

14. Reduced consumption of lubricating oil, due to the reduced weights and dimensions of the moving parts.

Steam vs. Diesel. The relative advantages of Diesel engines, as compared with steam engines or turbines, when applied to ship-propulsion, are somewhat affected by the character, service, and routes of the ship. They are, for example, greater in the case of heavy freighters making relatively long runs, in which the fuel supply must be large, and any reduction in fuel consumption has, therefore, a proportionately greater effect upon the net cargo capacity, than in the case of package freighters making short runs. They are greater on the ocean, where steam necessitates the carrying of fresh water, than on rivers and lakes.

The largest gain is naturally that directly resulting from the low fuel consumption of the Diesel engine. Roughly, the fuel consumption of larger installations, of the various types of ship-propelling machinery, including the auxiliaries required by it, but not including the ship service auxiliaries, reduced to pounds of fuel oil per B.H.P.-hr., compares as follows:

Diesel Engines, directly coupled	0.4 to 0.45
Steam Turbines, geared	0.9 to 1.1
Triple-expansion Steam Engines	1.1 to 1.4

Therefore, in the cost of fuel alone, the Diesel engine nets a saving of 45 to 60 per cent as compared with the best steam equipment.

The reduction in fuel consumption increases the net cargo carrying capacity of the Diesel-engined ship by the same amount, compared with a steamer also using fuel oil; but compared with a coal-burning steamer, the increase in cargo capacity is much greater. Weight for weight, there is not much difference between the bunker space required for the same quantity of oil or coal; but, as compared with a triple-expansion steam engine, the weight of bunkered fuel oil for the Diesel is less than one-third of the weight of the bunkered coal for the steam engine with the same cruising radius, and much of the oil can be bunkered in spaces which would under any circumstances be unavailable for cargo. The indirect effect of the foregoing is that, in spite of the higher cost of the Diesel engine equipment, the cost of the complete ship, based upon its net cargo capacity, is lower.

The Diesel engine may, without any time-consuming preliminary steam-raising or heating-up, be at any moment started and immediately brought up to full speed and capacity. It is, therefore, also prepared for any eventuality, without the stand-by fuel consumption required by steamers.

As compared with coal, the bunkering of oil fuel is a clean and simple process which may readily be carried out on the high seas, and, if done in port, does not hamper the loading and unloading of the cargo. The expense of trimming is entirely eliminated; the storage aboard involves no hazard of spontaneous combustion; it is a question of only a short time before fueling stations for

oil will be as conveniently and frequently located as they now are for coal; see p. 253.

In operation, the Diesel-engined ship is necessarily cleaner than the intermittently fired and, consequently, smoky coal-burner, and usually also than the steamer with oil-fired boilers which are subject to the vagaries and varying abilities of the firemen. This cleanliness reduces the maintenance costs and the time lost in cleaning and overhauling.

The power developed by the Diesel engines, and with it, the speed of the ship, are always under the direct control of the engineer, and are not dependent upon the watchfulness and conscientiousness of a fire-room crew, while the maneuvering of Diesel engines is as quick and reliable as that of steam machines.

Losses resulting from leaky steam pipes, and the dangers of the none too rare explosions of steam pipes and valves are entirely eliminated by the Diesel engine.

The annoyances, wages, food, and quarters of the fire-room crew are saved by the Diesel engine, while the conditions in the engine-room make for greater efficiency on the part of the engineers.

The comparative weight of complete installations of steam and Diesel propelling equipments depend so much upon the type of each that no broad statement can apply to all cases; nor is this a matter of great moment, as the difference, either way, is insignificant when compared with the cargo capacity of the ship.

In one respect, only, does the Diesel installation inherently compare unfavorably with the steam installation; namely, in the higher lubricating oil consumption of the former, two to five times that of the latter. A comparison of average installations on the basis of hourly consumption per B.H.P. shows, for the Diesel engine, 0.45 pounds of fuel and 0.007 pounds of lubricating oil; for the triple-expansion steam engine, 1.4 pounds of fuel and 0.002 pounds of lubricating oil.

Types of Diesel Engines

Four-cycle. A typical construction of cylinder, cylinder head, and piston of a medium-sized four-cycle marine Diesel engine is shown in Fig. 11. This is a **"Nelseco" Diesel engine,** in this size of which the cylinder head is integral with the cylinder, the admission and exhaust valves are disposed horizontally, and the piston is of the trunk type.

A typical construction of the same parts of a large four-cycle marine Diesel engine, of **"Werkspoor" design,** is shown in Fig. 12. The cylinder head, with ports for all operating valves, is a separate casting, the admission and exhaust valves are disposed vertically, the pistons are short and are guided by means of outside crossheads.

Two-cycle. The regular two-cycle marine Diesel engines may be divided into two classes, viz.: those provided with scavenging valves in the cylinder heads, and those provided with scavenging ports in the cylinder walls.

A typical construction of the former is shown in Fig. 13, which illustrates the cylinder portions of a **"Schneider" (French)** two-cycle marine Diesel. The scavenging valves are cam-operated; the exhaust ports are piston controlled.

The ordinary marine Diesel engine with port-scavenging, is provided with a single tier of scavenging ports, in the cylinder wall, solely under piston control. In its outward travel the piston first uncovers the exhaust ports

FIG. 11.—Nelseco Diesel engine.

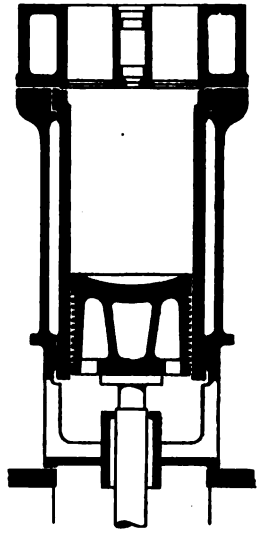


FIG. 12.—Werkspoor Diesel engine.

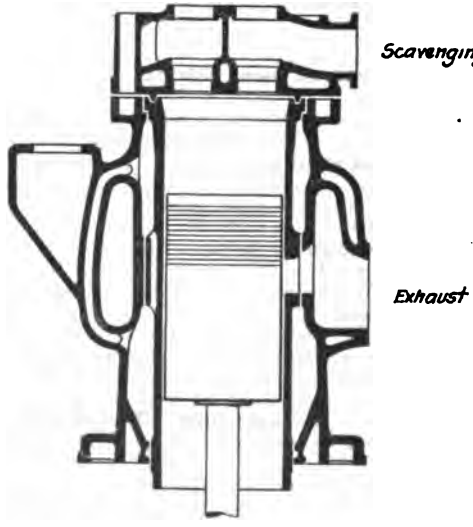


FIG. 13.—Schneider Diesel engine.

sufficiently to cause the pressure in the cylinder to fall below the scavenging pressure, after which the scavenging ports are uncovered by the farther outward travel of the piston. The tops of the scavenging ports are, therefore, set lower than the tops of the exhaust ports, and the inward-traveling piston closes them off before it closes off the exhaust ports.

A modification of port-scavenging is the **"Sulzer" two-cycle system,** illustrated in Fig. 14. In this construction the cylinder is provided with one tier of exhaust ports, and two tiers of scavenging ports. The latter are so located that the upper tier is uncovered by the piston appreciably earlier,

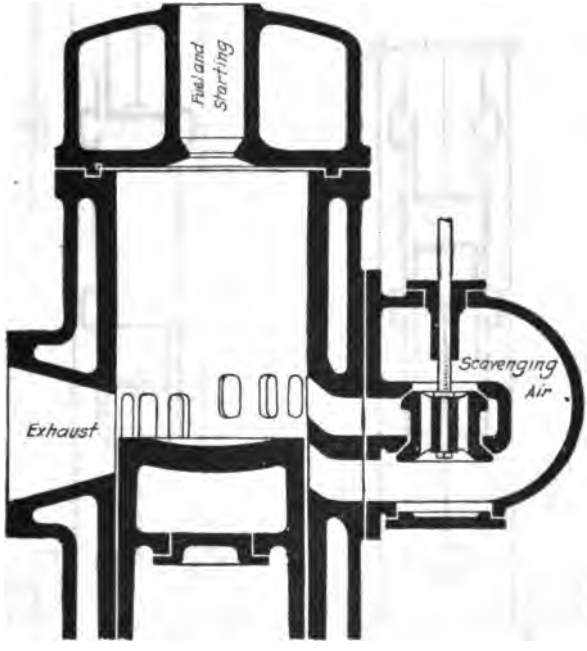


FIG. 14.—Sulzer Diesel engine.

and the lower tier appreciably later, than are the exhaust ports. The closing of all ports is performed by the piston; but the opening of only the exhaust and the lower scavenging ports is piston-controlled, there being a timed valve in the passage leading from the scavenging air receiver to the upper scavenging ports, which valve opens at about the same time as the piston uncovers the lower scavenging ports. The result is that a most effective scavenging is accomplished without any risk of a blow-back into the scavenging air receiver, and an increased weight of air is charged into the cylinder through the upper scavenging ports after the exhaust ports have been closed, making it possible to carry higher mean indicated pressures, without increasing the

maximum pressure, than would otherwise be the case. The valve of the upper scavenging ports does not close until after the piston has shut off these ports, and this closing may take place at any time between such piston closure and the next uncovering of these ports by the piston.

The "Junkers" Engine is a special form of two-cycle Diesel. The general features of this type are illustrated in Fig. 15, in which (*a*) shows the pistons in their extreme "in" positions, and (*b*) in their extreme "out" positions.

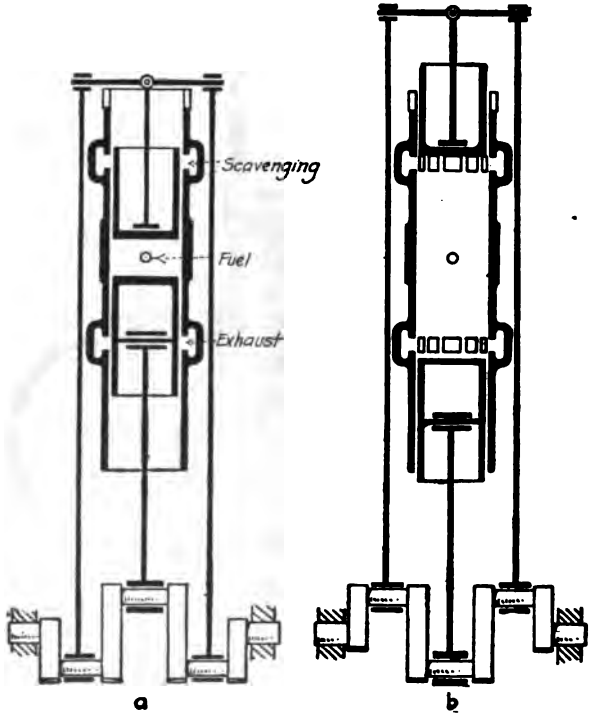


FIG. 15.—Junkers Diesel engines.

In this engine two pistons work in the same cylinder, in opposite directions. The cylinder is therefore a barrel, open at both ends. The crankshaft is provided with three cranks to each cylinder, the outer cranks being at 180 deg. to the inner crank. One piston couples directly to the centre crank. The other piston couples to the two outer cranks, through a yoke and side rods. The scavenging and exhaust ports are at opposite ends of the cylinder; one piston controls the scavenging, and the other piston controls the exhaust. The fuel valve is located substantially in the centre length of the cylinder.

The operation of the engine is on the true Diesel principle, the air being compressed between the two pistons instead of between a piston and a cylinder head. Obviously an excellent working balance is possible in this engine, and it is claimed that considerably higher compression pressures may be safely carried than in the ordinary Diesel, and that better fuel efficiency is thus attained. In conjunction with this type of engine is used a method of throttling the exhaust and thus increasing the weight of the air compressed between the pistons, as also the compression and combustion pressures, and thus forcing the engine to greater overloads than are possible in the normal types of Diesel engines.

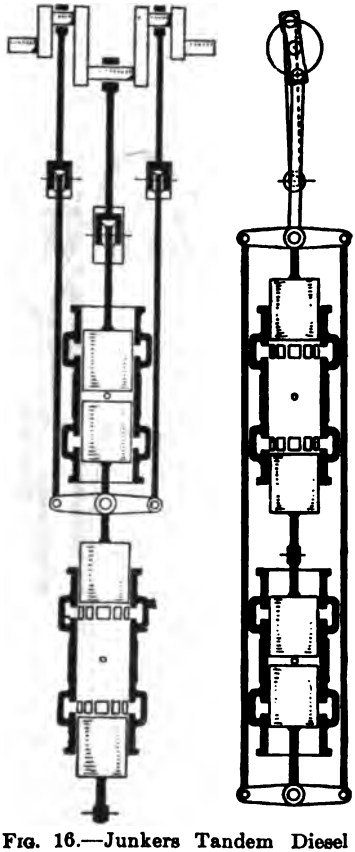


FIG. 16.—Junkers Tandem Diesel Engine.

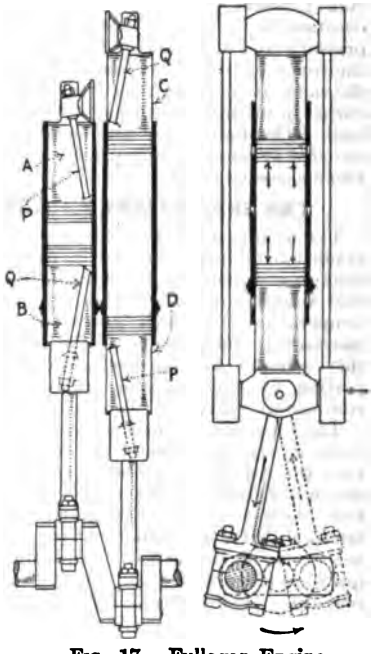


FIG. 17.—Fullagar Engine.

This type of engine has also been built in the tandem arrangement shown in Fig. 16, practically resulting in a double-acting engine.

The "Fullagar" Engine. A further modification of the "Junkers" idea is found in the "Fullagar" engine, illustrated in Fig. 17. The double-acting effect, obtained with the tandem "Junkers," is here attained with the two double-pistoned cylinders arranged side by side, and with the use of only two, instead of three, cranks. The operation is obvious from the illustration.

The "Sabathé" Engine is, strictly speaking, not a true Diesel in that it does not subject the air to mechanical compression to a temperature "far in excess of that necessary to ignite the fuel;" on the other hand, its main processes are similar to those of the true Diesel and its secondary processes practically resolve themselves into the definite performance of a function which is less definitely, although actually, part of the working processes of every true Diesel. In the "Sabathé" engine the compression is carried to a degree which will ensure the gasification and ignition of a small quantity of fuel injected approximately on the inner dead-centre. This initial fuel burns at constant volume, causing a considerable increase in pressure and temperature, in the cylinder, before the main fuel is injected, which injection of main fuel is made in such a manner that it burns at substantially constant pressure. The initial charge of fuel is constant in quantity, therefore the initial pressure and temperature at the time of the main fuel injection remain practically the same at all loads; the main charge is varied in quantity according to the engine load. The fuel valve and gear are constructed to cause the spray valve to open in two successive steps.

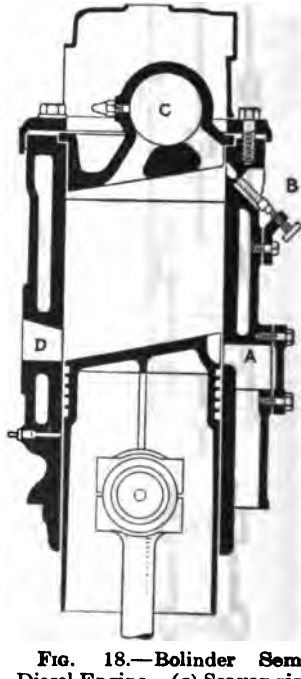


FIG. 18.—Bolinder Semi-Diesel Engine. (*a*) Scavenging ports, (*b*) spray valve, (*c*) ignition chamber, (*d*) exhaust ports.

THE SEMI-DIESEL ENGINE

The designation "Semi-Diesel" is herein applied to all heavy-oil internal combustion engines in which the fuel is gasified within the working cylinder, and the ignition temperature is in part generated by the mechanical compression of the air within this cylinder and in part supplied by a portion of the walls or surfaces of the combustion space.

The processes in such engines differ from those in the true Diesel engine chiefly in that the compression pressure of the air in the semi-Diesel cylinder is only about one-half of that in the Diesel, and that the ignition and combustion of the fuel are accompanied by a rise of pressure to a maximum more or less approaching the pressure maintained during Diesel combustion.

The heat supplied by the ignition chamber or wall, to aid in the gasification and ignition of the fuel for one stroke of the piston, is accumulated during the preceding combustion stroke.

The fuel supply and regulation are accomplished by means similar to those used for the Diesel; but, particularly in the smaller sizes, the fuel injection in many makes of these engines is performed mechanically, without the aid of compressed injection air.

The cylinders and heads are water-cooled, excepting the ignition chamber or the ignition surface, which are, if anything, protected against radiation losses.

Semi-Diesel engines are being built both four-cycle and two-cycle. In the two-cycle engines, the scavenging air is generally not supplied by special pumps but is compressed in the crank-case by the descending working pistons, with or without increased diameter.

The **thermodynamic efficiency** of semi-Diesel processes must always be inferior to that of the true Diesel. However, in small sizes, in which mechanical fuel-injection is practicable, the useful efficiency may closely approach that of the true Diesel of similar size, the thermodynamic deficiency being offset by the absence of mechanical power consumption by a compressor.

Fig. 18 illustrates the cylinder, cylinder head with ignition chamber, and piston of a **"Bolinder" engine.** This engine is two-cycle, the scavenging air being compressed in the crank-case by the descending piston, and entering the cylinder through the scavenging ports *A*. The fuel is injected by the spray valve *B*, into the ignition chamber *C*. The scavenging ports, as also the exhaust ports *D*, are piston controlled.

FUEL OILS

(See also p. 272.)

The standardization of the limits within which the chemical and physical properties of liquid fuels may vary without detriment to their storage aboard and their use in internal combustion engines, is still afar off. Some valuable data has, however, been published by the Bureau of Mines, in its Bulletin No 156, compiled by Mr. Herbert Haas.

It is unlikely that fuels other than petroleum products will, for a long time to come, be used in marine Diesel engines; and, for the time being, some engine builders set their own limits to their characteristics, while others leave these to be discovered by the engine user. Certain **general physical requirements** are absolutely essential, irrespective of the type of engine in which the fuel is to be used; these are:

1. Freedom from solid impurities, such as sand, clay, woody fibres, etc., which would clog the pipes and valves, and injure the pumps. The fuel should be strained through wire mesh when being stored aboard, and filtered through felt during its transfer to the service tanks.
2. Freedom from water, which would cause erratic operation of the engines.
3. The viscosity should be low enough to allow the fuel to flow freely to the pumps at the lowest temperatures likely to exist in any part of the ship.
4. The temperature at which the fuel begins to give off any appreciable quantity of vapor should be well above the highest temperatures likely to exist in any part of the ship in which the fuel would be even temporarily stored.

The **chemical properties** should be:

1. Low sulphur contents.
2. Low contents of coke- and ash-forming constituents.
3. Low contents of constituents which will not be completely gasified and consumed within the available time and at the temperatures existing in the engine combustion chambers.

Crude oils are rarely desirable marine Diesel fuels. Either they contain sufficient light volatiles to be unsafe for storage aboard; or they contain a large percentage of constituents of too slowly combustible properties.

Residuum or bottoms usually contain too many impurities and slowly combustible constituents, as well as being generally of too high a viscosity for handling without supplementary heating.

Distillates at the higher temperature ranges are the most desirable and economical Diesel fuels.

SECTION 8

VACUUM AND CONDENSERS

BY

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CONTENTS

	Page
Vacuum Measurement	967
Limits of Vacuum	969
Thermodynamics	970
Condensers:	
Main Surface Condensers	975
Surface Condenser Design	976
Scoop Condensers	982
Keel Condensers	983
Auxiliary Condensers	980
Jet Condensers	983
Jet Condenser Design	987
Circulating Pumps	988
Air Pumps:	
Wet Air Pumps	991
Dry Air Pumps	992
Wet and Dry Air Pumps	997
Vacuum Augmenters	1000
Condensate Pumps	1000
Accessories	1001
Arrangement of Condensing Plants	1006

VACUUM AND CONDENSERS

BY

G. L. E. KOTHNY

REFERENCES: Ellenwood "Thermodynamic Tables"; Hausbrand, "Evaporating Condensing & Cooling"; Orrok, Trans. A. S. M. E., vols. 32 & 34; Wm. Weir, Trans. Inst. of Eng. & Shipbl. in Scotland, 1912; Kothny Trans. Jun. Inst. of Eng., London, 1910.

Vacuum and Its Measurement. If the absolute pressure in an enclosed space is less than the absolute pressure of the surrounding atmosphere, it is customary to say that there is a vacuum in the space. To measure this vacuum a U-tube, partly filled with mercury, is used; one leg communicates with the space, the other being open to atmosphere. The absolute pressure corresponding to the vacuum acts on the mercury in the leg communicating with the space under vacuum, while the absolute pressure due to atmosphere acts on the other leg. The vertical distance between the two levels in inches or millimeters of mercury is a measure of the vacuum in the enclosed space.

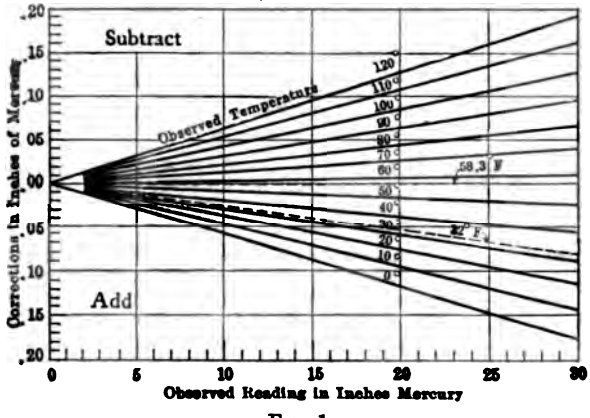


FIG. 1.

The absolute pressure in inches or millimeters of mercury in the enclosed space is obtained by subtracting the vacuum as obtained above from a barometer reading in inches or millimeters of mercury, taken simultaneously at the same place. Absolute pressure may be expressed in lbs. per sq. in. or head in inches (or millimeters) of mercury at a stated temperature.

Standard atmospheric pressure is equivalent to the height of a mercury column of 760 mm. (=29.921 in.) at sea level, 45 degrees latitude and 32 deg. F. Vacuum readings are generally expressed in inches of mercury referred to 30-in. barometer, which increases the **standard temperature** to 58.4 deg. F. as found from the following equation: $30 = 29.92[1 + a(t - 32)]$ where t = the standard temperature and $a = 0.000101$ = coefficient of cubical expansion of mercury per degree Fahrenheit.

Corrections to mercury column readings **due to temperature** should be made in accordance with Fig. 1, from Ellenwood.

Atmospheric pressure varies with the altitude. The **correction for altitude** at sea level is 0.109 in. of mercury per 100 ft. elevation. This correction is **negligible in marine practice** as there is little difference in sea level with the exception of some inland lakes.

Correction to standard gravity (at 45 degrees latitude) for mercury column readings is given in Table 1.

Table 1. Corrections of Barometer Readings to 45 Degrees Latitude

For these latitudes the correction is to be subtracted, deg.	Barometer readings in inches Hg.			For these latitudes the correction is to be added, deg.
	28-in.	29-in.	30-in.	
0	0.074	0.077	0.080	90
10	0.070	0.073	0.075	80
20	0.057	0.057	0.061	70
30	0.037	0.039	0.040	60
40	0.013	0.013	0.014	50
45	0.000	0.000	0.000	45

Atmospheric pressure is measured by means of **a mercury gauge,** shown in Fig. 2, or by means of an **aneroid barometer,** shown in Fig. 3. The mercury gauge, which is the more reliable, is used on Naval vessels while the aneroid is used in the Merchant Marine.

Vacuum is ordinarily **measured** by means of **dial spring gauges.** For more correct readings, during trials and tests, **mercury columns** are employed. Due to the movements of the ship mercury columns surge and therefore, are impracticable, unless used in the form of a short absolute pressure gauge as illustrated in Fig. 4.

Absolute pressure gauges should be constantly checked as even with the most careful handling air penetrates the mercury and in that case the readings are higher than the actual results obtained. Boiling the mercury to purge it of air and readjustment of the gauge scales becomes frequently necessary.

Necessity and Advantage of High Vacua. Steam exhausting to the atmosphere has to overcome an absolute pressure of 14.7 lbs. per sq. in. When exhausting into a vacuum the drop from the initial pressure is increased over that which occurs when exhausting to the atmosphere, and more energy is available. As much work can be extracted theoretically as well as practically from steam of 115 lbs. per sq. in or more, abs. press., below the atmospheric line, as above it. (From Fig. 5, heat drop from 115 lbs. to 14.7 lbs. is equal to the heat drop from 14.7 lbs. to 1 lb. absolute, viz. 150 B.t.u.)

Reciprocating engines cannot utilize more than 26 in. of vacuum because the size and weight of their L.P. cylinders would become impracticable if designed to work at a higher vacuum.

Steam turbines can utilize high vacuum to great advantage because their low pressure ends can be designed to handle large volumes. The reduction

in water rate for increased vacuum is illustrated in Fig. 6 which shows the correction coefficient, for various vacua of a high pressure steam turbine designed to operate at 28 in. vacuum. This is a measure of the gain in the turbine itself due to increased vacuum. Whether or not an actual net gain

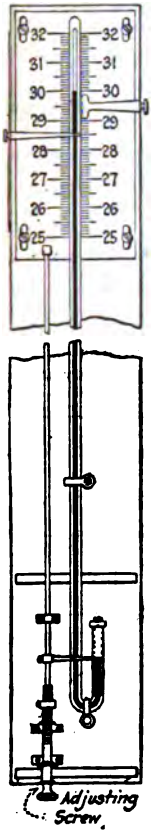


FIG. 2.

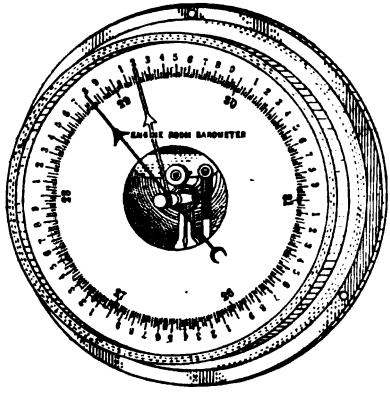


FIG. 3.

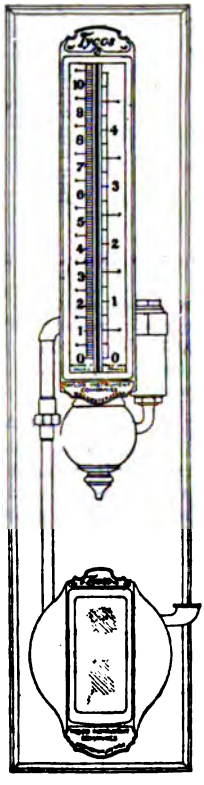


FIG. 4.

in economy is effected depends upon the additional power required for the condenser pumps and the temperature of available cooling water.

Limits of Vacuum. The following recommendations in regard to guaranteed vacuum are considered good practice: **For reciprocating** engines on all Naval, passenger and cargo ships, other than those especially designed to trade in cold latitudes, 25½ to 26 in. vacuum (based on 30-in. bar.) with circulating water at 85 deg. F. As the reciprocating engine cannot utilise

more than this amount of vacuum, carrying a higher vacuum results in keeping the feed water at an unnecessarily low temperature, which in turn results in waste of coal. For ships trading in cold latitudes, 26 in.

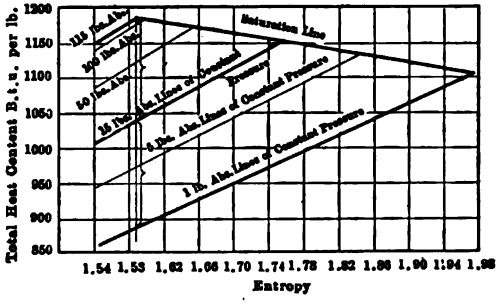


FIG. 5.

with sea water at 60 deg. F. should be used. **For steam turbines** on all types of vessels, 28½ in. with sea water at 60 deg. F., 28.0 in. with sea water at 70 deg. F., 27.6 in. with sea water at 80 deg. F. and 27.25 in. with 85 deg. F. sea water.

Thermodynamics

To condense steam it is necessary to transfer its latent heat. The heat in B.t.u. which has to be absorbed by the cooling water for a given water rate may be determined as follows:

Example: A steam turbine operating with 200 lbs. absolute (185.3 lbs. gauge) pressure, 100 deg. F. superheat, and exhausting into a condenser with 1 lb. abs. press., has a water rate of 12 lbs. per hour per B.H.P. Find total heat per lb. of steam to be absorbed by condensing water.

Solution: From entropy chart, initial total heat = 1257 B.t.u., initial entropy = 1.612.

With adiabatic expansion, the final total heat = 900 B.t.u.; heat supplied per H.P.-hr. = 12 × (1257 − 900) = 4284 B.t.u.; since one H.P. = 2545 B.t.u. per hr., the efficiency ratio $= \frac{2545}{4285} = 59.66$ per cent. Total heat per lb. of steam at end of expansion = 1257 − (0.5966 × 357) = 1043 B.t.u. Heat of liquid (assuming a 10-degree temperature difference) = 59 B.t.u. and heat to be absorbed by condensing water = 1043 − 59 = 984 B.t.u. per lb. steam.

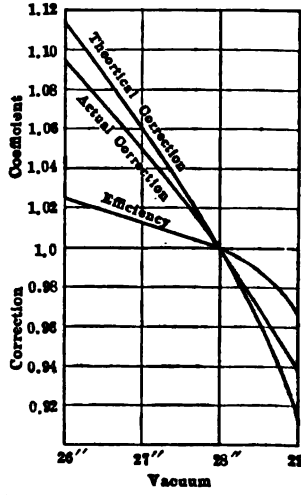


FIG. 6.

Ideal Condenser. A condenser with 100 per cent efficiency would give a vacuum corresponding to the vapor tension of the sea water discharge temperature and a condensate temperature equal to the sea water inlet temperature.

Let G_w = quantity of circulating water in lbs.
G_s = quantity of steam condensed in lbs.
t_1 = sea water inlet temperature.
t_2 = sea water discharge temperature.
t_3 = temperature of condensed steam.
t_4 = temperature of steam entering the condenser..
I = total heat in B.t.u. per lb. of steam entering the condenser.

In an ideal condenser t_2 should be equal to t_1 and $t_4 = t_3$ and

$$G_w(t_4 - t_1) = G_s(I - t_4) \text{ or } G_w = \frac{G_s(I - t_4)}{(t_4 - t_1)}$$

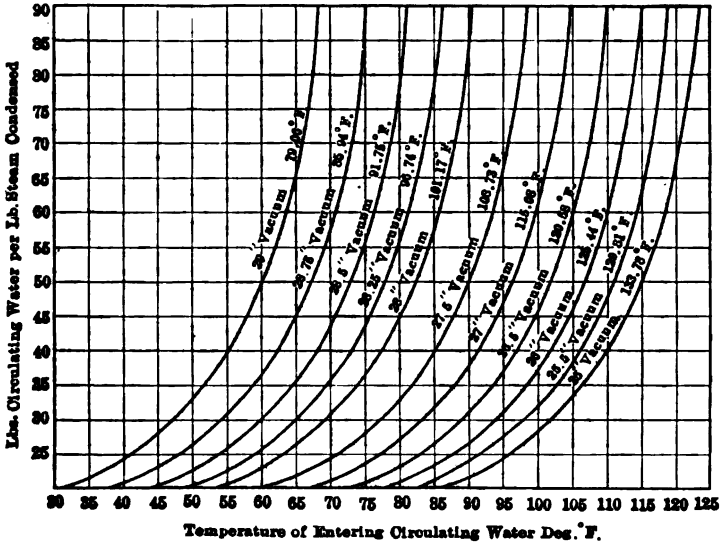


FIG. 7.

The theoretical ratio of circulating water for different vacua and water temperatures is shown in Fig. 7; these curves are based on 950 B.t.u. heat content per lb. of steam condensed. In practice, however, t_2 is always smaller than t_4, and t_2 always larger than t_1. Condensers are designed for a temperature difference between the discharge water (t_2) and the temperature corresponding to the vacuum (t_4). This is called **terminal difference.** This terminal difference is also influenced by the air of which there is always a certain amount present in addition to the vapor

To obtain the **actual ratio of circulating water** required for different vacua and water temperatures from diagram Fig. 7, add the desired terminal difference to the temperature of the injection water and read up from the corresponding point on the base line to the proper vacuum curve. The ratio appears at the left.

Example: 70 deg. F. injection water, 28 in. vacuum and 15 degrees terminal difference. Read up from 85 deg. F. intersecting the 28 in. vacuum curve at a point corresponding to the ratio 59 to 1.

Dalton's Law. In accordance with Dalton's law, the total absolute pressure in a condenser is the sum of the air pressure and vapor pressure, corresponding to the existing temperature, which each of them would exert if occupying the given space alone; $p = p_v + p_a$, where p is total pressure, p_v = vapor pressure, p_a = air pressure.

If V is the total volume, T the absolute temperature, R_1 and R_2 the respective gas constants, M_1 and M_2 the respective weights of the constituents of the mixture, the following formula determines the relations. $V.p = (M_1 \times R_1 + M_2 \times R_2) \times T$.

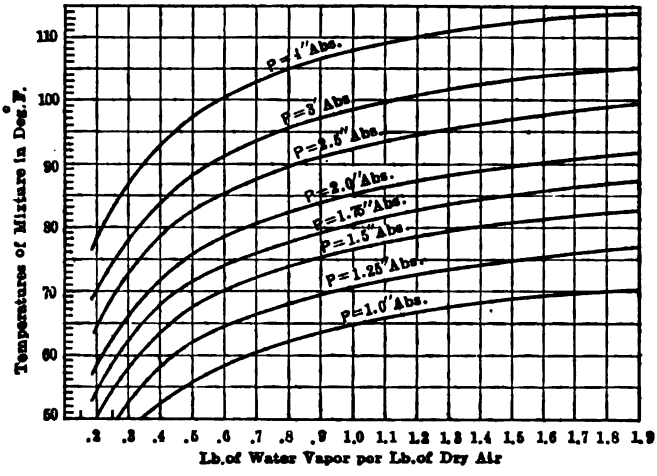


FIG. 8.

In other words, 1 lb. of saturated air contains a certain weight of water vapor, which weight is dependent on the temperature and on the absolute pressure. The higher the temperature the greater is the weight of contained water vapor for the same absolute pressure. As the vapor has to be handled with the air by the air pump, it is desirable to reduce the temperature of the mixture as much as possible so as to lessen the work of the air pump, by partly condensing the water vapor and removing the air in a more rarefied state. This is done by arranging a cooler for the mixture of air and vapor before leaving the condenser

The relations between water vapor and air at different absolute pressures and temperatures are illustrated in Fig. 8 which set of curves can be used to determine the capacity of an air pump for a given air leakage at a desired absolute pressure and temperature of mixture.

Example: Total air leakage in a 2500 S.H.P. Turbine Condenser = 17 lbs. of free air per hr. Temperature of mixture of air and vapor leaving condenser = 80 deg. F. Vacuum 2 in. absolute. To find the capacity of the required air pump read along the temperature line (80 degrees) intersecting the 2 in. vacuum curve, at a point corresponding

to 0.67. Each lb. dry air contains 0.67 lb. vapor, hence the air pump capacity has to be $1.67 \times 17 = 28.4$ lbs. per hr.

Transmission of heat takes place according to a law of proportionality to temperature difference. Under given conditions the **amount of tube surface depends on the rate of heat transmission** which is usually expressed in B. t. u. per hour per sq. ft. per degree F. mean temperature difference. This is known as the "**Heat Transmission Coefficient, K.**"

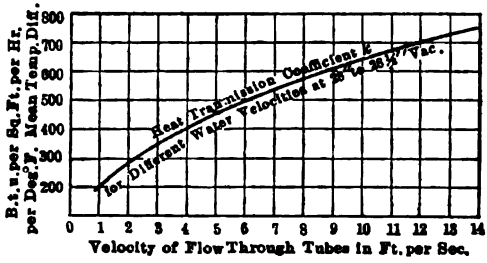

B.t.u. per Sq.Ft. per Hr. per Deg.F. Mean Temp.Diff.
800
700
600
500
400
300
200
Heat Transmission Coefficient K for Different Water Velocities at 28" 28½" Vac.
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14
Velocity of Flow Through Tubes in Ft. per Sec.

FIG. 9.

The resistance to heat transmission comprises three stages:

1st. Steam to external tube surface. As the velocity of the steam is fairly constant, this fraction of total resistance is only more or less influenced by accumulation of air around the tubes, the effective removal of which is desirable.

2d. Resistance through the metal of the tube, which depends upon the conductivity of the tube metal.

3d. Resistance to transmission from internal surface of tube to cooling water. This depends chiefly on the velocity of the cooling water. The viscosity of water is not an important factor in design. Critical velocities also need not be considered as the velocities employed are above the upper critical velocity beyond which all motion is turbulent.

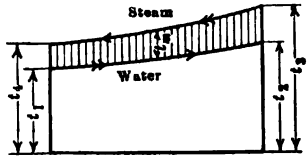


FIG. 10.

The **water velocity is the most important factor on which K depends,** provided the condenser is designed correctly and the proper size of air pump is provided for removing the air effectively. Fig. 9 shows the variation of the heat transmission coefficient for different water velocities for 28 to 28½ in. vacuum. The velocity of injection water in practice varies from 4½ to 7 ft. per second. Six feet is frequently used as it gives a reasonable compromise between a high rate of heat transmission and a moderate expenditure of pumping power.

Hot water will take up heat more rapidly than colder water but for average ship conditions, no accurate figures have been established on a quantitative basis.

The transmission of heat is also directly proportional to the mean temperature difference between the two fluids and according to **Grashof's formula (See Fig. 10) the mean temperature difference,** $t_m = \dfrac{(t_4 - t_1) - (t_3 - t_2)}{\log_e \dfrac{(t_4 - t_1)}{(t_3 - t_2)}}$.

wherein t_1, t_2, etc., have the same meaning as given on page 971.

For the purpose of comparing condenser results this formula is unhandy as t_3 (temperature of condensed steam) varies with different designs of condensers and the assumption now almost universally adopted is that the steam temperature throughout the condenser is the same, viz., that corresponding to the vacuum at the exhaust inlet (t_2). The formula for the mean tempera-

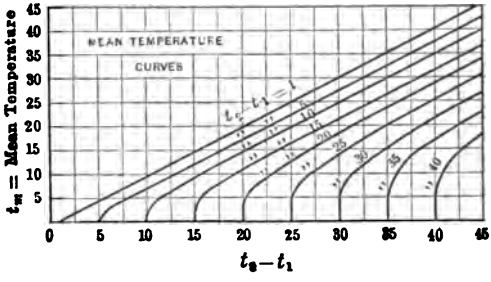


FIG. 11.

ture difference will then be simplified to $t_m = \dfrac{t_2 - t_1}{\log_e \dfrac{t_3 - t_1}{t_3 - t_2}}$. Values of t_m according to this formula can be found directly from Fig. 11.

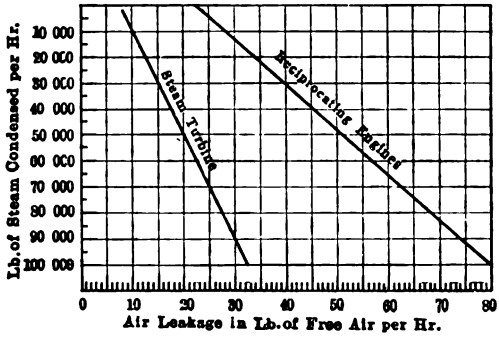


FIG. 12.

Cooling Surface. The surface required per lb. of steam condensed can be found from equation $S = \dfrac{H}{Kt_m}$ in which S = surface in sq. ft., K = heat transmission coefficient in B.t.u., t_m = mean temperature difference in deg. F. and H the total heat to be absorbed per lb. of steam. (See page 973.)

Air Leakage. Air is introduced into the surface condenser in two ways: (1) The air dissolved in the boiler feed water; this quantity is small and seldom exceeds 0.35 lb. per 10,000 lbs. of steam condensed; the hotter the tem-

perature, the less air is dissolved; (2) the air leakage through shaft glands and all joints and parts subjected to vacuum. This quantity forms the greater part, but can be kept reasonably low dependent on the maintenance of the equipment.

The **permissible total air leakage** for marine turbine and reciprocating engine surface condensers for different quantities of steam to be condensed is given in Fig. 12. In **Jet Condensers** additional air is introduced with the injection water, the amount depending on the temperature of injection water. Water at 70 deg. F. dissolves at a maximum about 0.00203 lb. of free air per U. S. gal. at 30 in. bar.

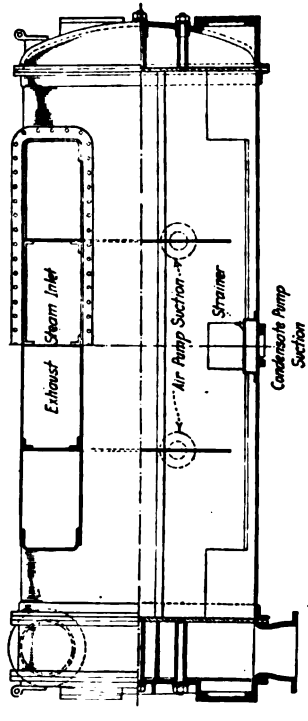
CONDENSERS

Condensers may be divided into two distinct types, **Surface** and **Jet.** In the former, the steam and injection water are separated by a metal tube or plate, while in the latter steam and injection water are intermixed.

Both types may be subdivided into **counterflow** and **parallel flow** types, depending on whether steam and injection flow in opposite or parallel directions.

A further subdivision can be made according to the method of air removal. If both water and air are handled by one pump the condenser is working on the **wet system,** while if separate pumps are employed, it is working on the **dry system.**

The types now most generally adopted, especially for surface condensers, are those working on the counterflow principle and on the dry system. Working on the wet system requires more energy for the operation of the air pump, also it necessitates undercooling of the condensate. Furthermore, where a closed feed water heater is used, compressing an emulsion of air and water, as is done by a wet air pump, is a most effective method of mixing air with condensate, which mixture, if returned directly to the boiler, will cause corrosion.



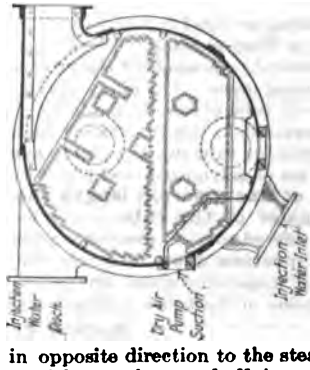


Fig. 13.

Main Surface Condensers

The type mostly used is that of the two water pass design, the water flowing in opposite direction to the steam, the air being removed by the dry system. The condenser **shell** is cylin-

drical in shape with the exhaust opening on the side or on the top, as shown in Fig. 13. It is made of steel or cast iron. A baffle plate is provided over the air suction to cool and condense vapors.

Some condensers are made of wedge or pear-shaped section. While this type maintains a very uniform steam velocity it is an expensive construction.

Tubes are almost invariably ¾ in. O.D. No. 18 B.W.G. for merchant ships and ⅝ in. O.D. No. 16 B.W.G. for naval vessels. The material is seamless drawn brass 70 per cent Cu and 30 per cent Zn, or Admiralty mixture 70 per cent Cu, 29 per cent Zn and 1 per cent Sn.

Tube plates are usually of Muntz metal rolled 60 per cent Cu, and 40 per cent Zn; Admiralty mixture is also used. Tube plates are from $1\frac{5}{16}$ in. to 1¼ in. thick according to the diameter and must be well stayed to prevent collapse. They are held to the shell by alternate collar bolts. By bolting the tube plates to the water covers, particularly to the rib between the two flows near the tube plate centre line an efficient support is obtained.

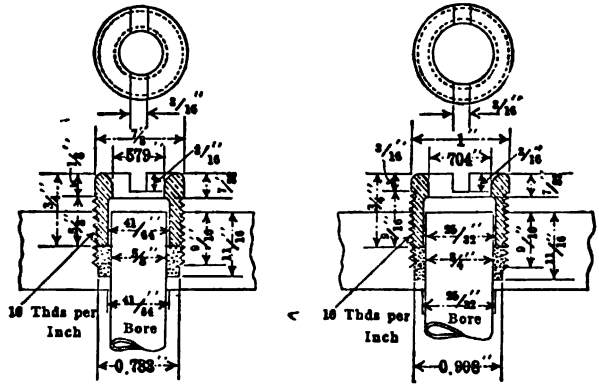


FIG. 14.

Tubes are held in the tube plates by screwed **brass ferrules** and corset lace or fibre packing. Ferrules, Fig. 14, are provided with an internal lip to prevent "creeping" of the tubes and are well rounded at the inlets to reduce frictional losses. On many Naval vessels the **tubes are expanded** in one or both tube plates, in the latter case the tubes have to be curved to allow for expansion, due to change in temperature. **Tubes are supported** by one or more tube support plates, made either of cast iron for Merchant vessels, or brass or galvanized steel plates for Naval vessels, to prevent sagging. **The free or unsupported length should not exceed 100 times the outside diameter of the tube.** The holes in the support plate should be well champfered so as to prevent cutting of the tubes.

Inspection doors of ample size should be provided in each water cover. These inspection doors are generally fitted with **zinc plates** to prevent corrosion of the tubes and tube plates by electrolytic action.

Surface Condenser Design

The principal factor in surface condenser design is the vacuum specified. A high vacuum condenser will require more attention to details than will a

low vacuum or an auxiliary condenser, which is mainly used for regaining fresh water.

The rules governing the design of a high vacuum condenser may be briefly summarized as follows:

1. **Steam** should enter the condenser and be conducted to all parts of the same with the **least possible resistance.** For this purpose an **exhaust opening of liberal size** with a dome practically reaching over the whole length of the condenser shell should be provided. The **velocity of the steam** at the exhaust opening should be from 300 to 400 ft. per second. The **steam path** in the condenser should be straight and of decreasing sectional area so as to maintain a practically uniform steam velocity and thereby prevent stagnation of air. This is done by using differential spacing of the tubes, or by arranging steam lanes, both of which methods reduce the pressure drop through the condenser. For reciprocating engine condensers a perforated baffle plate across the exhaust opening should be provided.

2. **The water path** should have a **uniform cross-sectional area.** The **velocity** chosen will depend largely on conditions relating to each individual installation; it should be borne in mind that the heat transmission rate increases with the velocity, as does also the friction head. The quantity of injection water will also determine the velocity through the tubes and its outlet temperature. The **quantity of injection water** for marine **turbine condensers** is from 60 to 80 times the quantity of condensed steam and for **reciprocating engine condensers** from 35 to 50 times this quantity.

3. **Air** should be **withdrawn** from the condenser at the **lowest possible temperature,** passing over the coldest tubes. The air pump suction should be as far as possible from the exhaust steam inlet and so arranged that "short circuiting" of the steam between these points is avoided. Precaution should be taken to prevent the entrainment of falling condensate.

4. **The condensate should be removed at as high a temperature** as possible and free from air. It is advisable to arrange a **strainer** in the hot well of the condenser.

Spacing of Condenser Tubes. The spacing of condenser tubes depends on the size of condenser. For condensers up to and including 48 in. inside shell diameter, the minimum spacing is $1\frac{5}{16}$ in. for $\frac{5}{8}$-in. tubes and $1\frac{1}{16}$ in. for $\frac{3}{4}$-in. O.D. tubes when using standard ferrules as shown in Fig. 14.

For larger shells up to 6 ft. inside diameter different spacing in the upper and lower bank should be used. In the lower bank not less than 1 in. for $\frac{5}{8}$ in. tubes and not less than $1\frac{1}{8}$ in. for $\frac{3}{4}$ in. tubes should be used.

Table 2. Standard Seamless Drawn Brass Tubes for Surface Condensers

Size of tube, inches O.D.	Square feet of outside surface per lineal foot	Lineal foot per square foot of outside surface	Thickness of tube metal		Weight in pounds	
			B.W.G.	Inches	Per lineal foot	Per square foot outside surface
$\frac{5}{8}$	0.164	6.112	16	0.065	0.42	2.57
			18	0.049	0.327	1.999
$\frac{3}{4}$	0.196	5.093	17	0.058	0.469	2.365
			18	0.049	0.397	2.022

For shells over 6 ft. inside diameter or where the distance from the highest to the lowest tube is more than 6 ft. use three or more different spacings, increasing each ¼ in. over the previous one.

Condenser Tubes. Principal **dimensions** and theoretical **weights** are given in Table 2. Variations of the weights given in Table 2 are to be expected. These seldom exceed 5 per cent. Fig. 15 shows the quantities of injection water in gal. per min. that will flow through different sizes of condenser tubes at various velocities. The **number** of tubes required per water pass for two pass surface condensers with different sizes and lengths of tubes can be found from Figs. 16 and 17.

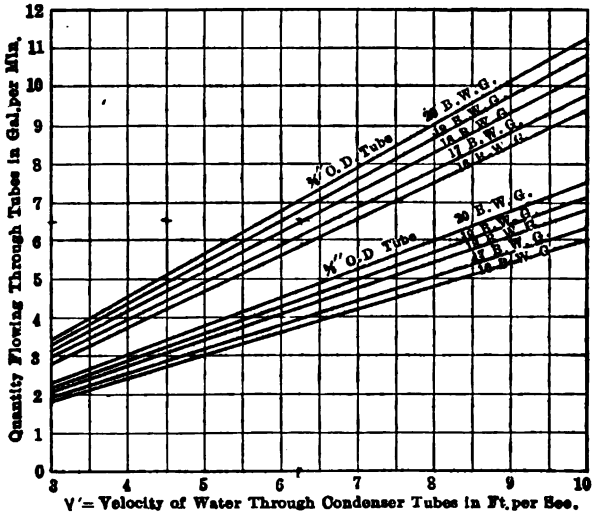

Quantity Flowing Through Tubes in Gal. per Min.
V' = Velocity of Water Through Condenser Tubes in Ft. per Sec.
¾" O.D. Tube
¾" O.D. Tube
20 B.W.G.
18 B.W.G.
16 B.W.G.
14 B.W.G.
18 B.W.G.

FIG. 15.

Condenser Friction. To circulate water through a surface condenser it is necessary to overcome losses in head caused by the friction of the water passing through the tubes, also that caused by the changes of velocity of the water entering and leaving the tubes and the water chambers. These losses vary with the inside diameter and the length of tube, also with the water velocity.

Figs. 18 and 19 show the **total loss through surface condenser** (not including static head due to difference in height of inlet and discharge openings) for 18 B.W.G. ⅝-in. and ¾-in. tubes, of different length and for different water velocities.

To determine the **total head of the circulating pump**, obtain from Figs. 18 and 19 the friction loss through condenser and add the static head as well as the friction head through piping.

Condenser Dimensions and Weights. In marine surface condensers tubes longer than 12 ft. 2 in. are seldom used on account of the limited space for withdrawing the tubes.

Auxiliary Condensers vary from 7 ft. to 10 ft. length between tube sheets, main condensers from 8 ft. to 12 ft. between tube sheets. If made of cast iron the circular shell is never less than 5/8 in. thick, the thickness depending on the size. Circular steel shells are never less than 5/8 in. thick. Steel shells are made up from rolled plates riveted or welded together. Due to the progress made in welding this method is now extensively used.

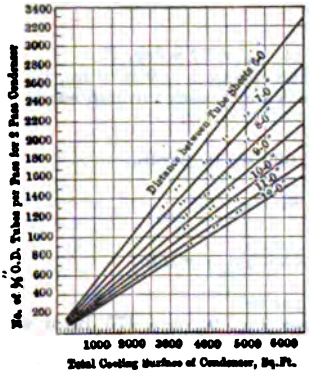


FIG. 16.

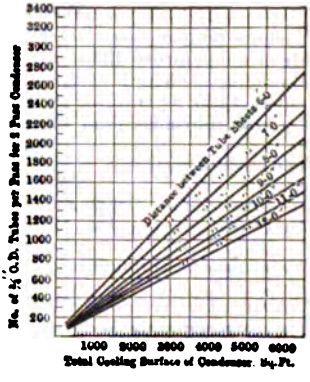


FIG. 17.

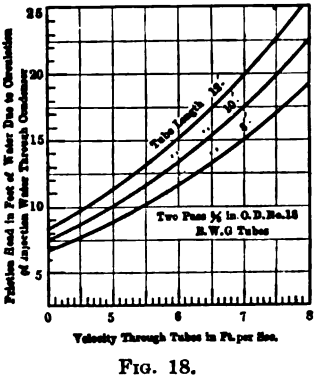

Friction Head in Feet of Water Due to Circulation of Injection Water Through Condenser
Tube Length 12'
10'
8'
Two Pass ¾ in O.D. No. 16 B.W.G Tubes
0 5 6 7 8
Velocity Through Tubes in Ft. per Sec.

FIG. 18.

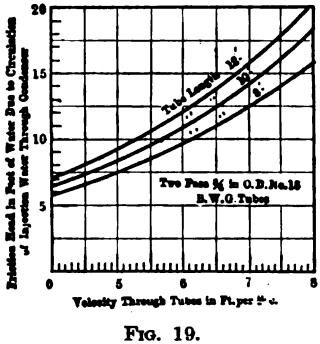

Friction Head in Feet of Water Due to Circulation of Injection Water Through Condenser
Tube Length 12'
10'
8'
Two Pass ¾ in O.D. No. 16 B.W.G. Tubes
0 5 6 7 8
Velocity Through Tubes in Ft. per Sec.

FIG. 19.

The weight of **cast iron surface condensers** varies from 9 lbs. to 6.5 lbs. per sq. ft. cooling surface (the larger figure being applied to small condenser). A 5000 sq. ft. condenser weighs about 36,000 lbs. The weight of a **steel shell condenser** is about 5 to 10 per cent less than that of a cast iron shell condenser of the same capacity.

Example of Surface Condenser Calculation. Determine necessary cooling surface and size of air pump for condensing 36,000 lbs. of steam per

hour at 1.5 in. abs. with 5500 gal. per min. circulating water at 60 deg. F. Steam supplied to turbine at 200 lbs. abs. press. and 100 deg. F. superheat.

Assumed pressure drop through condenser = 0.1 in. Hg.

Vacuum to be produced 1.4 in. Hg. abs. = 28.6 in. Hg. (30 in. bar.).

Heat to be absorbed by circulating water (see thermodynamics, page 970) is found to be 986 B.t.u. per pound of steam.

Temperature rise $t_r = \frac{986 \times 36{,}000}{5500 \times 500} = 13$ deg.F.

Vacuum temperature $t_3 = 90$ deg. F.

Mean temperature difference (from Fig. 11) $t_m = 22.8$ deg. F.

Assuming the length of tubes between tube sheets, $l = 9$ ft. 0 in., the size of tubes ¾ in. O.D., No. 18 B.W.G.; two water passes through condenser, and a heat transmission of 450 B.t.u.

The cooling surface $S = \frac{986}{450 \times 22.8} = 0.096$ sq. ft. per lb.

Total surface = 36,000 × 0.096 = 3460 sq. ft.

Tubes per pass (see Fig. 17) $n = 985$.

Water passing through tube $q = \frac{5500}{985} = 5.6$ gal. per min.

Velocity of water (see Fig. 15) $v = 5.4$ ft. per sec.

The corresponding heat transmission (see Fig. 9), $k = 475$ B.t.u. which would indicate a safety margin of 25 B.t.u. per lb. over the 450 B.t.u. assumed.

From the steam table the specific volume of steam at 1.4 in. abs. press. = 469 cu. ft.

Assuming 350 ft. per second exhaust steam velocity, the area of the exhaust opening $A = \frac{36{,}000 \times 469}{3600 \times 350} = 13.4$ sq. ft.

These conditions are satisfied by an opening 20 in. wide and 8 ft. 0 in. long.

Suction and discharge nozzles will be 15 in. diameter.

Dry air pump suction 5 in. diameter.

Condensate pump suction 6 in. diameter.

Condenser friction (see Fig. 19) = 9 ft.

Air leakage (from Fig. 12) = 16.5 lbs.

Assuming an air temperature of 70 deg., each pound of dry air at 1.4 in. Hg. abs. press. contains 0.7 lb. of vapors (see Fig. 8); hence total air pump capacity = 1.7 × 16.5 = 28 lbs. per hour.

Auxiliary Condensers

While in port, the steam exhausted by auxiliaries such as generators, winches circulating and other pumps is generally condensed by an auxiliary condenser. According to the degree of vacuum, three types are known:

1st. High Vacuum Auxiliary Condensers which serve steam turbine driven generators and are designed for a vacuum ranging from 26 to 28 in. The principles of their design are similar to those given on p. 975, see "Main Surface Condensers."

2d. Low Vacuum Auxiliary Condensers which condense steam from reciprocating engine driven generators, winches, pumps and other auxiliaries and are seldom designed for more than 20 in. vacuum. This type consists essentially of a surface condenser mounted directly over a reciprocating direct acting combined air and circulating pump. Such a condenser is shown in Fig. 20.

3d. Atmospheric Auxiliary Condenser. These are used for the same purposes as the low vacuum auxiliary condensers. The only difference being that no air pump is used, the condensate draining by gravity into the feed and filter tank. Draining by gravity requires a high location of the condenser. The cooling surface can be considerably less than that of a low vacuum condenser for the same amount of steam condensed. A direct-acting duplex pump generally supplies the circulating water to the atmospheric auxiliary condenser.

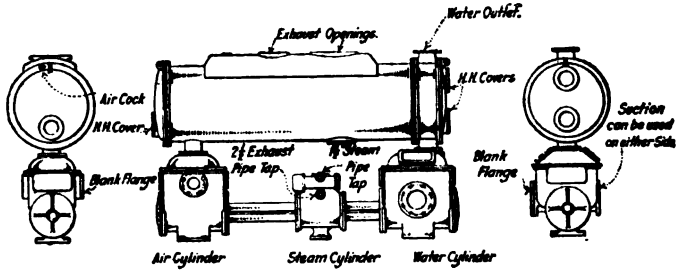


FIG. 20.

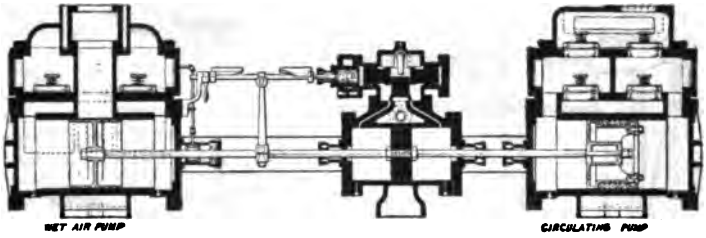


FIG. 20a.

Table 3. Standard Low Vacuum Auxiliary Surface Condenser

Cooling surface, sq. ft.	Tubes		Cir. air and cir. pump				Max. amount of steam condensed per hr., with cir. water at 85 deg. F.		Cir. water, gal. per min.	Diameter				Approx. weight, lbs.
	O. D., in.	Length, ft.—in.	Steam cyl., diam., in.	Cir. pump cyl. diam., in.	Air pump cyl. diam., in.	Stroke, in.	20-in. vacuum	Atmos. pressure		Exhaust outlet, in.	Cir. water		Air pump discharge, in.	
											Inlet, in.	Discharge, in.		
600	⅝	8-2	10	12	12	12	6,000	10,000	325	8	6	5	4	9,500
900	⅝	8-2	10	14	14	12	9,000	15,000	475	10	8	6	5	12,000
1,200	⅝	10-2	12	16	16	16	12,000	20,000	625	12	10	8	6	15,000
1,500	⅝	10-2	12	18	18	18	15,000	25,000	780	14	10	8	6	20,000

Low vacuum auxiliary condensers should be designed along the following lines:

The **water velocity** through the tubes should be fairly high. Too low a water velocity will greatly accelerate the scaling of the tubes when the ship is passing through tropical waters.

The **cooling surface** should not be too large as experience has shown that a large cooling surface is no protection against a quick deposit of oil, as is often assumed.

Exhaust inlets should be provided with **baffles** to prevent the impinging of the steam against the tubes.

A soda cock and **boiling-out valve** should be provided. The boiling out of the auxiliary condenser is more frequently necessary on account of the considerable amount of lubricating oil from the auxiliaries that is carried over by the steam.

Some manufacturers provide **filter chambers** at the bottom of the auxiliary condenser. This is **not recommended**, as cleaning these filter chambers necessitates a shut-down of the condenser. Furthermore, the condensate is more effectively filtered without interruption of service in the feed and filter tank.

Materials. Auxiliary Condensers are generally made of cast iron shells with cast iron water boxes and covers. The tubes are generally of $\frac{5}{8}$ in. O.D. No. 18 B.W.G. Table No. 3 shows some standard sizes as used in the United States.

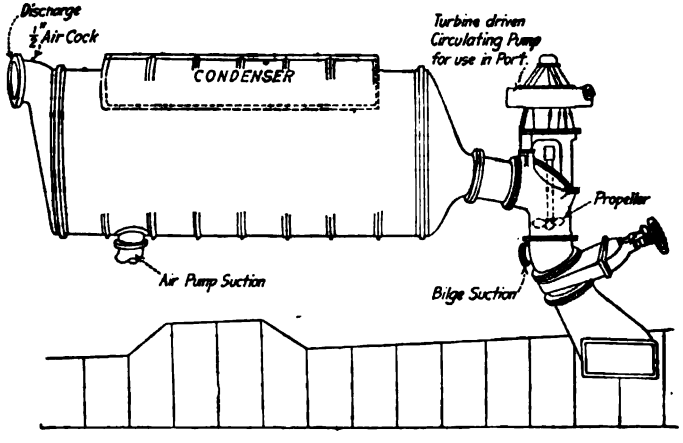


FIG. 21.

Scoop Condensers

Scoop Condensers are frequently used on fast vessels, particularly on destroyers and fast Yachts. Their design does not differ materially from the others, except that they use only one water pass.

The injection water is circulated through the condenser by the pressure created, while the vessel is moving fast, by a scoop arranged on the outside

of the hull below the water line, *i.e.*, the water is scooped through the condenser.

A small circulating pump has to be provided to circulate sufficient water through the condenser when the vessel moves slowly or lays in port.

The arrangement of a scoop condenser with the auxiliary circulating pump is shown in Fig. 21

Keel Condensers

Keel condensers are used on small passenger vessels running in shallow waters, also on launches, tugs and lighters. They consist of pipes arranged on the outside of the hull near the keel, into which the exhaust steam is discharged. (See Fig. 22.) A wet air pump is used in connection with same.

The pipes are of steel or iron if the vessel runs in salt water, except where the vessel is copper sheathed or has a bronze outboard bearing.

Copper or brass pipes may be used on vessels running in fresh water; or if running in salt water, copper pipes are permissible only if the propeller wheel and the tail shaft with its bearing are made of bronze. Otherwise destructive galvanic action will be experienced.

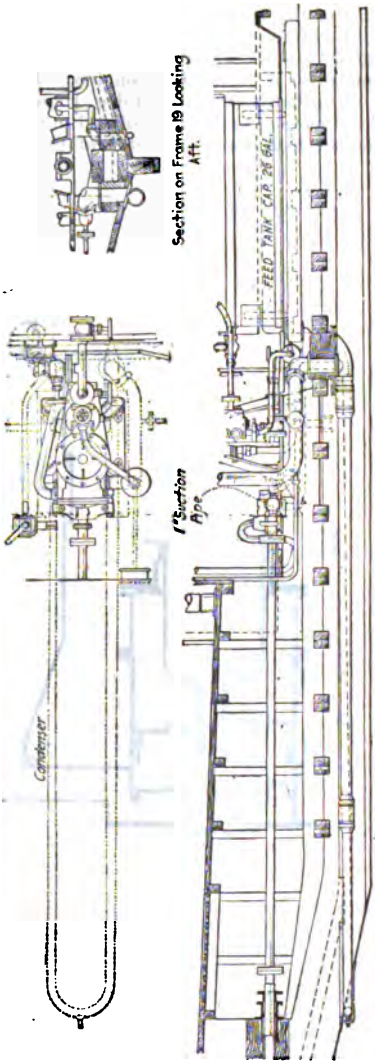


FIG. 22.

Jet Condensers

Jet Condensers are installed only in vessels running on fresh water. Steamers equipped with Jet Condensers cannot be used on the ocean or in brackish water, as the condensed steam would be lost for boiler feeding.

The type of Jet Condenser used is that of the parallel flow low-level jet. It consists of a cone or cylindrical shaped chamber in which the exhaust steam enters at the top and is mingled with the cold injection water, also entering at the top; condensed steam and injection water fall together to the bottom of the condenser and are removed either by a wet air pump or

by a centrifugal pump. In the latter arrangement a separate dry air pump removes the air and non-condensible gases.

Some of the water removed from the condenser is used for boiler feeding, while the excess is discharged overboard.

Fig. 23 shows a section through a Jet Condenser with single acting wet air pump for a 1600 I.H.P. engine. This condenser is designed for 22 in. with 30 in. barometer and 60 deg. F. injection water temperature.

The injection water is regulated entirely through the spray valve in the top of the injection pipe inside of the condenser. The quantity of water

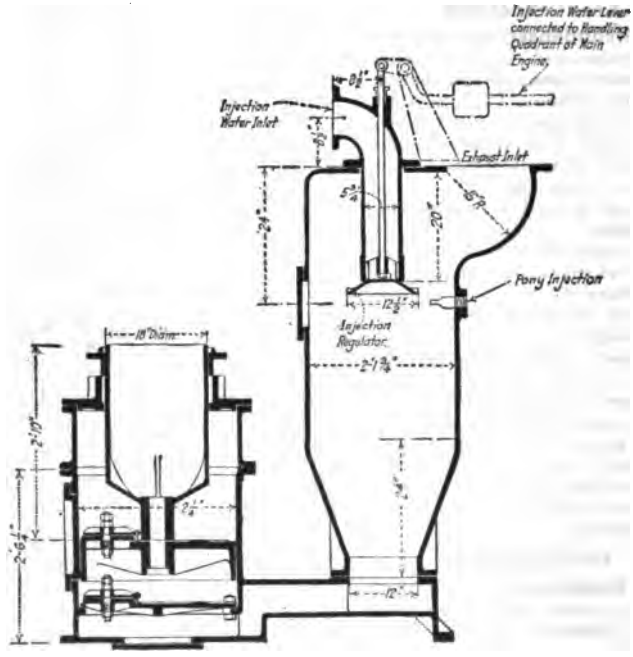


FIG. 23.

required rises and falls with the speed of the engine, and the air pump displacement is likewise controlled by the speed of the engine.

It is not customary to provide vacuum breakers on jet condensers of this type as the engineers are trained to immediately close the injection control valve (which is connected to the handling quadrant of the main engine) when the engine is shut down or even a moment or two before. The buckets of the air pump are not packed, their tightness depending entirely on the film of water between the bucket and barrel. In handling fresh water, there is always a large amount of sand and grit passing through the air pump particularly when working around the harbor, and it has been found from

experience that packed pumps do not give a satisfactory service, as the wearing of the pump barrel is so very rapid and the pump barrel requires frequent relining, which owing to the size of the bucket is very expensive.

Figure 24 shows the arrangement of a jet, condenser and air pump of this type.

During the last few years considerable progress has been made in design of **jet condensers for high vacuum.** The difficulties which the wet air

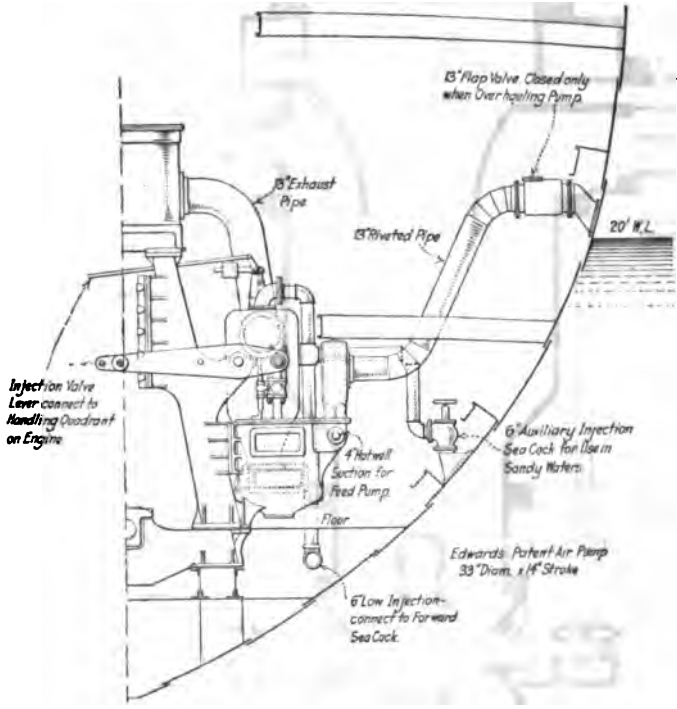

13" Flap Valve Closed only when Overhauling Pump
8" Exhaust Pipe
13" Riveted Pipe
20' W.L.
Injection Valve Lever connect to Handling Quadrant on Engine
4" Hotwell Suction for Feed Pump
6" Auxiliary Injection Sea Cock for Use in Sandy Waters
Floor
Edwards Patent Air Pump 33" Diam. x 14" Stroke
6" Low Injection connect to Forward Sea Cock

FIG. 24.

pump offers when being used for **removing the injection water** and condensed steam, have been overcome by handling the air and water separately by different pumps. The **centrifugal pump,** is used for removing the injection water. It **has larger flexibility as regards to variations in quantity and head,** and it suffers less than the reciprocating pumps when handling gritty water. **The air pump most suitable** for the use in connection with jet condensers **is the steam operated air ejector.** (See Air Pumps, p. 990.) Both pumps work independent from the main engine, whether reciprocating engine or steam turbine. This arrangement has the advantage of maintaining a constant vacuum whether the main engine runs slow or at full speed. It also permits easier and quicker starting. The

initial cost of the installation, as well as the cost of operation are considerable less than the wet type air pump connected directly to the crosshead of the main engine.

A jet condenser with steam ejector air pump and turbine driven removal pump is shown in cross section in Fig. 25.

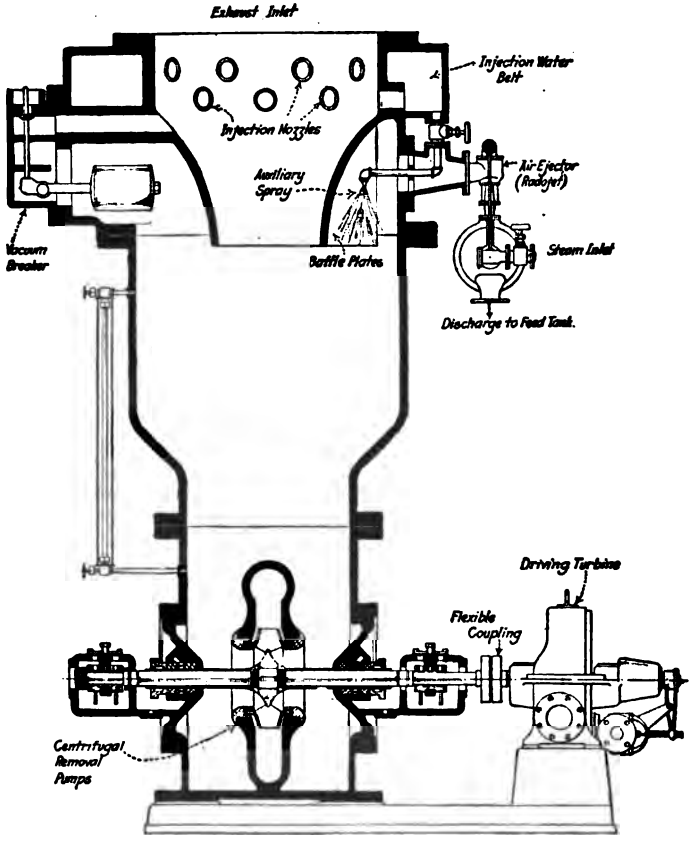


FIG. 25.

The **injection water** is lifted to the condenser by the difference in pressure between the atmosphere and that existing in the condenser and is distributed around the entire circumference through annular opening and enters the condenser head through distributing nozzles.

These nozzles break the water up into a fine spray so that it mixes intimately with the steam which enters through the exhaust opening. Hand holes are provided to permit occasional cleaning of these nozzles.

The mixture of condensed steam and water falls to the bottom of the condenser and is discharged against any external head by the double suction centrifugal pump . . . .

The **air and non-condensible gases** pass through an additional spray arranged ahead of the air suction opening, and are taken away by a steam air ejector.

A **Vacuum Breaker** operated by a float is arranged in the upper part of the condenser; it admits air directly into the condensing space and stops the inflow of water in case of accidental stopping of the centrifugal pump. A float controlled injection valve is sometimes provided in addition to the vacuum breaker.

Jet Condenser Design

The following should be observed in designing high vacuum Jet Condensers:

Steam should enter the condenser with a velocity of 300 to 400 ft. per sec. Sufficient condensing space should be provided to permit a thorough condensation without a pressure drop from the air pump suction to the exhaust steam inlet.

Water should enter the condenser in form of a fine spray so as to permit a rapid and intimate mixing with the steam. As the injection water enters by virtue of the vacuum the total suction head, including friction loss through piping to the inlet of the nozzles should not exceed 14 ft. for a vacuum of 6 in. Hg. absolute or 18 ft. for a vacuum of 2 in. Hg. absolute or less at sea level. For every 1000 ft. elevation above sea level, the suction lift should be reduced one foot. The velocity of water in the injection pipe should not exceed 12 ft. per sec.

Air should be withdrawn from the condenser at the lowest possible temperature. An additional injection water spray should be arranged ahead of the air suction opening. The entrainment or mingling of falling water with the air should be prevented.

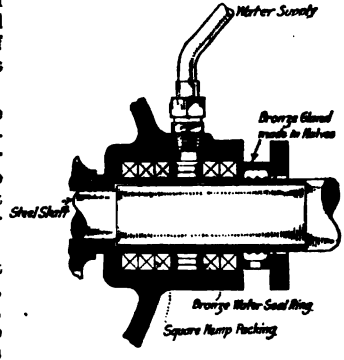


FIG. 26.

The Condenser Shells and most other parts are usually made of cast iron, with the exception of the pump impeller, impeller casing wearing rings and the shaft covers, which are made of brass or bronze. **Stuffing Boxes** of the centrifugal pump shaft have to be water-sealed, to insure absolute air tightness, Fig. 26. Water is supplied from the discharge of the pump or some other source. **Manholes** for the inspection of the nozzles and the interior of the condenser should be provided.

The Size of the Marine Jet Condenser is determined by the amount of injection water required for the condensation of steam. The amount of injection water can be found by referring to Fig. 7 (giving the theoretical ratios of injection water for different vacua and water temperatures) and using the terminal temperature differences for Marine Jet Condensers, given below for determining the actual ratio n.

Vacuum in in., Hg. abs.	1½	2	3	4
Terminal difference, Deg. F.	12	9	7.5	6

The removal pump has to be designed to remove $(n + 1)$ times the amount of steam against a total head of $H = h_1 - h_2 + h_3$, wherein

h_1 = head corresponding to vacuum.

h_2 = height of water level inside of condenser above centre of removal pump.

h_3 = external static head plus head due to friction losses in piping, etc.

The size of air pump is found in the same way as described under surface condensers, but it should be borne in mind that about 0.00203 lb. of air per each gallon of injection water at 70 deg. F. must be added to the air leakage, before applying the Dalton law.

The weights of Jet Condensers with Centrifugal Removal pumps and steam ejector air pumps but without driving turbines are about as follows:

Quantity of injection water, gal. per min.....	500	1000	2500	5000
Weight of condenser in lbs. per 1 gal. per min.....	8	6.5	4	3.25

Example of Jet Condenser Calculation. Determine size of Jet Condenser and its air pumps for condensing 36,000 lbs. of steam per hr. at 1.5 in. abs. with injection water at 60 deg. F. Steam is supplied to main turbine at 200 lbs. abs. press. and 100 deg. F. Superheat.

Assumed pressure drop through condenser = 0.05 in. Hg.

Vacuum to be produced = 1.45 in. Hg. abs. = 28.55 in. Hg. with 30-in. Barometer.

Heat to be absorbed by injection water = 986 B.t.u.

Terminal temperature difference = 12 deg. F.

Temperature corresponding to 1.45 in. Hg. absolute = 91 deg. F.

Temperature of injection water discharge = 79 deg. F.

Temperature rise = 19 deg. F.

Ratio of injection water to steam = 986 : 19 = 52 times.

Quantity of injection water = 36,000 × 52 = 1,872,000 lbs. per hr.

Capacity of removal pump = 1,872,000 + 36,000 = 1,908,000 lbs. per hr. or 3816 gal. per min.

Sect. Area of Injection Pipe = 1,872,000 + (3600 × 62.5 × 12) = 0.69 sq. ft.

Diameter of Injection Pipe = 0.94 ft. or about 12 in.

Area of Exhaust Opening = 13 sq. ft.

Diameter of Exhaust Opening = 48 in.

Amount of air entering the condenser = 16.5 + (0.00203 × 22,896) = 63 lbs. per hr.

Assuming an air temperature of 70 deg. F., each pound of dry air at 1.45 in. Hg. abs. contains 0.63 lb. of vapors.

The total air pump capacity = 1.62 × 63 = 102 lbs. per hr.

CIRCULATING PUMPS

The centrifugal type has supplanted all other types of main circulating pumps on account of its compactness, absence of reciprocating parts, simplicity and greater flexibility. Only on small auxiliary condensers direct acting reciprocating piston pumps combined with a wet air pump located underneath the condenser and supporting same are used.

Centrifugal Circulating pumps (see also p. 1123) on board ship are of the **low pressure** type. The total head seldom is more than 30 ft. Their

impellers are generally of the **double-inlet** design. According to the speed they are classified as **slow and high speed** pumps.

Slow speed pumps are directly reciprocating engine driven and run from 150 to 300 R.P.M. Their efficiencies range from 65 to 80 per cent.

High speed pumps are driven by steam turbines or electric motors, either directly or with gear reduction, and run from 2000 R.P.M. to 350 R.P.M. according to the size and head. The efficiencies of these pumps

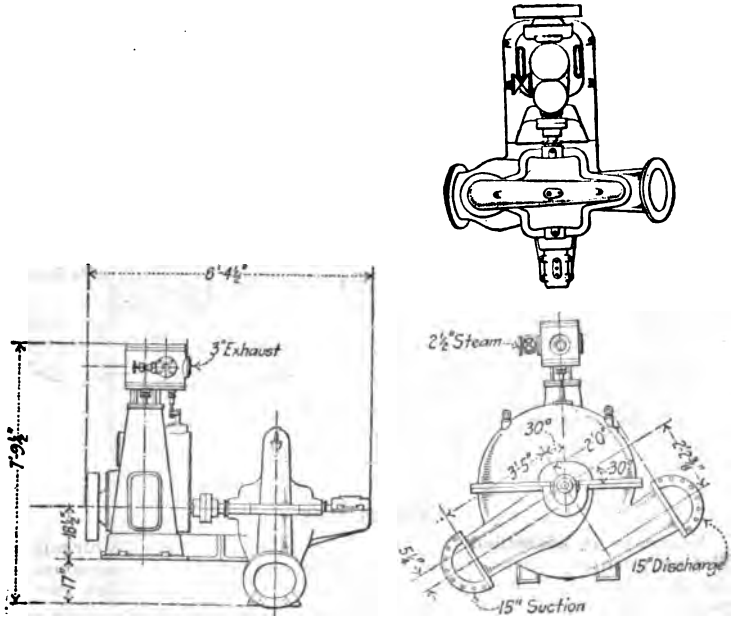


FIG. 27.

vary from 50 to 75 per cent depending on head, quantity and speed. Both types have a horizontally split casing made of cast iron, the upper half of which can be removed without breaking the suction or discharge connections.

Brass or bronze casing wearing rings are arranged inside the casing. The impeller is made of bronze; the shaft of tobin bronze or steel with bronze lining. The stuffing boxes are water sealed. Vent and drain cocks are provided on the casing. Pumps are designed for a maximum quantity and a maximum head. Variations in both are controlled by speed regulation by means of a throttle valve, which is provided instead of a speed governor. On steam turbine driven pumps a safety trip governor is provided in addition to the throttle to prevent overspeeding.

The velocity of water in the suction and discharge pipes should not be more than 10 ft. per sec. for long pipe lines containing many bends, and not more than 12 ft. per sec. for short straight pipe lines.

Fig. 27 shows a reciprocating engine driven circulating pump having a capacity of 5500 gal. per min. against a 30 ft. head.

Fig. 28 illustrates a steam turbine gear driven pump of 7500 gal. per min. against a 30 ft. head.

A direct acting circulating pump as used in connection with auxiliary condenser is shown in Fig. 20*a*, p. 981.

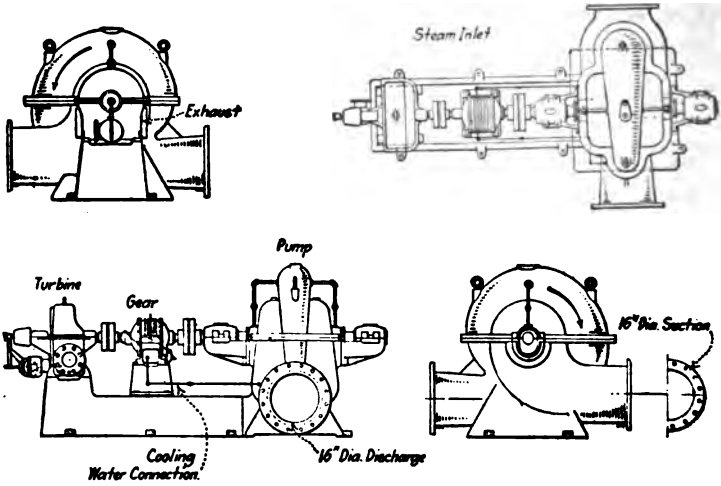


FIG. 28.

AIR PUMPS

Air pumps are **classified** into two distinct groups, wet and dry air pumps. **Wet air pumps** remove both air and condensate, while dry air pumps remove the air and vapors only. Wet air pumps are not suitable for high vacuum requirements because the temperature of the condensate must be considerably lower than the theoretical temperature due to the vacuum, which requires cooling of the condensate, and in turn the loss of B.t.u. in the feed water. Compressing the emulsion of air and water by a wet air pump is a very effective method of aerating the feed water, which will cause corrosion in the boilers. If noiseless and satisfactory operation of the wet air pump is desired, the pump must be located close to the condenser, not less than 3 ft. below the bottom of same; its discharge head must not be more than 5 ft. above the discharge valves and the operating piston speed should not be more than 60 ft. per min. The steam consumption of the reciprocating wet air pump at high vacuum, based on actual air handling capacity, is considerably higher than that of the dry air pump and its co-working condensate pump together.

The **dry air pump** will deal with the air and vapors at a temperature close to that of the injection water inlet and it will enable the condensate pump to handle the condensate at about the vacuum temperature.

In good modern practice, **independent air pumps** are used exclusively, due to the many advantages they possess over attached air pumps.

The independent air pump admits of a vacuum being formed and retained in the condenser before the main engines are started, and at any time when they are stopped, which enables the engines to be manipulated with greater ease and certainty, particularly at times when quick response is of prime importance. The speed of the independent air pump can be regulated according to the requirements; also loss of feed water is avoided as it continuously drains the main engine cylinders. The **attached air pump** is required to work at the same speed as the main engine, regardless of the work it has to do.

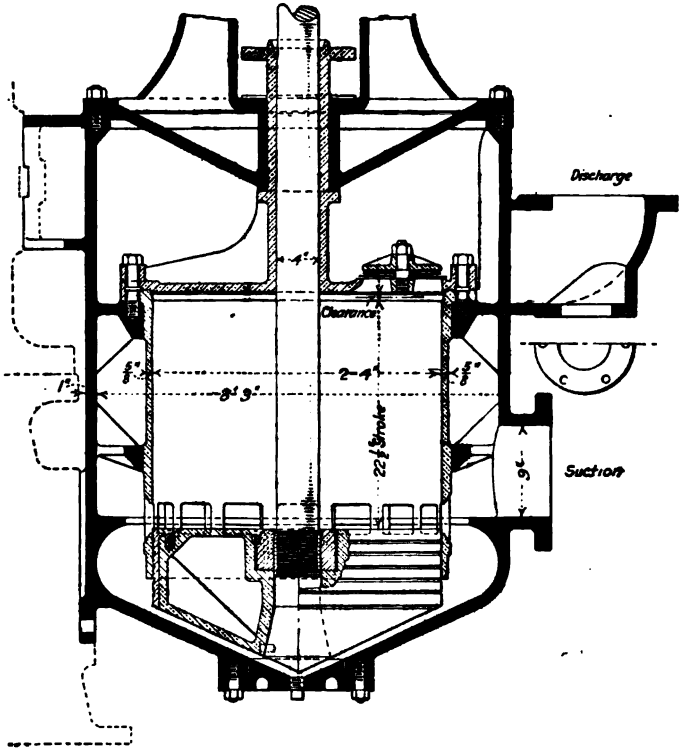


FIG. 29.

Wet Air Pumps

Wet Air Pumps are of either the reciprocating or rotative type. According to the location of their shaft, they are either vertical or horizontal. Horizontal reciprocating wet air pumps are generally double-acting, while those of the vertical type are mostly single-acting. Fig. 29 shows an **Edwards single vertical air pump**; it has no suction valves. The discharge valves on the

top of the cylinders check the discharge water and air from returning to the pump on the down stroke.

Fig. 30 shows **attached type wet air pump** with three sets of valves; foot valves, bucket valves, and head valves.

Fig. 31 shows a wet air pump of the **vertical twin beam type.** This pump is single direct acting and has either one or two steam cylinders. A beam connects the two main piston rods and operates the auxiliary valve gear. A wet air pump with horizontal shaft is shown in Fig. 20a.

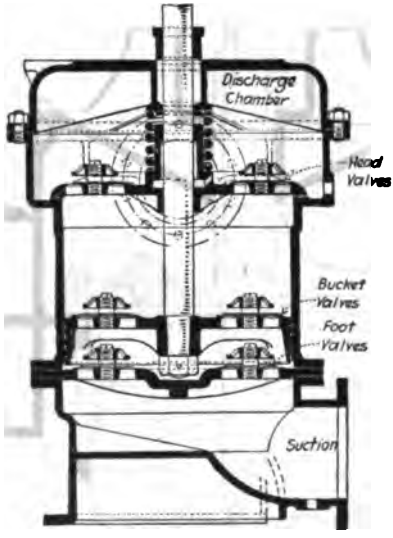


FIG. 30.

Fig. 32 shows a cross section through a wet air pump of the rotary type known as **Rotrex Pump.** It is a positive displacement pump; the cylinder rotor being mounted eccentrically on a shaft and an oscillating cam operated from the rotor shaft by means of cranks and connecting rods, separating the discharge and suction side of the pump cylinder, at the same time following the motion of the rotor with a small clearance. All clearances are water sealed, the rotor at no time being in contact with the cylinder or cam.

Dry Air Pumps

Dry Air Pumps are either of the reciprocating, rotary hydraulic entrainment or steam ejector type. According to the location of their shaft the first two are either horisontal or vertical.

Reciprocating Dry Air Pumps are of the double-acting slide valve type fitted with an equalizing port, arranged to connect the two ends of the cylinder at the completion of each stroke after closing the suction port, thus almost eliminating the effect of clearances and preventing loss of efficiency which

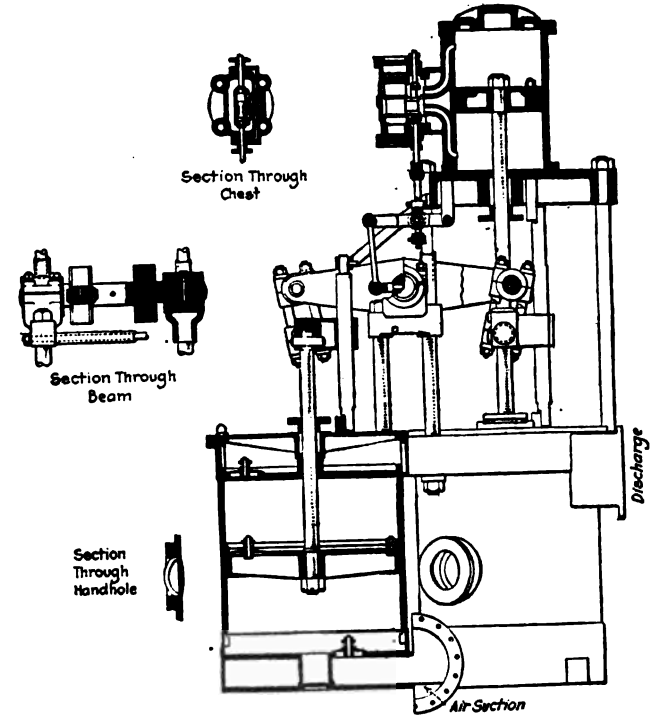

Section Through Chest
Section Through Beam
Section Through Handhole
Discharge
Air Suction

FIG. 31.

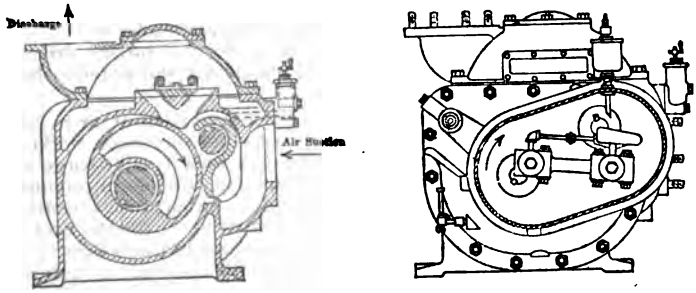

Discharge
Air Suction

FIG. 32.

would otherwise result from re-expansion at the beginning of the suction stroke. Pumps of this type have a fairly high volumetric efficiency, when designed for small clearances. The efficiency is greatly affected by wear and also by vapors entering the pump cylinder. Particular care must be taken to prevent water from entering the pump which due to the small clearance spaces would cause breakage. Due to this the pump is seldom used, and installations of the same are only found on naval vessels where there is a well-trained engineering crew capable of taking care of the fine adjustments and continuous attention which it requires.

Rotary Hydraulic Entrainment Pumps use water as an air entraining medium, this water being either broken up into fine sheets or films or being discharged as a continuous annular film by the centrifugal action of an impeller.

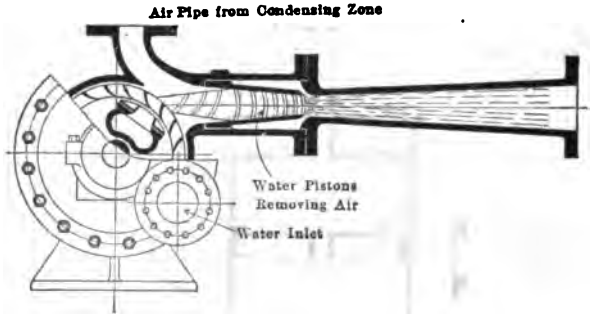


FIG. 33.

Fig. 33 illustrates the action of the **Le Blanc air pump.** Sealing water is admitted at the centre, enters the rotating veins through a stationary nozzle and is then discharged in the form of thin square sheets into the diffuser, entrapping the air between these sheets and compressing it to atmospheric pressure. The sealing water has to be fresh water, as otherwise the vapors withdrawn with the air from the condenser would be lost. The sealing water passes through an air separating tank and cooler and is reused.

Fig. 34 shows the general arrangement of the vertical turbine driven Le Blanc air pump with condensate pump, air separator and water cooler, all combined in one unit.

Air Ejectors depend for their action on the entrainment of air by steam jets, and consist of two steam ejectors, the first and second stage, working in series. They have no moving parts, are light in weight and occupy small space (about $\frac{1}{4}$ to $\frac{1}{10}$ of that of the twin beam air pump), require no foundations, no lubrication or attention during operation; their operation is noiseless and simple and the steam consumption is low. All the heat contained in the steam used in the air ejector is returned to the boilers by condensing the steam used with the feed water. All these advantages make the steam air ejector a desirable air pump for marine installations, whether for main turbines or reciprocating engines, and it is gradually displacing the other types of dry air pumps.

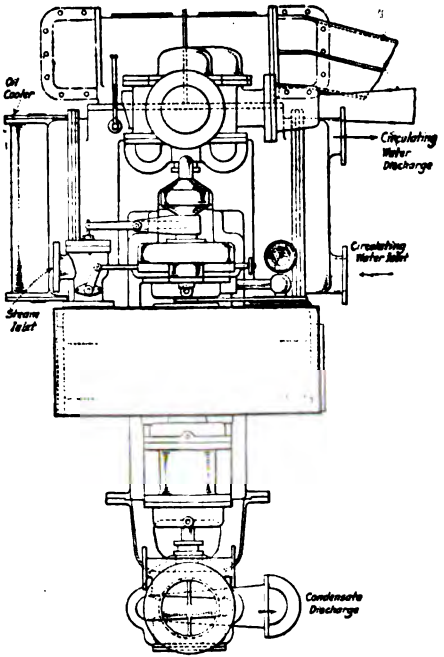


FIG. 34a.

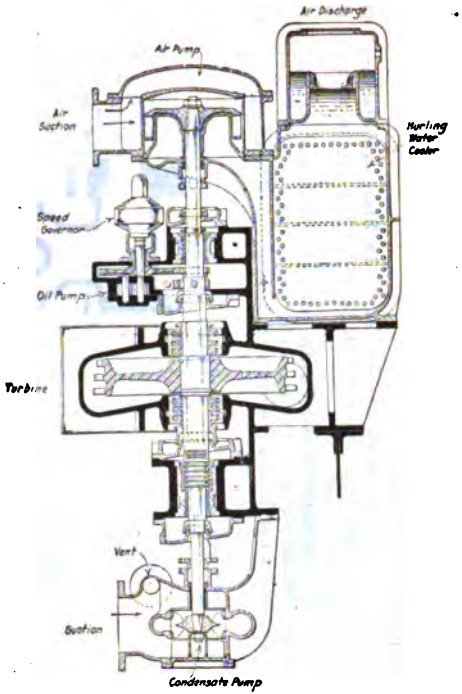


FIG. 34b.

Steam Air Ejectors, due to their light weight, are generally supplied in pairs for marine condensers, this division, having the advantage of reducing the steam consumption by operating with one ejector under light loads or when the installation is air tight, also providing an additional capacity in case of unexpected leakage.

Fig. 35 shows a sectional arrangement of the **Le Blanc air ejector.** The steam nozzles in both stages are of the De Laval type and are arranged in the same axis with the diffuser.

Fig. 36 shows the **Radojet Air Pump,** which has its name from the radial flow nozzle used in the second stage. This nozzle consists of two disks; one stationary, called nozzle and the other adjustable, called nozzle point. The

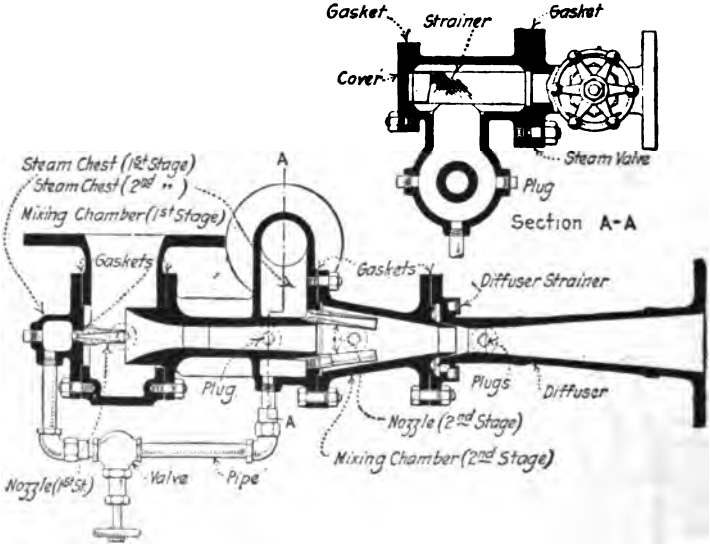


FIG. 35.

nozzle contains the throat. By changing the distance between the nozzle and the nozzle point expansion of the steam can be varied without changing the steam consumption. The annular steam jet has a greater penetrating force and its entrainment surface is larger than that of a cylindrical jet of equal length, consequently better efficiencies are obtained. Radojets are designed for a minimum steam pressure of 105 to 110 lbs. gage. A reducing valve set for about 5 lbs. above the minimum steam pressure is arranged in the steam supply line to insure practically constant pressure at the inlet of the Radojet with variations in the boiler pressure.

Steam air ejectors are also built with either jet or surface condensers between first and second stage. In this arrangement the steam used in the first stage is condensed by the inter-condenser and the second stage has to

remove air and vapors only. While a saving in steam consumption is effected, the heat contained in the steam used in the first stage is lost, and this arrangement also lacks all the advantages of the straight air ejector, such as simplicity, light weight, small space required, etc.

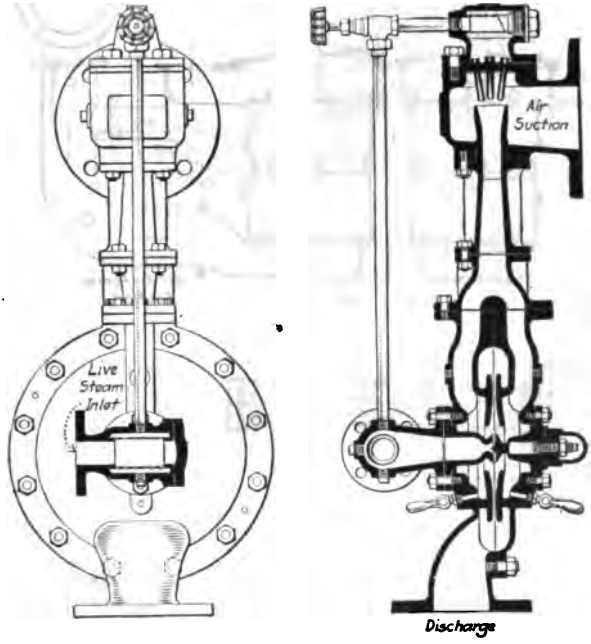


FIG. 36.

Wet and Dry Air Pumps

The main features in this pump are the combination of a wet and dry air pump in one unit, each having a separate suction leading to different parts of the condenser. The pump is of the vertical direct acting twin beam type. The wet cylinder is always located below the steam cylinder, as this pump is the only one which works under considerable load. The dry air pump receives a supply of cool injection water for condensing any vapors mixed with the air, this water being cooled in a separate cooler supplied with circulating water from the main circulating pump, see Fig. 37. The dry pump discharges through a spring loaded valve, adjusted to maintain about 20 in. vacuum into the wet pump at a point below its head valves. A diagrammatic section of the **"Dual" wet and dry air pump** is shown in Fig. 37.

The combination of the wet and dry air pump in one unit makes it possible to obtain a vacuum as high as $1\frac{1}{2}$ in. absolute, but the steam consumption of this pump is high when compared with that of air ejectors.

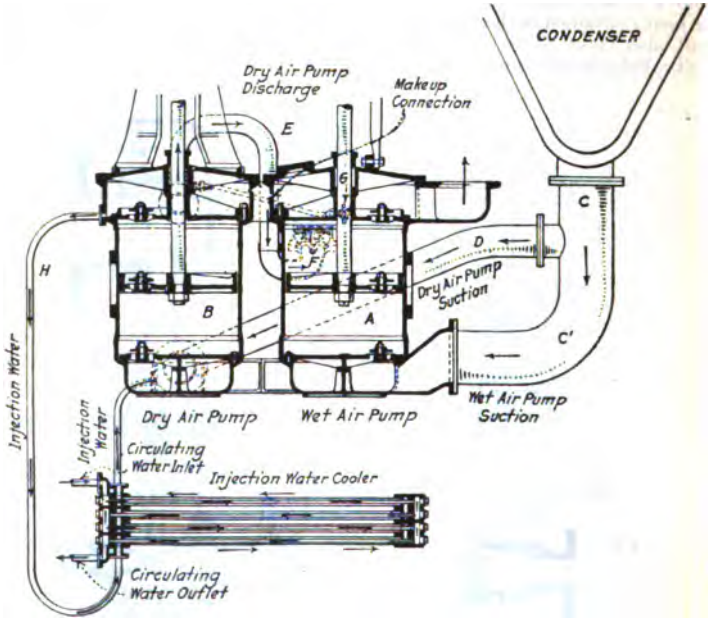

CONDENSER
Dry Air Pump Discharge
Makeup Connection
E
G
C
D
H
B
A
Dry Air Pump Suction
C'
Injection Water
Injection Water
Dry Air Pump
Wet Air Pump
Wet Air Pump Suction
Circulating Water Inlet
Injection Water Cooler
Circulating Water Outlet

FIG. 37.

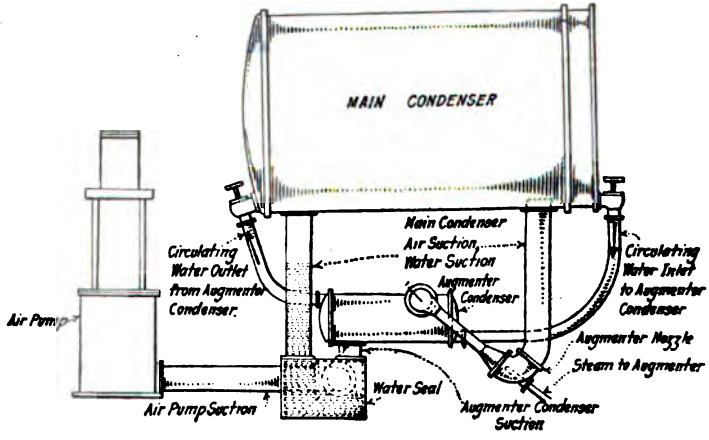

MAIN CONDENSER
Main Condenser Air Suction, Water Suction
Augmenter Condenser
Circulating Water Outlet from Augmenter Condenser
Air Pump
Circulating Water Inlet to Augmenter Condenser
Augmenter Nozzle
Steam to Augmenter
Water Seal
Air Pump Suction
Augmenter Condenser Suction

FIG. 38.

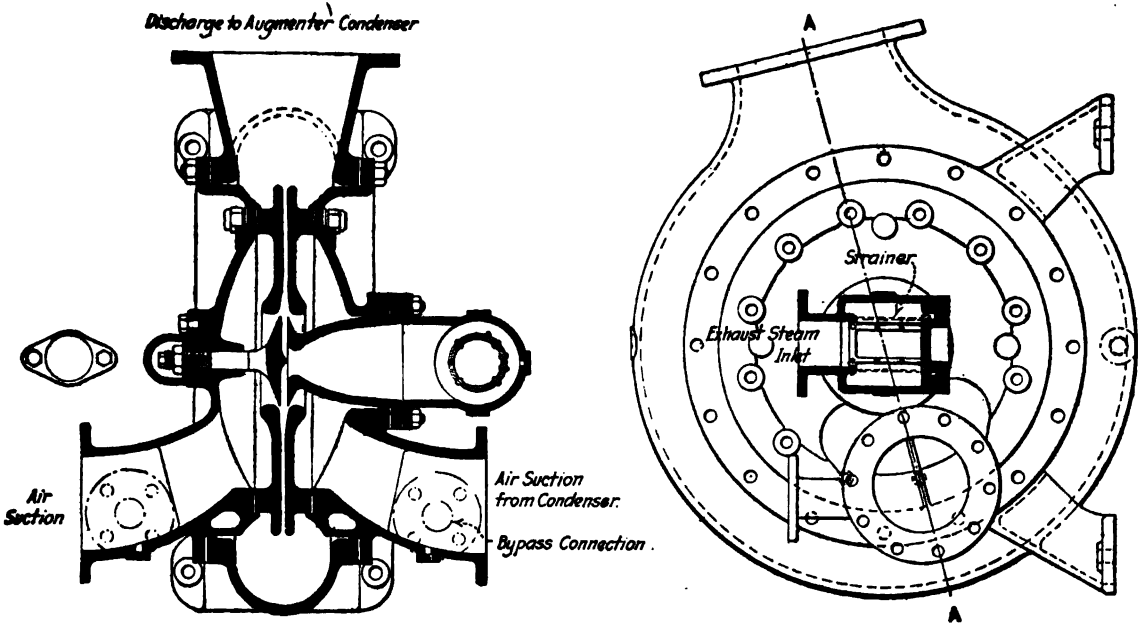

Discharge to Augmenter Condenser
Air Suction
Air Suction from Condenser
Bypass Connection
A
Strainer
Exhaust Steam Inlet
A

FIG. 39.

VACUUM AUGMENTERS

Vacuum Augmenters were first introduced by Parsons; Fig. 38 shows the arrangement of the **Parsons Augmenter.** They are used to improve the performance of the wet air pumps. Air and vapors are exhausted from the main condenser by means of a steam jet into the augmenter condenser, in which the operating steam is condensed. The wet reciprocating air pump removes the air and condensed operating steam from the augmenter condenser and also the condensate from the main condenser.

It is necessary to provide a water seal in the condensate suction pipe from the main condenser. The height of this water seal depends on the difference in pressure produced by the steam jet.

Live steam, from 125 to 250 lbs. gage pressure is used for the steam jet, the amount of which is from 0.8 to 1.5 per cent of the steam consumed by the main engine.

Fig. 39 shows the **Radojet Augmenter,** which is arranged in the same way as the Parsons Augmenter, but uses exhaust steam of 3 to 5 lbs. gage pressure, the amount of which is not more than 0.25 to 0.3 per cent of that used by the main turbine.

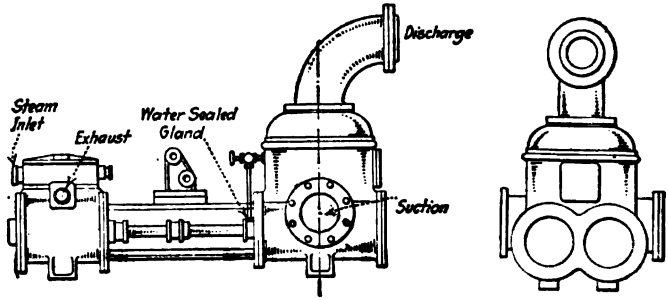


FIG. 40.

CONDENSATE PUMPS

Condensate Pumps are either of the direct acting reciprocating or of the centrifugal type. **Reciprocating Condensate Pumps** are used for vacuua up to 26 in. Water sealed glands must be provided to prevent air leakage through the stuffing box. The piston speed should not be more than 25 ft. to 30 ft. per min. The steam end should be designed large enough to take care of twice the normal capacity. The operation of this pump should be controlled by a float arranged in the hot well of the condenser, to prevent racing of the pumps when the supply of condensate is small. Fig. 40 shows a horizontal direct acting reciprocating duplex condensate pump.

Centrifugal Type Condensate Pumps are either direct steam turbine driven or motor driven. Impellers must be specially designed to remove the condensate from a high vacuum, while maintaining a low head on the suction side. The stuffing boxes must be water sealed to avoid air leakage. The pump suction must be provided with an air vent to avoid air or vapors entering the impeller. The water velocity in the suction pipe should be not more than 2 ft. to 4 ft. per sec., depending on the size of pump, submergence and

vacuum. Fig. 41 illustrates a direct steam turbine driven condensate pump. Motor driven pumps are sometimes used on ships which are electrically propelled.

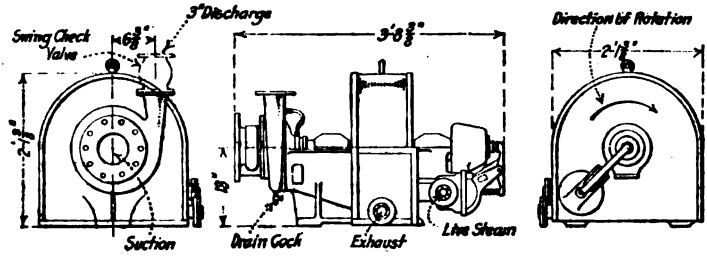
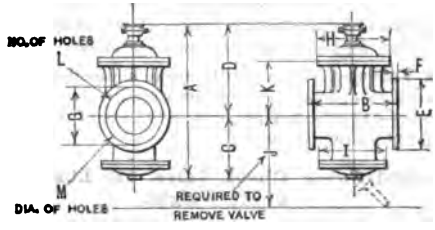


FIG. 41.

ACCESSORIES

Automatic Steam Pressure Reducing Valves are used for maintaining a constant steam pressure at the inlet of the auxiliaries with a varying boiler pressure. A "Leslie Valve" is illustrated in Fig. 42. Steam enters through the inlet port *S* to controlling valve *J*, which is adjusted by the tension of the adjusting spring *L* which presses against the diaphragm *G*

Table 4. Pressure Reducing Valve



Size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*	*M*	Weight
½"	9¾"	6½"	2¼"	7½	4½	9/16"	3¼"	5 1/16 "	3½ "	3¼"	3½ "	4	5/8"	29
¾"	9¾"	6½"	2¼"	7½"	4½"	9/16"	3¼"	5 1/16 "	3½ "	3¼"	3½ "	4	5/8"	29
1 "	9¾"	6½"	2¼"	7½"	4½"	9/16"	3¼"	5 1/16 "	3½ "	3¼"	3½ "	4	5/8"	29
1¼"	10 5/8"	6½"	2 5/8"	7¾"	5 "	9/16"	3¾"	5 1/16 "	3¾ "	3 5/8"	3 13/16"	4	5/8"	31
1½"	11¼"	6½"	3 "	8¼"	6 "	9/16"	4½"	5 1/16 "	4 "	4⅛"	3⅛ "	4	¾"	33
2 "	13⅛"	7½"	3¾"	9⅞"	6½"	5/8 "	5 "	5 15/16"	4 1/16"	4⅝"	4⅝ "	4	¾"	57
2½"	14¼"	9 "	3⅞"	10 3/8"	7½"	5/8 "	5 7/8"	7¼ "	5½ "	5⅞"	5¼ "	4	⅞"	75
3 "	15⅞"	10 "	4⅝"	11¼"	8¼"	1 1/16"	6⅝"	7¾ "	6⅛ "	6⅞"	6⅛ "	8	⅞"	95

Flanges are drilled according to A.S.M.E. standard for 250 working pressure.

Steam enters on top of piston *F* pressing the same downward and opening the main valve *D*. As soon as the steam pressure on the outlet side *T* becomes higher than that for which the valve is set, the steam passing through port *U* and pressing against the diaphragm *G* overcomes the tension of adjust-

ing spring L, thereby closing the controlling valve J which in turn causes the closing of the main valve D. Table 4 gives a list of standard sizes.

It should be borne in mind, that automatic pressure reducing valves require a pressure difference of at least 30 per cent between boiler pressure and delivery pressure in order to operate satisfactorily. The greater the pressure difference the more satisfactory the operation. Such a valve should be set when the boiler pressure is at maximum.

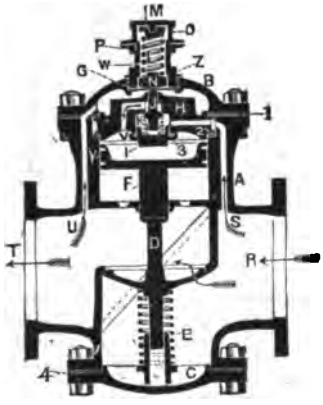


FIG. 42.

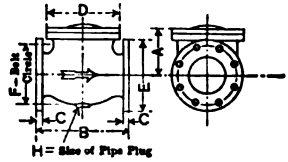
Relief Valves are frequently provided on the exhaust trunks between the main condenser and turbine. They are generally set for 5 lbs. gage. Their **outlet should be arranged upward with a water seal.** This enables the operating engineer to see instantaneously whether the valve is air-tight or not.

The **size** of this valve depends on the size of the main turbine and the type of air pump used in connection with the main condenser. When using the reciprocating type of air pump, the valve must be considerably larger than when using a steam air ejector, because the latter permits the free discharge of steam through the ejector and centrifugal condensate pump, should the steam from the turbine build up pressure in the condenser.

A 3-in. relief valve is generally used with a 3000 S.H.P. turbine provided with steam air ejector and centrifugal type condensate pump.

Swing Check Valves are frequently used in discharge pipes of centrifugal type condensate pumps, and of air ejectors to facilitate the starting of the pumps. Fig. 43 illustrates such a valve of the light-weight type. Table 5 gives a list of standard sizes.

Table 5. Standard Swing Check Valve, 125 Lbs. Pressure, Special Light Weight



Size	A	B	C	D	E	F	G	H	Weight, lbs.
2	3⅞	8	⅝	6⅛	6	4¾	4⅜	⅜	25
2½	4⅜	8½	11/16	6⅝	7	5½	4⅜	⅜	32
3	4 11/16	9½	¾	7¼	7½	6	4⅜	⅜	42
3½	5⅛	10½	13/16	8	8½	7	4¾	½	55
4	5¾	11	15/16	8½	9	7½	8⅜	½	65
5	6½	13	15/16	10¼	10	8½	8¼	⅝	86
6	7	14	1	11½	11	9½	8¼	⅝	120
8	8¼	17	1⅛	14⅛	13½	11¾	8¾	¾	160

G = No. and size of bolts.

Automatic Temperature Control Valves are used in connection with the steam air ejector system to prevent water in the feed and filter

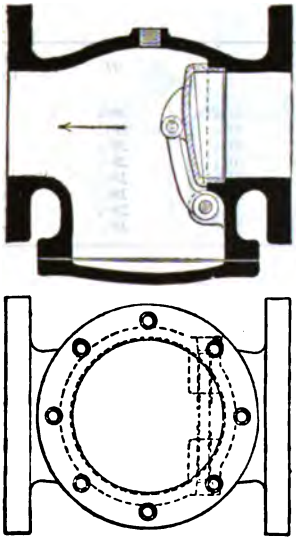


FIG. 43.

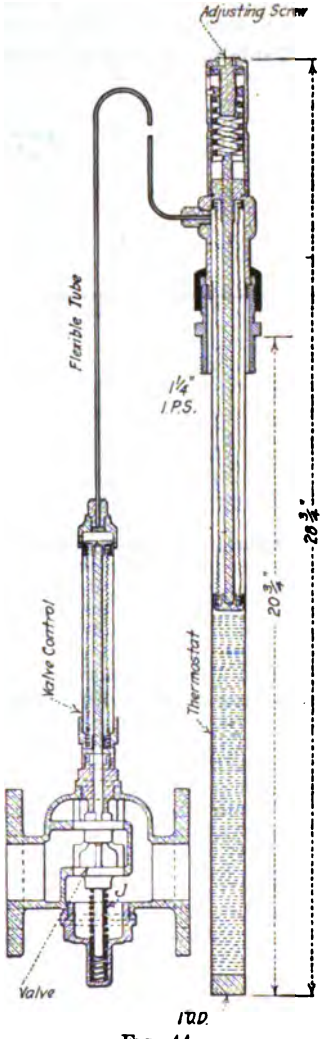


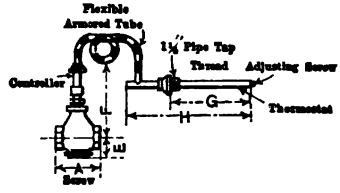
FIG. 44.

tank from boiling. They regulate the water supply by thermostatic contro- and generally consist of a thermostat and a thermostatic valve, both con-

nected by means of an armored flexible tube. The thermostat is set for a pre-determined temperature and when this temperature is reached it opens the valve. The "**Sarco**" **Thermostatic Valve** is shown in Fig. 44 and Table 6 gives a list of standard sizes.

Table 6. Dimensions of Automatic Temperature Regulator

(C. H. Wheeler Co.)



Size	A	E	F	G	H	Weight, lbs.
½	2¾	1¼	10 15/16	20¾	28¾	10
¾	3⅜	1 9/16	10 5/8	20¾	28¾	10
1	3 11/16	1 3/16	10 7/8	20¾	28¾	12
1¼	4⅛	3 1/16	10 13/16	20¾	28¾	15
1½	5	3 9/16	12	20¾	28¾	25
2	6	4¼	15⅞	20¾	28¾	30
2½	7¼	4¾	16 15/16	20¾	28¾	40
3	8¾	5 13/16	17⅝	20¾	28¾	56

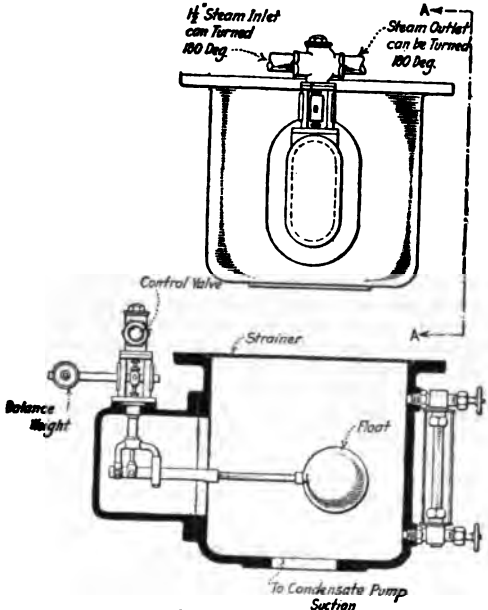
Automatic Pump Control Valves. When using the reciprocating type condensate pump it is necessary to regulate the speed of the pump according to the amount of condensate the pump receives from the main engine. Without such control the pump would race when the supply of condensate is very small. Fig. 45 illustrates an automatic control valve for a reciprocating type condensate pump, a float is arranged in the hot-well of the condenser, and operates the control valve.

Draining L. P. Turbine Casings, see also p. 628. In marine condenser installations, the condenser is usually located higher than the L. P. turbine casing. This prevents draining by gravity from the casing into the condenser. Other means for draining prior to starting and during maneuvering have to be provided.

In installations where it is possible to locate the air or condensate pump from three to four feet below the lowest point of the L. P. turbine casing, the drains may be lead directly into these pumps. The drain pipe should have a gradual fall toward the pump with few bends and should be as short as possible.

In installations where the above is not possible, an arrangement as shown in Fig. 46 is recommended. A steam operated syphon (**injector**) entrains the drains and discharges them through a swing check valve into the main condenser about two to three feet above the bottom of the condenser As vacuum exists in both turbine casing and condenser, the steam has only to lift the drains against a head equal to static head plus friction through piping.

A single stage injector of ½-in. diameter steam inlet is used for a 3000 S.H.P. turbine. It is hand operated during starting and maneuvering. During normal operation at full load with dry or superheated steam, draining



Sectional Elevation A-A

FIG. 45.

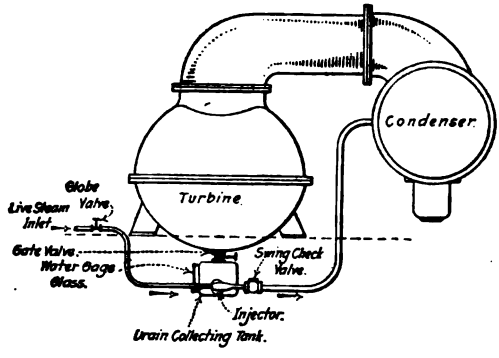


FIG. 46.

is not necessary as the exhaust steam with its high velocity entrains any condensation and carries it over into the condenser.

Vacuum traps are not recommended on board of vessels, particularly in connection with high vacuum condensers.

ARRANGEMENT OF CONDENSING PLANTS

The arrangement of condensing plants depends entirely on the type of condenser, the type of pumps used and the space available in the engine room for its auxiliaries. For these reasons, it is impossible to establish general rules. The following may be used as a guide:

Circulating Pumps

Make suction and discharge line as short as possible. Avoid sharp bends or pockets. Discharge pipe should be at least one to two feet above highest tube in condenser to insure that the condenser tubes are never dry. Provide a large strainer in sea chest, also a live steam connection for blowing out and thawing out same. Provide quick opening gate valves in suction and discharge line to permit overhauling condenser. Weight loaded check valves may be used in discharge line. When the discharge is arranged below the water level, provide air vent cocks at highest point of discharge line. Provide a pipe connection with a gate valve from bilge to the suction line of pump. If pump is reciprocating engine driven or direct acting, connect steam end to auxiliary steam line. If turbine driven, connect to H. P. steam line. Arrange drains in steam line close to pump.

Wet Air Pumps

Locate air pump so that discharge valve deck is not less than four feet below lowest point of main condenser, and so that discharge head to feed and filter tank does not exceed five feet. Make suction pipe as short as possible with continuous fall of not less than 30 degrees from condenser to pump, avoiding pockets and sharp bends. Provide air vent in discharge pipe. Connect steam end to auxiliary steam line. Provide drain in steam ine close to pump.

Dry Air Pumps

Locate the dry air pump suction, above air opening in condenser, making suction line as short as possible; avoid sharp bends and pockets, permitting drain of the condensation by gravity to the condenser. Arrange gate valves close to pump suction. When using air ejectors always locate them above highest water level in feed and filter tank, to enable the exhaust to drain by gravity toward feed and filter tank. Arrange light swing check valve in discharge line from air ejector; avoid sharp bends and pockets. Provide large vent in chamber of feed tank into which air ejector discharges. Connect steam end to H. P. line, arrange automatic pressure reducing valve in same, and provide drain close to ejector. Provide device to prevent boiling of feed water in feed tank.

Condensate Pumps

If of the centrifugal type and for high vacuum condensers, locate pumps so that the pump shaft is not less than three feet below the lowest point of the condenser. If of the reciprocating type and for 26 in. vacuum, the suction opening should not be less than five feet below the water level in the hot-well.

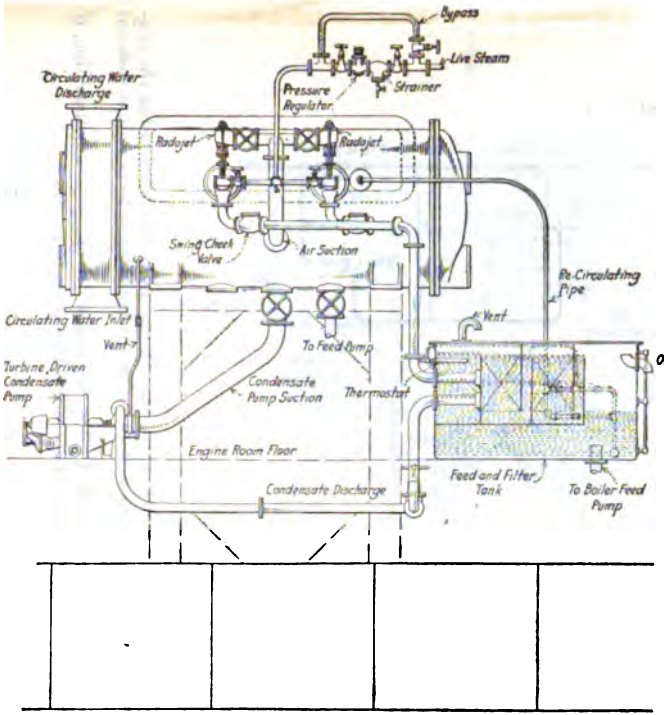


FIG. 47a.

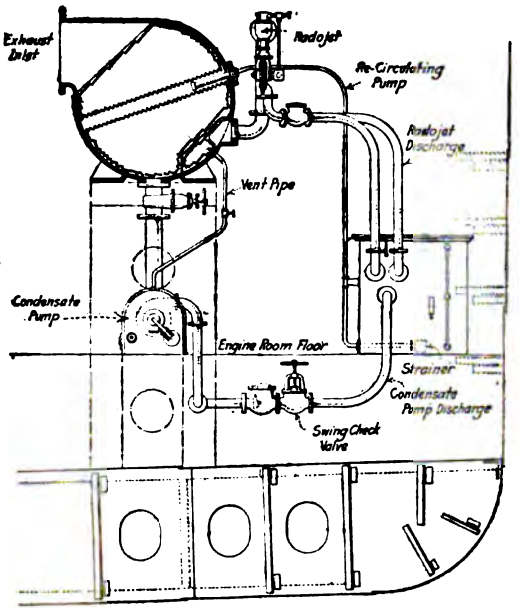


FIG. 47b.

For both cases the suction pipe should be as short as possible with large bends, if such are necessary, and with a gradual fall toward the pump suction opening. A gate valve should be arranged close to the condenser. A vent connecting the pump suction inlet to the condenser should be provided.

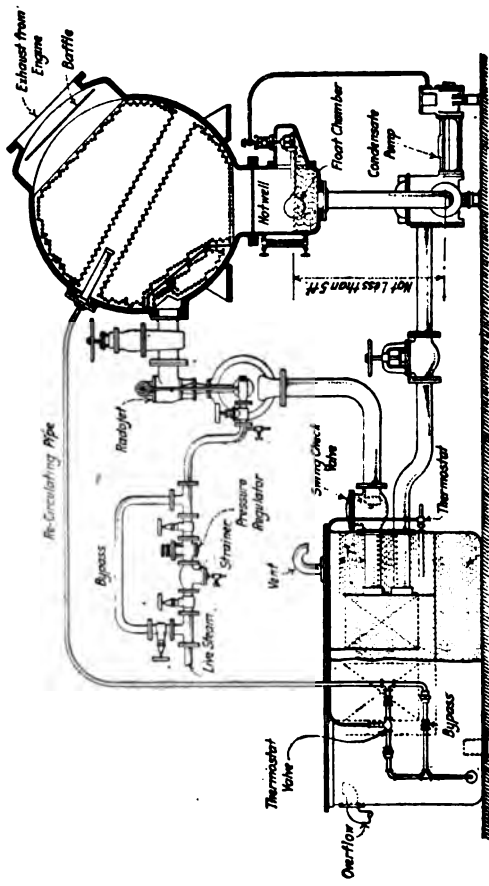


FIG. 48.

Provide a swing check valve in the discharge line when using the centrifugal type pump, also a gate valve for both types to permit occasional inspection.

Connect steam end to H. P. line, if turbine driven, or to auxiliary line, if direct acting. Provide drains in steam line close to pump.

General

Fig. 47 shows the arrangement of a **surface condenser installation for a steam turbine** and Fig. 48 **for reciprocating engine.** The arrangement of the circulating pump is shown in Fig. 49.

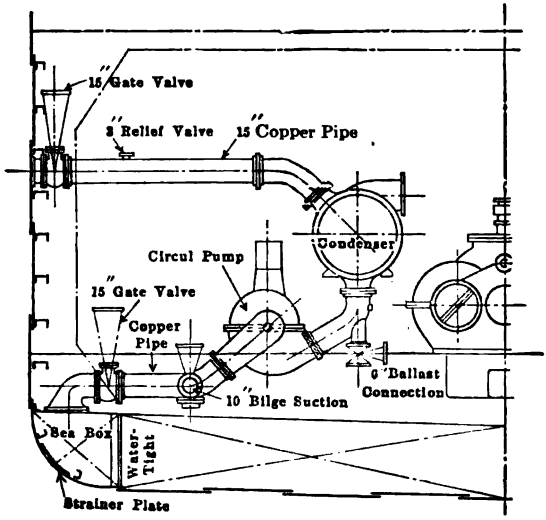


FIG. 49.

Acknowledgment is made for material in this section to:

American Ship Building Co., Cleveland, Ohio.
Great Lakes Engineering Works, Detroit, Mich.
Navy Department, Bureau of Steam Engineering.
Sun Shipbuilding Co., Chester, Pa.
Taylor Instrument Co., Rochester, N. Y.
Schaeffer H. Budenberg Co., New York City.
C. H. Wheeler Mfg. Co., Philadelphia, Pa.
Warren Steam Pump Co., Warren, Pa.
G. & J. Weir, Ltd., Catheart, Scotland.
Westinghouse Electric & Mfg. Co., East-Pittsburgh, Pa.
Sarco Co., New York City.
Leslie Co., Lyndhurst, N. J.

SECTION 9

SHIP FORMS AND POWERING AND SCREW PROPELLERS

BY

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CONTENTS

SHIPS FORMS AND POWERING

By E. M. BRAGG

Ships Forms

	Page
Choice of Dimensions	1013
Length	1013
Breadth	1013
Draft and Depth	1014
Economical Speed	1014
Prismatic Coefficient	1014
Curve of Sectional Areas	1014
Parallel Middle Body	1018
Vertical Prismatic Coefficient	1021
Height of Metacenter	1021
Load Water Line	1022
Body Plan	1022

Simplified Forms

McEntee's Form	1023
Whittlesey's Form	1023

Powering of Ships

Steven's Formula	1025
Effective Horsepower from Model Tests	1026

SCREW PROPELLERS

By C. W. DYSON

	Page
Thrust Deduction and Wake Gain	1028
Slip Block Coefficients:	
Wing Screws	1030
Single Screw Ships	1031
Tunnel Boats	1031
Determination of Thrust Deduction Factor	1032
Mean Relative Tip Clearance	1033
Resistance of Hull Appendages	1034
Basic Condition for Analysis and Design	1035
Cavitation	1043
Analysis and Design of Propellers	1046
Design of the Propeller	1055
Forms for Computation	1056
Standard Propeller Hubs	1064

SHIP FORMS AND POWERING

BY

E. M. BRAGG

SHIP FORMS

REFERENCES: Taylor, "Speed and Power of Ships," J. Wiley & Sons. Baker, "Ship Form, Resistance, and Screw Propulsion," Constable & Co., Ltd. Biles, "The Design and Construction of Ships," C. Griffin & Co., Ltd. Transactions of the Institute of Naval Architects (London). Transactions of the Society of Naval Architects and Marine Engineers (New York).

Choice of Dimensions. The **length, breadth,** and **draft** of a ship are usually determined by other conditions than those of minimum resistance to propulsion. The length will be determined by the speed at which the ship is to run, the length of voyage, and the character of service. It is not economical to run a large ship on a short voyage nor a small ship on a long voyage. The **breadth** will be determined largely by the question of stability. The **draft** will be limited by the depth of water in the harbors to be visited.

Length. The relation of length and speed to character of service is discussed by *Mr. John Anderson* in a paper before the *Institution of Naval Architects, London, March* 20, 1918. The efficiency of the unloading machinery is a matter of considerable importance as this determines largely the amount of time spent in port. In Mr. Anderson's paper two conditions are assumed; first, that each hatch is served by winches working to full capacity, and second that the winches are working to only 45 per cent of their full capacity; for winches and unloading cycles see pp. 1234–1239. The effect of this element upon the length of ship is shown in Table 1.

Table 1

Length of voyage, knots	Unloading capacity	Speed in knots				
		8	9½	11	12½	14
1000	45 per cent	293	305	315	325	335
1000	Maximum	340	345	355	365	375
4000	45 per cent	370	380	390	410	420
4000	Maximum	395	415	435	465	485
8000	45 per cent	400	405	425	450	470
8000	Maximum	420	430	460	500	545

Breadth. The breadth of ships is usually (length + 10) + 12 to 15 ft. for ordinary cargo and passenger ships, while those carrying fluid cargoes may need to be 3 ft. broader than this to get sufficient stability.

Draft and Depth. The **maximum draft** will be equal to the depth of the ship minus the freeboard. The **freeboard is determined** by certain rules which can be obtained from the registering societies. While the draft is the most economical dimension to increase as far as propulsion is concerned, if it calls for increased depth of ship it will increase the first cost of the ship. From the point of view of strength a broad ship need not be as deep as a narrow ship. The usual relation of length, breadth, and depth is given by Fig. 1. It is best not to have the ratio of length to depth exceed 13.5 as the rules call for a considerable addition of metal when this ratio is reached.

So far the dimensions have been determined from considerations of service, stability, and strength. It now remains to choose the best shape for the underwater body

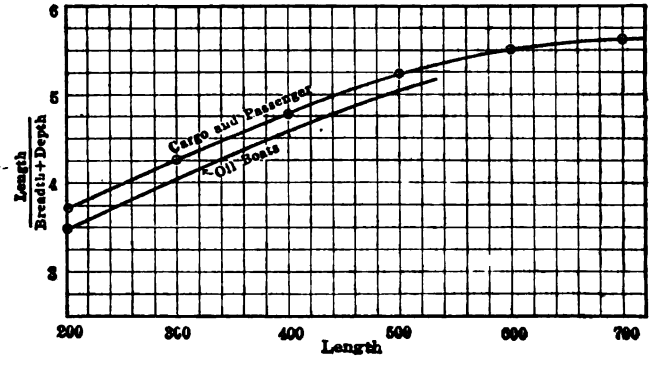


FIG. 1.

Economical Speed. The economical speed for a ship cannot be determined from considerations of resistance alone but involves the length of voyage, character of cargo, type of propelling machinery, cost of fuel, wages of crew, etc. However, it is usually considered not wise to drive a ship beyond the point where the resistance increases as the third power of the speed, i.e., the point where a 5 per cent increase in speed calls for a 15 per cent increase in power. If the model is a suitable one for the speed the wave resistance will be less than 50 per cent of the frictional resistance for cargo ships, but for fast passenger ships may be nearly equal to the frictional resistance.

Prismatic Coefficient. The relation between speed-length ratio and prismatic coefficient is given in Fig. 2. The speed to be used is the trial trip speed. The **service speed** will be about 10 per cent less than the trial trip speed. It is well established now that the **prismatic coefficient should not exceed** 0.79 in the case of ocean-going ships if regularity of service is desired.

Curve of Sectional Areas. Fig. 3 gives the ordinates which the sectional area curve should have at the various stations for any desired prismatic coefficient. These sectional area curves will be fairly straight at bow and

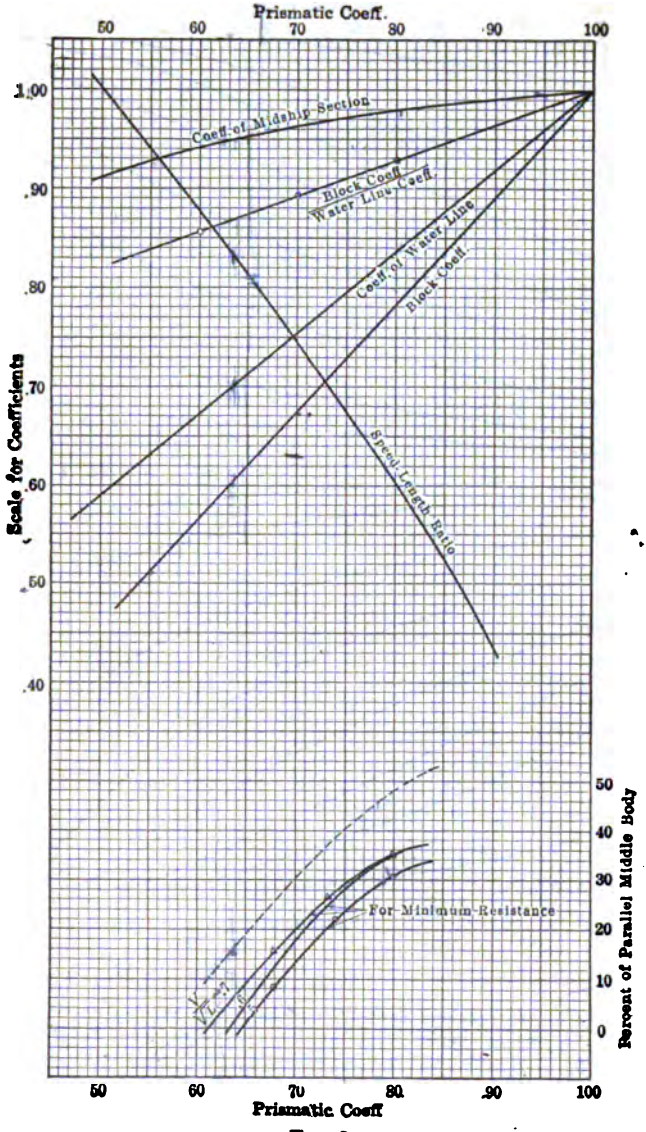

Prismatic Coeff.
50
60
70
80
90
100
1.00
.90
.80
.70
.60
.50
.40
Scale for Coefficients
Coef. of Midship Section
Block Coeff.
Water Line Coef.
Coef. of Water Line
Block Coeff.
Speed-Length Ratio
For Minimum Resistance
50
40
30
20
10
0
Percent of Parallel Middle Body
50
60
70
80
90
100
Prismatic Coeff

FIG. 2.

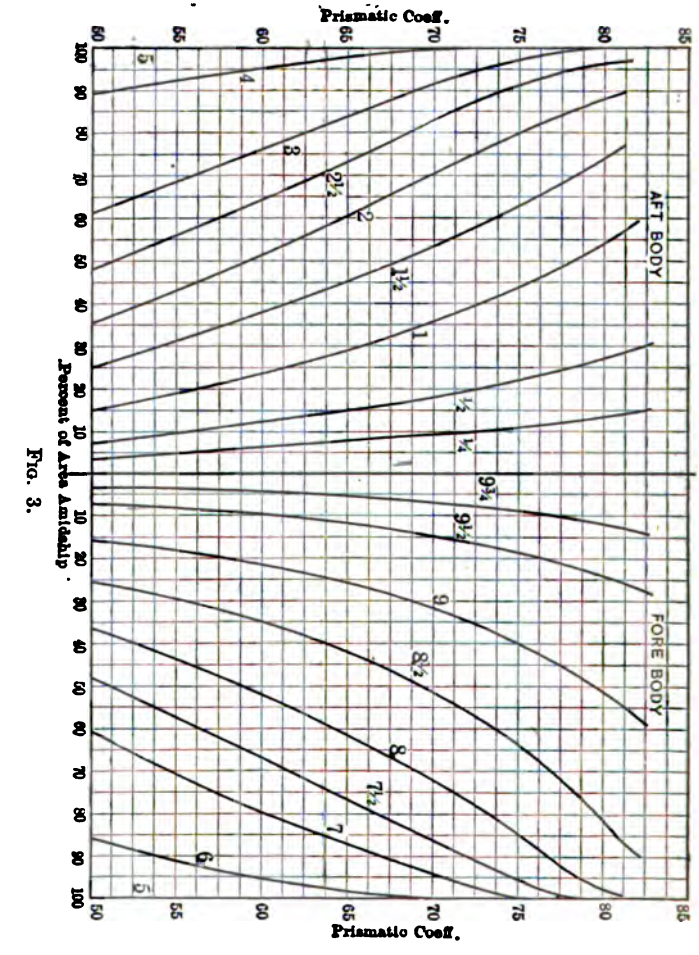

Prismatic Coeff.
50 55 60 65 70 75 80 85
Percent of Area Amidship
100 90 80 70 60 50 40 30 20 10
AFT BODY
FORE BODY
5
4
3
2½
2
1½
1
½
¼
9¼
9½
9
8½
8
7½
7
6
5

FIG. 3.

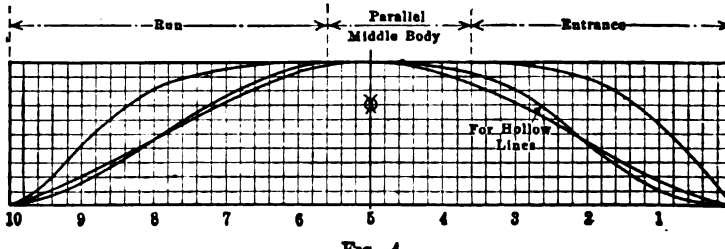

Run
Parallel Middle Body
Entrance
For Hollow Lines
10 9 8 7 6 5 4 3 2 1 0

FIG. 4.

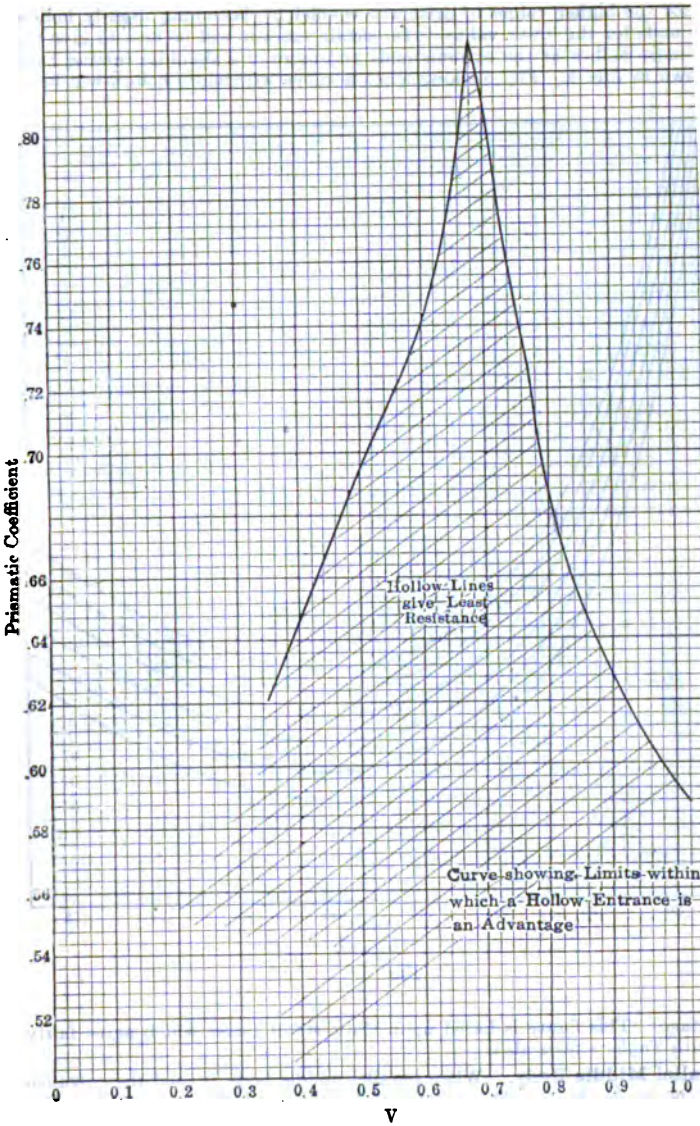

Prismatic Coefficient
.80
.78
.76
.74
.72
.70
.66
.64
.62
.60
.58
.56
.54
.52
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
Hollow Lines give Least Resistance
Curve showing Limits within which a Hollow Entrance is an Advantage
V / √L

FIG. 5.

stern for the higher values of prismatic coefficient, becoming slightly hollow at the ends for the lower values. In certain cases it will be best to give the curve more hollow at the extreme ends and more of a shoulder further back as shown in Fig. 4. Fig. 5 shows when a curve with more hollow will be of

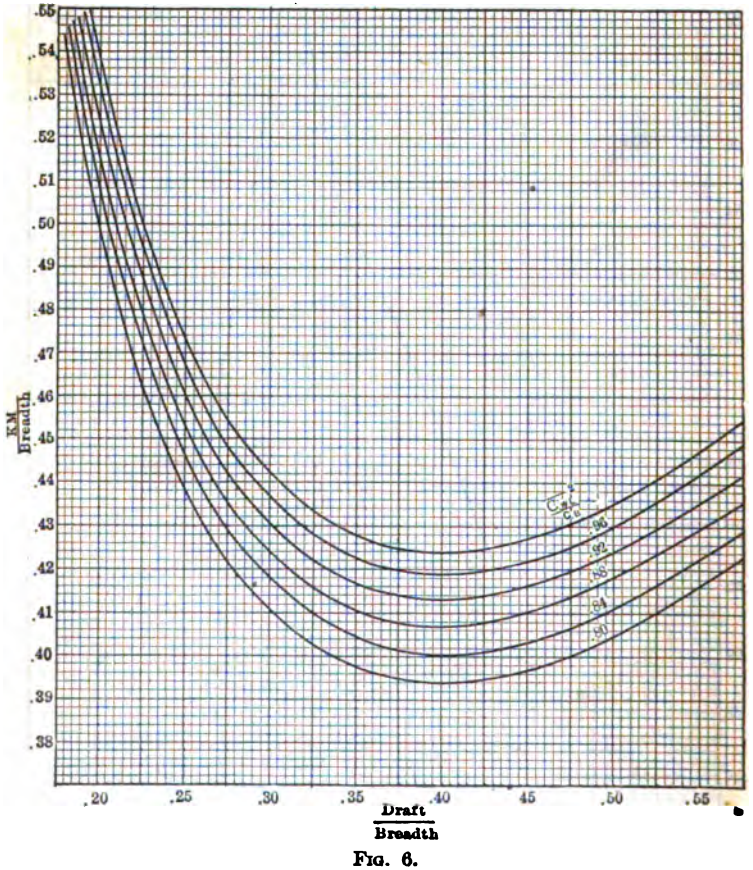


FIG. 6.

advantage. This figure is based upon the tests of *Baker* which agree fairly well with *Taylor's* tests also.

Parallel Middle Body. When parallel middle body is used the entrance should be made shorter than the run (see Fig. 4) in the case of the fuller ships. Fig. 2 shows the amount of parallel middle body, as determined by *Taylor's* experiments, which gives the least resistance for any given prismatic

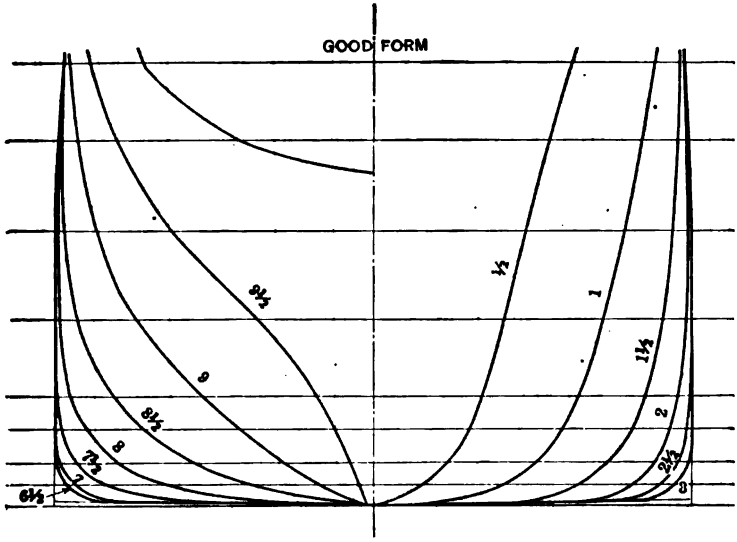


FIG. 7.

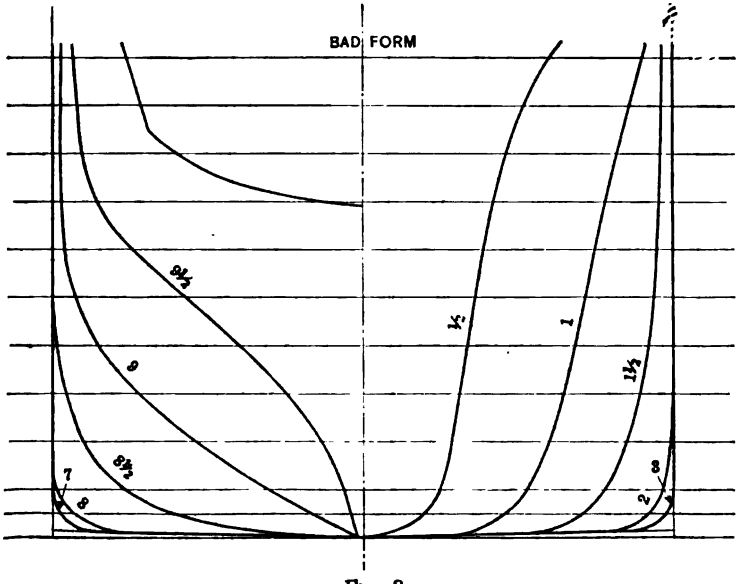


FIG. 8.

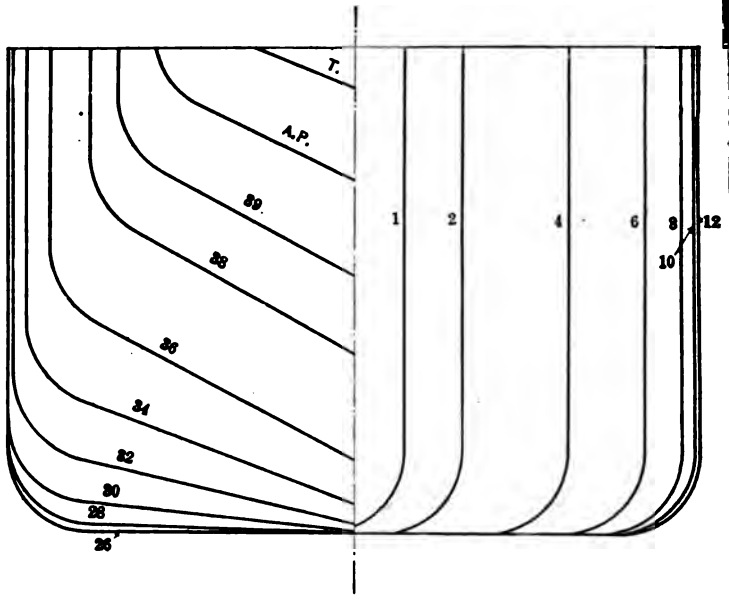


FIG. 9.—McEntee's simplified form.

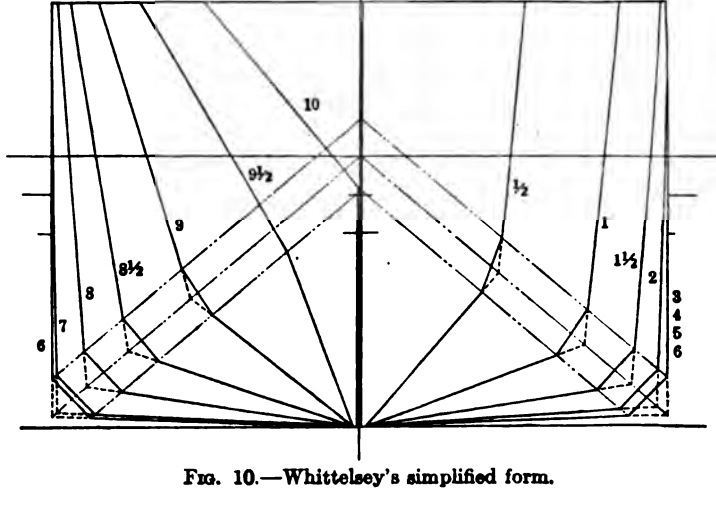


FIG. 10.—Whittelsey's simplified form.

coefficient. The broken line shows the amount which can be used and cause only a ten per cent increase in the wave resistance. If the model is a good one a 10 per cent increase of wave resistance would cause about a 4 per cent increase of total resistance.

The parallel middle body should not be placed symmetrical with the midship section but about two-thirds of it should be placed forward of amidships and about one-third aft. *Naval Constructor William McEntee* experimented with a self-propelled model having 33 per cent parallel middle body and found that the best condition was with the centre of the parallel middle body 43 per cent from the bow (see *Society of Naval Architects and Marine Engineers*, 1918).

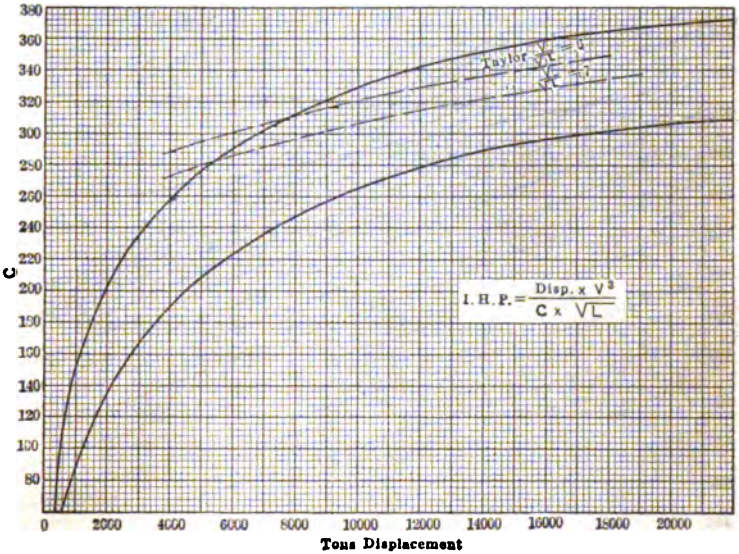


FIG. 11.

Vertical Prismatic Coefficient. The vertical prismatic coefficient is of importance as well as the longitudinal prismatic coefficient. The vertical prismatic is equal to the block coefficient divided by the water-line coefficient. This coefficient has not been studied as much as the other coefficient but should have about the values shown in Fig. 2, to get an easy driving model.

Height of Metacentre. At this point the proportions chosen should be checked for stability. The height of the centre of gravity of the ship will be known fairly closely. Fig. 6 gives the height of the metacentre above the keel in terms of the breadth. The values of the water-line coefficient and block coefficient to be used in getting the quantity $\overline{Cwl}^2 + Cb$ in Fig. 6 should be measured at a draft of 0.4 the breadth. However, the relative values of these coefficients does not change much for quite a change in draft

and they may be taken at any draft fairly close to the one mentioned. The height of the metacentre above the centre of gravity should have a minimum value of about one foot when the vessel is carrying a homogeneous cargo. The breadth of ship and water-line coefficient should be so chosen as to give the needed height of metacentre and at the same time to satisfy the conditions previously pointed out as necessary for strength and for good form of model.

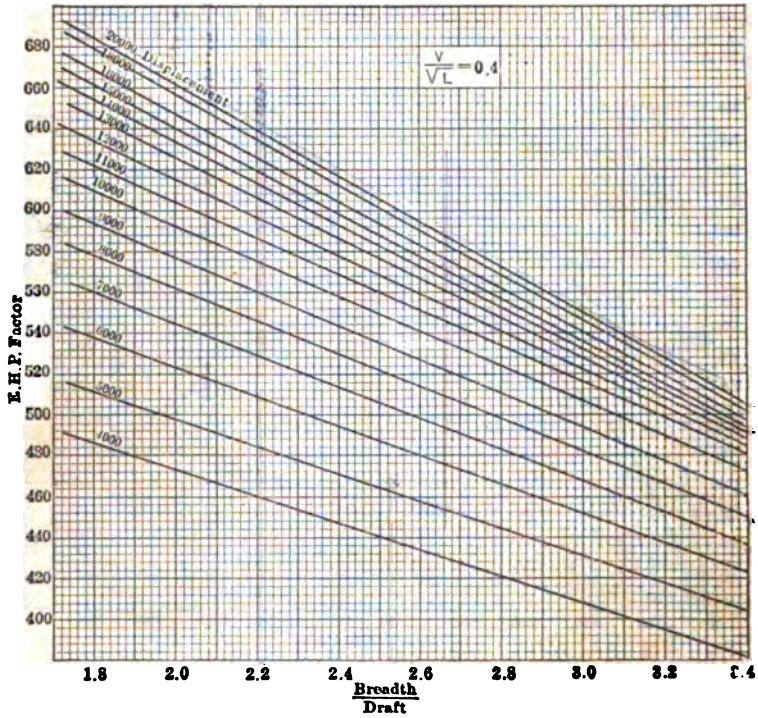


FIG. 12.

Load Water Line. Before the body plan of the ship can be made the load water line must be drawn. The load water-line coefficient has already been determined and the breadth at the different stations should be so chosen as to give a water plane of this fullness. The load water plane should be considerably fuller aft than forward so as to give easy buttock lines.

Body Plan. With the breadth of each section determined at the load water line, and the total area of each section known, the body plan should be drawn so as to give sections of the shape shown in Fig. 7. Model tank tests show that a model with sections like those shown in Fig. 7 will drive easily

up to a speed-length ratio of about 0.7, while a shape like that shown in Fig. 8, drives hard. As pointed out before, the attempt to use too much parallel middle body and a short run will give a hard driving model. A model similar to Fig. 8 was modified to the extent of reducing the parallel middle body by about 7 per cent at the aft end. This caused a loss of about ½ per cent in the displacement but reduced the resistance by about 5 per cent.

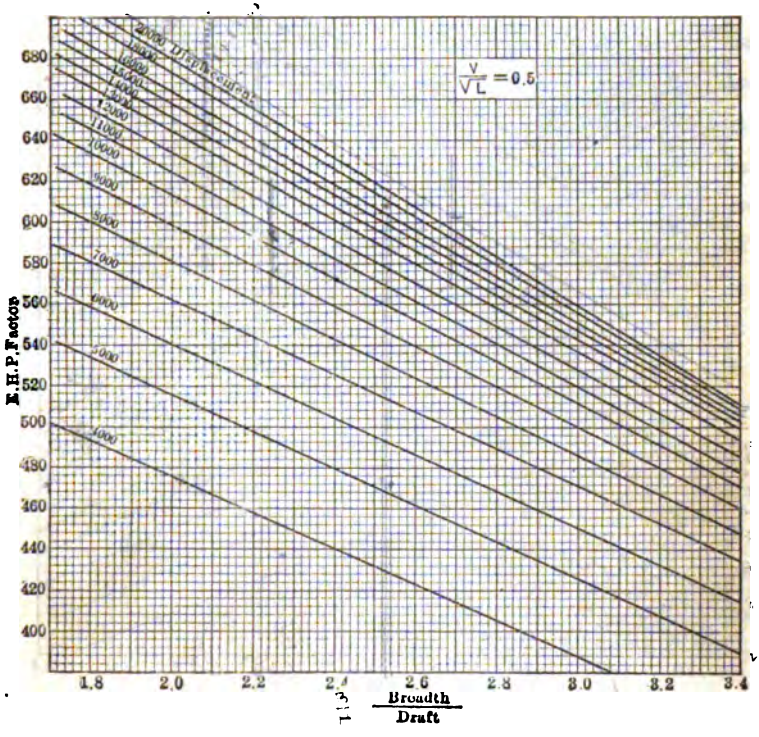


FIG. 13.

SIMPLIFIED FORMS

McEntee's Form. In a paper read in Nov., 1917, before the *American Society of Naval Architects and Marine Engineers*, *Naval Constructor McEntee* gave the results of tests made at the *U. S. Naval Model Tank* upon a simplified form shown in Fig. 9. This form drove slightly harder than the ordinary form that it was tested against.

Whittelsey's Form. In a paper read before the *American Society of Naval Architects and Marine Engineers*, *Nov.*, 1918, *Professor Sadler* gave the

results of tests made at the *University of Michigan* upon a simplified form advocated by *H. N. Whitelsey*. (See Fig. 10.) Models of this type with 0, 10, 20, and 40 per cent of parallel middle body were tested against other models similar in every way except that the sections were "ship shape" instead of straight lined. The tests showed that at a speed-length ratio of 0.7 the simplified form drove as easily as the ordinary form. Below 0.7

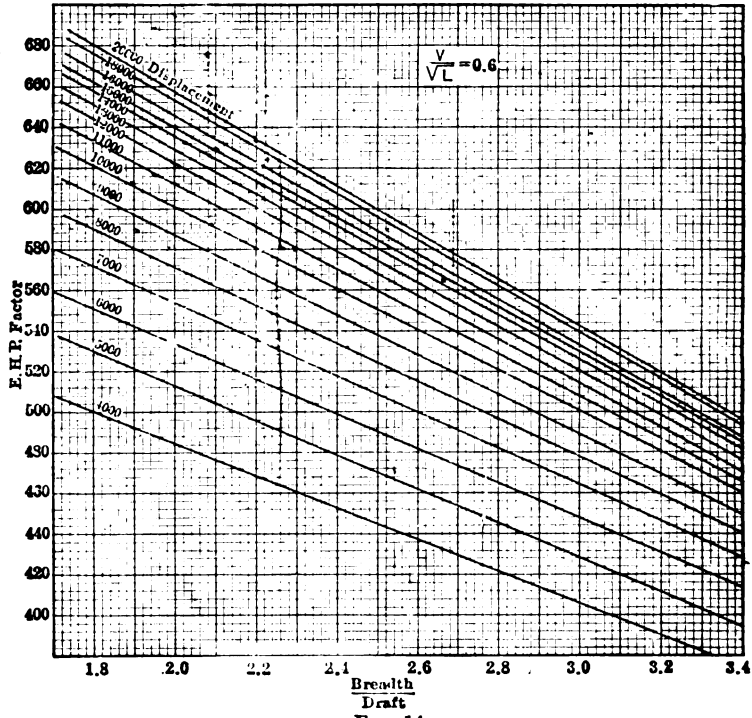


FIG. 14.

speed-length ratio the simplified form drove easier, while above 0.7 it drove slightly harder than the ordinary form. The advantage of the simplified over the ordinary form increased as the draft decreased.

When the corner volume was retained in Fig. 10 the displacement was increased about 1 per cent, but the resistance was increased about 3 or 4 per cent.

POWERING OF SHIPS

Bibliography: Speed and Power of Ships, D. W Taylor. Ship Form, Resistance, and Screw Propulsion, G. S. Baker. Naval Architecture, C. H. Peabody. Chapter XII.

In settling upon the power necessary to drive a ship at a certain speed certain latitude must be allowed. Sister ships, built from the same plans, under the same supervision, and equipped with the same propelling machinery often give speeds differing by one-half or three quarters of a knot.

Steven's Formula. The resistance of a ship is affected by so many factors that no simple formula can give very closely the power necessary to

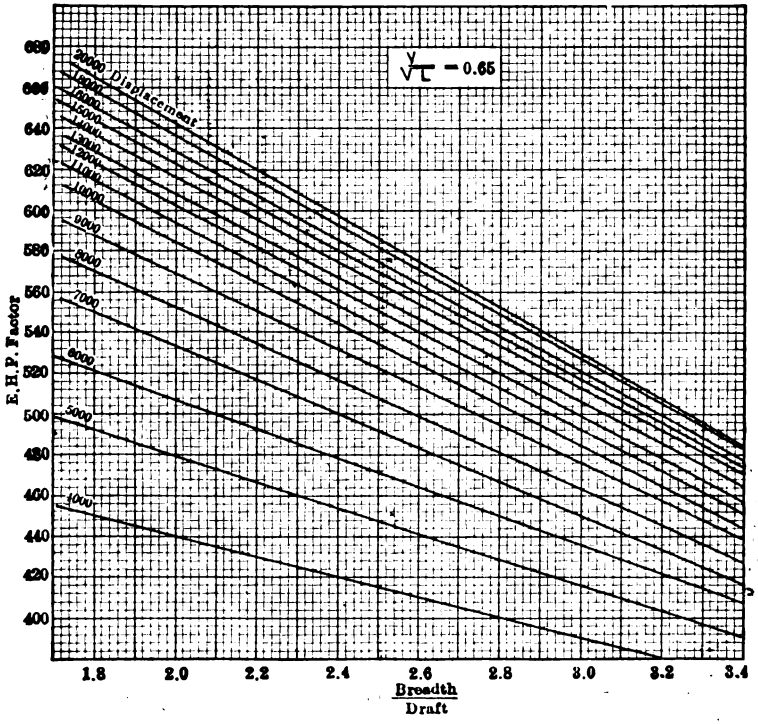


FIG. 15.

overcome resistance at all speeds and drafts. A modification of the well-known Admiralty formula, recommended by *Mr. A. E. Stevens* gives fairly good results. This formula is: $\mathbf{I.H.P.} = (\mathbf{D} \times \mathbf{V}^3) \div \mathbf{C}\sqrt{\mathbf{L}}$, **D** is displacement in long tons, **V** is speed in knots, **L** is length between perpendiculars in feet, and **C** is a factor given in Fig. 11. The two full-lined curves in Fig. 11, plotted upon displacement, show the limits through which this factor varies. While the variation is considerable it is not as great as with the **Admiralty formula.** The trials from which these curves were derived were of ships with a ratio of breadth to draft of about 2, and speed-length

ratios of 0.5 to 0.6 Ships with shallower draft and higher speed-length ratios will fall near the lower curve while deeper draft ships of moderate speed-length ratios and good form will fall near the upper curve. Ships of higher speed-lengths ratios and with propellers making a high number of revolutions will fall below the lower limit.

The two broken curves are taken from the data given in *Taylor's "Speed and Power of Ships,"* for a breadth-draft ratio of 2.25. **Taylor's curves**

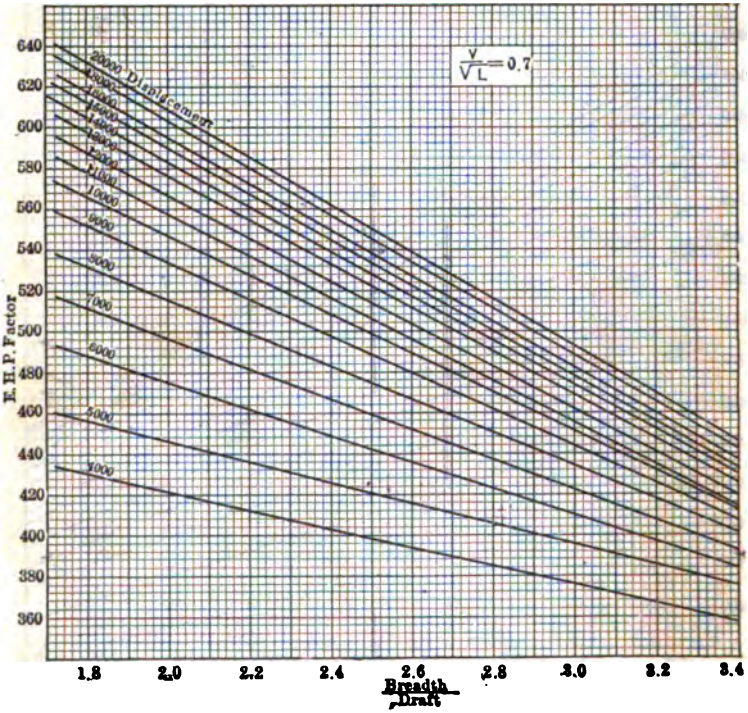


FIG. 16.

do not extend beyond a displacement-length ratio of 160 while the assumed ships were all fuller than that. The assumption was made that the wave resistance per ton of displacement was the same at these higher displacement-length ratios as those given for a value of 160. It was also assumed that the I.H.P. = 2 × E.H.P.

Effective Horsepower from Model Tests. The results of tests made upon some 20 models at the *University of Michigan* have been plotted according to the formula given above, for different speed-length ratios and varying ratios of breadth to draft, using the *effective* horsepower instead of the

indicated horsepower. For any given speed-length ratio and any given ratio of breadth to draft the spots occupied a certain area, and Figs. 12–16 show the mean lines drawn through those areas. Up to speed-length ratios of about 0.65 forms such as shown in Fig. 7, with easy bilges, long runs, and moderate parallel middle body will have values of C larger by 30 or 40 than the mean values shown in the figures, while forms such as shown in Fig. 8 and forms with short runs and excessive parallel middle body will have C values smaller by 30 or 40 than those given. Above the speed-length ratio of 0.65 the "club-footed" section (see Fig. 8) shows to better advantage than at the slower speeds. These results are for bare models and allowance for appendages should be made.

SCREW PROPELLERS

BY

C. W. DYSON

REFERENCES: Dyson, "Screw Propellers for Hydraulic and Aerial Propulsion," J. Wiley & Sons, Inc. Taylor, "Speed and Power of Ships," J. Wiley & Sons, Inc.

The Quantities entering into the performance of a screw propeller are:

1. The form of the underwater body of the ship.
2. The location of the propeller in relation to the hull.
3. The diameter of the propeller $= D$ (in feet).
4. The pitch of the propeller $= P$ (in feet).
5. The projected blade area ratio of the propeller $= P.A \div D.A.$
6. The revolutions of the propeller per minute $= R_d$.
7. Speed of the ship in knots $= v$.
8. Speed of advance of propeller in relation to water at stern in which it is working.
9. Indicated horsepower $=$ I H.P.$_d$, or shaft horsepower $=$ S.H.P.$_d$

THRUST DEDUCTION AND WAKE GAIN

The form of the after body lines of the underwater body of the ship and the location of the propeller in relation to these lines affect the mean direction of flow of the currents of water entering the propeller. While passing through the propeller some change in the mean direction of flow is produced, tending to throw the water more directly astern. The reduced propelling effect produced by the angularity of entrance of the currents into and the work done in changing the direction of flow of the currents while passing through the propeller necessitate the expenditure of an excess of power over that which would be required were the lines of flow of these currents normal to the disc of the propeller. **This excess expenditure of power is indicated under the name of Thrust Deduction.**

Thrust Deduction is often stated to be the loss of propelling effect caused by the reduction of pressure of the water in the wake of the hull where the propellers are located. This is incorrect, for if this were the cause of the loss, the power required per revolution of the propeller would be reduced below that required where no loss existed. This, however, is not the case, as the same formula for apparent slip is used when the loss exists as when it does not, the apparent slip in both cases depending, among other factors, directly upon the power applied to the propeller, but not upon the fore-and-aft thrust delivered by the propeller.

Wake Gain. As a vessel moves forward through the water she displaces the water which again flows in behind her. Some of this replacement flow occurs from forward, but not all of it, as the hull of the vessel prevents the full supply from being provided in that direction. The remainder of the flow comes from aft and results in reducing the relative velocity of hull to water below that between hull and distant water. This decrease in relative velocity is known as **Wake.** The actual slip of the propeller which works

in this wake is known as the **real slip**, while the slip as measured between the speed of advance of the screw as it would be if working in a solid non-yielding nut and its actual speed of advance is known as the **apparent slip**. The ratio between the apparent slip and the *real* slip may be expressed as follows:

$$\frac{\text{Apparent slip}}{\text{Real slip}} = 1 - \text{Wake (in per cent of speed of advance of vessel).}$$

The effect of the wake on the propeller varies in degree according to the position of the propeller relative to the hull. The general effect is to reduce the revolutions of the propeller below what they would be with equal power applied should no wake exist and at the same time to increase the thrust delivered by the propeller, thus tending to nullify some of the malign effect of thrust deduction. The closer in to the after body of the hull and the farther aft within the limits of the hull underwater body length the propeller is located, the greater will be the thrust deduction loss, but the greater will be the wake and the consequent wake gain. Calling the power required to drive a vesssel without either *thrust deduction* or *wake gain*, unity, and using t to represent *thrust deduction and* w to represent *wake gain*, the factor by which unity power must be multiplied on account of the existence of these loss and gain factors is represented by

$$K = 1 + t - w.$$

SLIP BLOCK COEFFICIENTS

Coefficients of the Hull. The hull coefficients in common use for the comparison of hulls of vessels are four in number: **1. Block Coefficient** = (Displacement in tons × 35) ÷ (Length on Load-water-line × Greatest immersed beam × Mean draught), all of these latter dimensions being in feet.

2. Prismatic Coefficient = (Displacement in tons × 35) ÷ {Area of immersed midship section (square feet) × Length on Load-water-line (in feet)}; also called longitudinal coefficient.

3. Vertical Prismatic Coefficient = (Displacement in tons × 35) ÷ {Area Load-water-plane (square feet) × Mean draught (in feet)}.

4. Midship Section Coefficient = (Area of Immersed Midship Section (in square feet)} ÷ {Greatest immersed beam (in feet) × Mean draught (in feet)}.

Thrust deduction, wake gain, and **revolutions** of the propeller for any given power of engine and any speed of vessel **depend upon the after body lines of ship** under consideration and therefore upon the above coefficients. It would be preferable, however, to use the coefficients for the after body of the ship only, if these were available, but this is not the case at the present day. **In determining approximately the lines of the ship which affect the propeller performance, the Block Coefficient and Midship Section Coefficient are used.**

If for a given length of middle body of the ship the afterbody lines were of constant form and the midship section coefficient were constant for each particular *Block Coefficient*, it would be a simple matter to prepare charts from which the standard block coefficient could be picked off, as by holding the two given factors constant the length of the ship can be increased or decreased by varying the length of the middle body and yet the conditions of flow of the water to the propeller remain constant. The *Block Coefficient* of the ship would, however, increase or decrease with the length of the middle

body and very nearly as the Beam (in feet) + Length on Load-water-line (in feet).

The trials of such a standard set of vessels have been used in obtaining the factors for use in the design of propellers, and the hull coefficients of Block and Midship Section used in laying down a system by which the Standard (Slip) Block Coefficient of hulls of ordinary orthodox form obtained.

The Equations for estimates of Standard Slip Block Coefficients for various conditions are as follows:

Wing Screws

Class 1. Propellers located comparatively close to the hull and the hull of such form that a vertical line through the centre of the propeller pierces the skin of the ship at or below the water line, also where propellers are of small diameter and located well forward of the stern post, the vertical through the centre of hub piercing the skin of the ship at or below the water line.

Formula A. Slip Block Coefficient ($S.B.C.$) =

$$\frac{\text{Beam} + \text{Length on Load Water Line}}{0.28\,(1 - \text{Block Coef.}) + (\text{Beam} + \text{Length on Load Water Line})}$$

$$\frac{B + L.L.W.L.}{0.28\,(1 - B.C.) + (B + L.L.W.L.)},$$

or, for close approximation to the above, Length between Perpendicular, $L.B.P.$, may be substituted for $L.L.W.L.$, then,

$$S.B.C. = \frac{B + L.B.P.}{0.28\,(1 - B.C.) + (B + L.B.P.)}.$$

Class 2. Where the after body of the hull fines so rapidly in the athwartship dimension that the vertical through the hub centre passes entirely clear of the ship or pierces the hull well above the water line, the diameter of the propeller being greater than 0.75 × normal draught, the equation is, as before

$$(A)\ S.B.C. = \frac{B + L.L.W.L.}{0.28\,(1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.28\,(1 - B.C.) + (B + L.B.P.)}.$$

Class 3. Where the hull is as in (1), but the propellers are small in diameter and located aft in way of stern post, the vertical through centre of hub piercing the submerged hull as in (1), the equation becomes

$$(B)\ S.B.C. = \frac{B + L.L.W.L.}{0.248\,(1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.248\,(1 - B.C.) + (B + L.B.P.)}.$$

Class 4. Where the after body is as in (2), but the propellers are located aft and close to the stern post and are of a diameter less than 0.7 normal draught of the ship, the propeller being located very low so that its lower blades pass close to or below the base line of the ship, the equation becomes

$$(C)\ S.B.C. = \frac{B + L.L.W.L.}{0.567\,(1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.567\,(1 - B.C.) + (B + L.B.P.)}.$$

Class 5. Where the after body of the hull fines very rapidly in the vertical, but very slowly in the water line plane, the vertical through the hub centres piercing the hull of the vessel well above the water line, the equation becomes

$$(1)\ (A)\ S.B.C. = \frac{B + L.L.W.L.}{0.28\ (1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.28\ (1 - B.C.) + (B + L.B.P.)}.$$

Single Screw Ships

Class 6. Where the hull is of such form that a vertical through the hub centre pierces the hull well above the water line, the hull being of not less than 20 feet draught at the stern and the propeller not less than 0.75 × draught in diameter, or (2) where the hull is of such form that the propeller is completely shrouded by the load water plane of the ship, the equation for the Slip Block Coefficient is

$$(E)\ S.B.C. = \frac{B + L.L.W.L.}{0.164\ (1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.164\ (1 - B.C.) + (B + L.B.P.)}.$$

Class 7. Where the hull is of such form that the vertical through the hull centre pierces the hull well above the water line and where the draught at the stern is less than 14 feet or the diameter of the propeller is less than 0.7 the draught at the stern, the equation takes the form

$$(C)\ S.B.C. = \frac{B + L.L.W.L.}{0.567\ (1 - B\ C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.567\ (1 - B.C.) + (B + L.B.P.)}.$$

Submarines

Class 8. Single-hull type with propellers located abaft and clear of hull,

$$(F)\ S.B.C. = \frac{B + L.L.W.L.}{0.064\ (1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.064\ (1 - B.C.) + (B + L.B.P.)}.$$

Class 9. Double-hull type with propellers located beneath hull.

$$(G)\ S.B.C. = \frac{B + L.L.W.L.}{0.044\ (1 - B.C.) + (B + L.L.W.L.)} = \frac{B + L.B.P.}{0.044\ (1 - B.C.) + (B + L.L.W.L.)}.$$

Tunnel Boats

Class 10. The S.B.C. will be, at the greatest, that found by using formula (*A*), and may be still less. Data on the performance of such vessels are very meager at the present time.

Correction of S.B.C. for Variation of Midship Section Coefficient from Standard

The Standard Midship Section Coefficients agree with the values of the S.B.C. as found by equation (*A*). Should there be a wide variation between the actual and the standard, certain corrections should be made to conditions existing under Classes 1, 3, 4, 5, 6 and 7 to obtain the proper S.B.C.

The values of the Standard Midship Section Coefficients corresponding to the values of S.B.C. as obtained by equation (*A*) are

Table 1

S.B.C.	.95	.9	.85	.8	.75	.7	.65	.6	.55	.5	.45	.4	.35	.3
Mid. Sec. Coef.	.996	.994	.992	.99	.988	.985	.981	.978	.952	.891	.803	.712	.625	.535

To illustrate the method of correction for variation in midship section from standard, suppose the S.B.C. as found by equation (*A*) be 0.6, but the midship section coefficient be 0.95, then the corrected S.B.C. for the class 1 Condition will be

$$S.B.C.\ (\text{Class } 1) = 0.6 \times \frac{0.978}{0.95} = 0.608.$$

Now suppose the vessel be single-screw of Class 6, where

$$(E)\quad S.B.C. = \frac{B + L.L.W.L.}{0.164\ (1 - B.C.) + (B + L.L.W.L.)},$$

$$(A)\quad 0.608 = \frac{B + L.L.W.L.}{0.28\ (1 - B.C.) + (B + L.L.W.L.)},$$

then

$$S.B.C. = \frac{0.608 \times [0.28\ (1 - B.C.) + (B + L.L.W.L.)]}{0.164\ (1 - B.C.) + (B + L.L.W.L.)}$$

which will be the corrected Slip Block Coefficient for the single screw vessel of this same form.

DETERMINATION OF THRUST DEDUCTION FACTOR

As already stated, the Thrust Deduction Factor K for any hull depends upon the form of the after body of the hull, the Slip Block Coefficient, and upon the position of the propeller relative to the hull; thus the S.B.C. to use in determining the value of K for vessels of Classes as given under section on Slip Block Coefficient, are

Class 1, is that found by Equation (*A*) corrected for variation in midship section coefficient from Standard.

Class 2, is that found by Equation (*A*).

Class 3, is that found by Equation (*B*) corrected for variation in midship section coefficient from Standard.

Class 4, is that found by Equation (*C*) corrected for variation in midship section coefficient from Standard.

Class 5, (1), is that found by Equation (*A*) corrected for variation in midship section coefficient from Standard.

Class 5, (2), is that found by Equation (*D*) corrected for variation in midship section coefficient as before.

Class 6, (1) and (2), is that found by Equation (*E*) corrected as for class (5).

Class 7, is that found by Equation (*E*) corrected as for class 5.

Class 8, the value of K is unity.

Class 9, is unity when good hull clearance and good distance between propellers is maintained. When close to hull use Equation (G).

Class 10, is of constant value and $K = 1.195$.

MEAN RELATIVE TIP CLEARANCE OF PROPELLERS

In vessels of Classes 1, 3, 4, 6 (2), and 9 the relative value of K is variable and is dependent upon the relative mean tip clearance between the propeller blade tip in closest position to the hull and the hull, the equation for finding the relative mean tip clearance being

$$\text{M.T.C.} = \frac{H + 10H_1}{11}, \text{ where}$$

H = (Actual immersion, in feet, of tip of upper blade of propeller when in the vertical position × 14) + (Actual height of this tip above horizontal base line of ship, in feet, this base line being a line drawn tangent to the keel at its greatest immersion.)

H_1 = (Actual minimum tip clearance between circumference of propeller disc and hull × 10) + (Actual height of centre of hub of propeller, in feet, above the base line.)

The value of K for vessels of these classes for different values of *S.B.C.* as found above and for varying values of *M.T.C.* are given in Table 2.

Table 2. Values of K for M.T.C. and S.B.C.

S.B.C.	Relative mean tip clearance in feet						
	1.5	2.0	2.5	3.0	3.5	4.0	4.5
	Values of K*						
0.30	1.000	1.000	1.0000	1.000	1.000	1.000	1.00
0.35	1.000	1.000	1.0000	1.000	1.000	1.000	1.00
0.40	1.070	1.000	1.0000	1.000	1.000	1.000	1.00
0.45	1.093	1.030	1.0000	1.000	1.000	1.000	1.00
0.50	1.125	1.076	1.0050	1.000	1.000	1.000	1.00
0.55	1.175	1.100	1.0675	1.000	1.000	1.000	1.00
0.60	1.250	1.140	1.0900	1.064	1.010	1.000	1.00
0.65	1.340	1.190	1.1250	1.086	1.064	1.025	1.00
0.70	1.450	1.275	1.1830	1.115	1.085	1.066	1.04
0.75	1.580	1.370	1.2550	1.175	1.112		
0.80	1.750	1.470	1.3400	1.240			
0.85	2.000	1.600	1.4250				
0.90	2.260	1.750					

* For submarines of the Lake type on the surface, the value of K is as taken from this Table 2. For the submerged condition, the loss will only be ½ as great, thus if K on the surface = 1.34, in the submerged condition it will equal 1.17. With comparatively large tip clearances between propellers and hull and between adjacent propellers, K for this class vessel may safely be assumed as unity.

A good general rule to follow is to **keep the tips of the blades as far from the hull body as possible and yet close enough in to the hull**

to retain the full value of the wake while at the same time the shaft lines are kept parallel to the direction of motion of the vessel so that the currents of water entering the propeller are given an opportunity to straighten out and enter the propeller in a direction more nearly perpendicular to its disc, while the direction of thrust of the propeller will be as nearly opposite to the direction of motion of the vessel as possible.

For shallow-draft vessels such as destroyers the actual tip clearance should not be less than two feet, while for vessels of deep draught where the propeller is overhung by the immersed body of the hull, the actual tip clearance should be not less than three feet, and more if possible.

The above Tables of K values are for Standard projected area ratio form blades. Where blades are of the forms known as **Fan, Broad Tipped,** or **Narrow Tipped,** as described later, these values as given for vessels in which the values of K depend upon the relative mean tip clearance, may be considerably modified, increasing for the first two types and decreasing for the third type, the actual value for the first two forms being the mean between that for the actual tip clearance and that for the Basic propeller from which these other forms are deduced.

Table 3. Values of K for Vessels of Classes 2 and 6 (1)

S.B.C.	0.600	0.625	0.650	0.675	0.700	0.725	0.75
K	1.000	1.040	1.065	1.080	1.093	1.115	1.16

S.B.C.	0.775	0.800	0.825	0.850	0.875	0.900	0.925
K	1.200	1.270	1.340	1.425	1.570	1.780	2.400

Table 4. Values of K for Vessels of Class 5

S.B.C.	0.40	0.45	0.50	0.55	0.600	0.65	0.70	0.75
K	1.00	1.01	1.07	1.10	1.135	1.19	1.28	1.40

S.B.C.	0.80	0.85	0.90
K	1.58	1.94	3.20

Table 5. Values of K for Vessels of Class 7

S.B.C.	0.55	0.60	0.65	0.700	0.75	0.800	0.85	0.900
K	1.00	1.07	1.40	1.175	1.27	1.385	1.54	1.775

RESISTANCE OF HULL APPENDAGES

The **hull appendages usually fitted** depend upon the character of the vessel and are generally as follows:

1. Battleships, multiple propellers: Rudder, bilge keels, shaft struts and docking keels.
2. Merchant ships, multiple propellers: Rudder, bilge keels, shaft struts or spectacle frames.
3. Fast and light vessels, multiple propellers: Rudder, shaft struts and, in some cases, bilge keels and injection scoops.
4. Single-screw vessels: Rudder and rudder post and, in large vessels, bilge keels.

Table 6. Appendage Resistance in Per Cent of Bare Hull for Battleships and Destroyers

D.K.—Docking Keel. B.K.—Bilge Keel. Rud.—Rudder. Sh.—Shaft St.—Strut.			2 Sh. 2 St. 4 Sh. 4 St. B.K. & Rud.	2 Sh. 4 St. of fine section. B.K. D.K. & Rud.	2 Sh. 4 St. of heavy section. B.K. D.K. & Rud.	Destroyers 2 Sh. 2 St. B.K. Rud. & scoops
$\frac{v}{\sqrt{L.L.W.L.}}$	B.K. & Rud.	2 Sh. 2 St. B.K. & Rud.				
0.3	0.024	0.054	0.093	0.105	0.128	0.093
0.4	0.028	0.065	0.115	0.129	0.156	0.115
0.5	0.034	0.083	0.146	0.164	0.195	0.146
0.6	0.044	0.118	0.192	0.216	0.258	0.192
0.7	0.059	0.170	0.260	0.279	0.312	0.260
0.8	0.060	0.139	0.202	0.224	0.282	0.202
0.9	0.021	0.080	0.133	0.154	0.213	0.133
1.0	0.008	0.057	0.098	0.115	0.153	0.098
1.1	0.010	0.055	0.094	0.109	0.133	0.094
1.2	0.020	0.070	0.117	0.127	0.143	0.117
1.3	0.028	0.080	0.120	0.131	0.147	0.120
1.4	0.019	0.066	0.104	0.116	0.132	0.104
1.5	0.004	0.051	0.083	0.093	0.104	0.083
1.6	0.004	0.050	0.080	0.090	0.100	0.080
1.7*	0.004	0.050	0.080	0.090	0.100	0.080

*From this point the appendage resistance remains constant, probably falling for still higher values.

Table 7. Appendage Resistances in Per Cent of Bare Hull, Merchant Vessels

$v + \sqrt{L.L.W.L.}$	0.400	0.500	0.600	0.700	0.800	0.90	1.00
App. resis., per cent	0.188	0.138	0.115	0.105	0.101	0.10	0.10

The resistance of the appendages appears to be a varying percentage of the bare hull resistance, based on the values of **v (Speed in knots) $+ \sqrt{\text{L.L.W.L.}}$**

For vessels of lines similar to battleships and destroyers Table 6 gives the approximate values of the appendage resistance in per cent of the bare hull resistance:

For vessels of Merchant Type, given as Class 5 under discussion of S.B.C., where the appendages are usually Rudder, Struts, and Bilge Keels, the percentage appendage resistances appear to be as given in Table 7.

These percentages from either Table **are additive** to the bare hull resistance and must be applied where the total effective horsepower required to tow the vessel is given as the base on which to design the propeller and estimate the necessary power. Any abnormal appendages must be estimated independently and added to the total e.h.p. as above obtained.

BASIC CONDITION FOR ANALYSIS AND DESIGN OF SCREW PROPELLERS

So long as the blade sections of propellers approximate closely to the standard form of blade section described later, all propellers, no matter what the form of projected area ratio, may be reduced to a standard form of this ratio, and the performance of this standard formed propeller be estimated

Table 8. Basic Condition of Analysis and Design of Screw Propellers (Three Blades)

$\frac{P.A.}{D.A.}$	$I.T_D.$ lbs.	T.S. ft. per minute	P.C. per cent	Slip Block Coefficient										
				0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
							I–S							
.15	1.10	2830	69.9	.8	.809	.827	.85	.869	.889	.902	.929	.95	.97	.992
.175	1.45	3520	70.5	.8	.809	.827	.849	.869	.887	.902	.929	.95	.97	.992
.2	1.85	4250	70.9	.799	.809	.827	.848	.868	.884	.902	.928	.949	.969	.991
.225	2.28	4880	70.7	.798	.809	.826	.846	.867	.883	.902	.927	.948	.968	.990
.25	2.76	5500	70.1	.796	.808	.824	.843	.865	.882	.901	.924	.946	.967	.989
.275	3.25	6100	69.2	.794	.808	.823	.842	.863	.881	.900	.922	.944	.964	.988
.300	3.75	6650	68.2	.792	.807	.821	.841	.861	.880	.998	.920	.941	.962	.988
.325	4.27	7150	66.7	.789	.803	.820	.840	.860	.878	.896	.918	.940	.960	.984
.350	4.8	7660	65.2	.785	.801	.817	.838	.858	.876	.894	.915	.938	.958	.981
.375	5.4	8120	63.7	.781	.799	.814	.835	.854	.873	.891	.912	.935	.955	.978
.400	6.04	8600	61.9	.777	.796	.810	.831	.850	.870	.888	.908	.931	.951	.975
.425	6.65	9050	60.3	.773	.791	.806	.827	.846	.866	.884	.904	.927	.947	.972
.450	7.32	9530	58.6	.769	.786	.801	.823	.842	.861	.880	.900	.923	.943	.969
.475	8.03	10020	56.9	.763	.780	.796	.818	.837	.855	.875	.894	.919	.939	.963
.500	8.75	10550	55.4	.756	.773	.791	.811	.831	.849	.969	.888	.913	.933	.956
.525	9.50	11150	53.9	.748	.766	.784	.804	.824	.841	.860	.881	.905	.927	.950
.550	10.3	11850	52.6	.739	.758	.775	.796	.813	.832	.851	.873	.897	.917	.940
.575	11.1	12620	52.5	.729	.747	.765	.784	.802	.822	.840	.863	.885	.905	.928
.600	11.97	13550	52.5	.715	.735	.752	.771	.790	.807	.827	.848	.871	.870	.913
.625	12.83	14580	52.5	.700	.720	.736	.755	.772	.792	.810	.831	.851	.852	.893
.650	13.71	15720	52.5	.682	.700	.717	.735	.753	.772	.792	.810	.831	.830	.873

when working under a given fixed condition of resistance, and by means of coefficients obtained from actual performances of many propellers behind many different varieties of hulls, the proper corrective factors to obtain the performances under any other conditions of resistance can be estimated and applied.

This condition of fixed resistance is known as the Basic Condition of Design and the Propellers and Data conforming to it are known as Basic. The Table of these conditions (Table 8) gives for increasing values of Projected Area Ratios for three-bladed propellers, propellers of this number blades having been used in obtaining the data, the following values:

1. $I.T._D$ = Basic indicated thrust in pounds per square inch of disc area, including the hub.

2. $T.S.$ = Basic tip speed of propeller in feet per minute.

3. $1 - S = 1 -$ Basic. Apparent Slip for any $S.B.C.$, the apparent slip being in per cent. As the conditions are fixed, the relation between the real slip and the apparent slip for any $S.B.C.$, $I.T._D$ and $T.S.$ will be constant and therefore the apparent slip may be substituted for the real slip in the computations

4. $P.C.$ = Basic Propulsive Coefficient in per cent.

(*A*) From the values given in Table 8, having given P (=Pitch of Propeller, in feet), D (= Diameter of Propeller, in feet), $\frac{P.A.}{D.A.}$ (= Projected Area Ratio of 3 Blades of the propeller), $4/3\ \frac{P.A.}{D.A.}$ (=Total Projected Area Ratio—4 blades), $2/3\ \frac{P.A.}{D.A.}$ (= Total Projected Area Ratio—2 blades), the values $I.H.P.$ (= Basic Indicated Horsepower), $E.H.P.$ (= Basic Effective (Tow Rope) Horsepower) and $S.H.P.$ (= Basic Shaft Horsepower) Can be obtained by the following formulæ:

$$I.H.P. = \frac{D^2 \times I.T._D \times P \times R}{C}$$

$$C = \begin{cases} 252.4 \text{ for 4-bladed propeller.} \\ 291.8 \text{ for 3-bladed propeller.} \\ 389.0 \text{ for 2-bladed propeller.} \end{cases}$$

$S.H.P. = I.H.P. \times 0.92$ for Standard Conditions.

$E.H.P. = I.H.P. \times P.C.$, where $P.C.$ is the propulsive coefficient taken from Table 8 for the total projected area ratio, all other data being taken for $P.A. \div D.A.$, no matter what the number of blades.

In addition may also be obtained the values V = Basic speed of the ship, R = Basic revolution of the propeller, where

$$V = \frac{R \times P\,(1 - S)}{101.33} \text{ and } R = \frac{T.S.}{\pi D}.$$

(*B*) From this same Basic data as given in Table 8, should it be desired to design the propeller for Basic conditions of resistance, and having given $I.H.P.$ or $S.H.P.$ and V, and assuming any one or several values of $P.A. \div D.A.$, the values D, P, R, and $P.C.$ can be obtained, as the value of $P.C.$ for $P.A \div$ D.A. or for the total projected area ratio is fixed by the Table, while

$$P \times R = \frac{V \times 101.33}{1 - S},$$

$$D = \sqrt{\frac{C \times I.H.P.}{I.T._D \times P \times R}},\ C \text{ having same values as before,}$$

$$P = \frac{\pi \times D \times 101.33 \times V}{T.S. \times (1 - S)} \text{ and } R = \frac{T.S.}{\pi D}.$$

$E.H.P. = I.H.P. \times P.C.$ as before and $S.H.P. = I.H.P. \times 0.92$ as before.

Should the mechanical efficiency be greater or less than 0.92 say A, then the actual indicated horsepower required will not be $I.H.P.$ but will be $I.H.P. \times 0.92 + A$, while the $E.H.P.$ and the $S.H.P.$ will remain as before.

Should Thrust Deduction exist, the actual value of the Indicated Horsepower will be $K \times I.H.P.$, that of the shaft horsepower $K \times S.H.P.$, while that of $E.H.P.$ will remain $I.H.P. \times P.C.$

The apparent slip in this latter case will become KS instead of S, and the revolutions will be

$$R_d = \frac{V \times 101.33}{P \times (1 - KS)}$$

(C) Should $E.H.P.$ and V be given, the formulæ take the following forms:

$I.H.P. = E.H.P. + P.C.$

K = Thrust Deduction Factor.

$K \times I.H.P.$ = Total Indicated Horsepower.

$K \times I.H.P. \times 0.92$ = Indicated Shaft Horsepower.

$K \times I.H.P. \times \frac{0.92}{A}$ = Total Indicated Horsepower for engine mechanical efficiency differing from 0.92.

$$D = \sqrt{\frac{C_1 \times E.H.P.}{E.T._P \times \frac{P.A.}{D.A.} \times V}}, \text{ where}$$

$C_1 =$ { 2.491 for 4-bladed propellers. 2.88 for 3-bladed propellers. 3.84 for 2-bladed propellers.

$$E.T._p = \frac{P.T._p}{1 - S}.\quad P.T._p = I.T._p \times P.C. \text{ and } I.T._p = I.T._D + \frac{P.A.}{D.A.}.$$

Should it be desired to use the value of $I.H.P.$ as obtained above, the equation for diameter becomes

$$D = \sqrt{\frac{C_1 \times I.H.P. \times (1 - S)}{I.T._D \times V}},\ \text{where } C_1 \text{ has the values given above.}$$

P, R and R_d are found as before.

(D) Finally, in addition to $I.H.P.$ or $S.H.P.$ and V in (B) or $E.H.P.$ and V in (C) there may also be assumed the value of D. In such cases the value of $P.A. + D.A.$ and the total value of projected area ratio will be found in

addition to P, R, R_d, and the power values sought. The equations become either

$$I.T._D + (1 - S) = \frac{C_1 \times I.H.P.}{D^2 \times V} \text{ for } (B), \text{ or}$$

$$E.T._P \times \frac{P.A.}{D.A.} = \frac{C_1 \times E.H.P.}{D^2 \times V} \text{ for } (C), \text{ the values of}$$

$I.T._D + (1 - S)$ for different values of $P.A. + D.A.$ being given in Table 9, while those for $E.T._P \times \frac{P.A.}{D.A.}$ for different values of $P.A. + D.A.$ are given in Table 10.

Table 9. Values of $I.T._D + (1 - S)$ for Varying Values of P.A. + D.A.

S.B.C.	Values of $P.A. + D.A.$									
	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6	0.65
	Values of $I.T._D + (1 - S)$									
0.90	1.93	2.87	3.89	5.04	6.31	7.80	9.37	11.20	13.43	15.57
0.85	1.95	2.90	3.93	5.10	6.38	7.89	9.50	11.35	13.61	15.71
0.80	1.97	2.93	3.97	5.16	6.45	7.96	9.58	11.46	13.74	15.94
0.75	2.00	2.97	4.02	5.22	6.52	8.08	9.71	11.65	13.93	16.14
0.70	2.02	3.00	4.07	5.29	6.61	8.18	9.85	11.79	14.11	16.30
0.65	2.05	3.04	4.12	5.35	6.69	8.27	9.98	11.94	14.28	16.54
0.60	2.07	3.08	4.17	5.42	6.77	8.37	10.07	12.08	14.47	16.75
0.55	2.09	3.11	4.21	5.48	6.85	8.45	10.21	12.23	14.65	16.96
0.50	2.11	3.14	4.26	5.53	6.90	8.55	10.30	12.36	14.81	17.18
0.45	2.14	3.17	4.30	5.59	6.98	8.65	10.41	12.51	14.95	17.39
0.40	2.16	3.20	4.35	5.65	7.06	8.75	10.53	12.65	15.15	17.59
0.35	2.19	3.24	4.40	5.71	7.14	8.86	10.66	12.79	15.32	17.81
0.30	2.21	3.29	4.45	5.79	7.23	8.96	10.79	12.95	15.52	18.05

Table 10. Values of (P.A. + D.A.) × E.T.$_P$ for Varying Values of P.A. + D.A.

Slip B.C.	Values of $P.A. + D.A.$									
	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6	0.65
	Values of $(P.A. + D.A.) \times E.T._P$									
0.90	1.370	2.01	2.65	3.29	3.90	4.57	5.19	5.89	7.05	8.17
0.85	1.380	2.04	2.68	3.32	3.95	4.62	5.25	5.96	7.15	8.25
0.80	1.400	2.06	2.71	3.36	3.99	4.67	5.31	6.03	7.21	8.37
0.75	1.420	2.09	2.74	3.40	4.04	4.73	5.37	6.13	7.31	8.47
0.70	1.430	2.11	2.77	3.45	4.09	4.79	5.44	6.20	7.41	8.56
0.65	1.450	2.14	2.81	3.49	4.14	4.85	5.51	6.28	7.50	8.69
0.60	1.470	2.16	2.84	3.52	4.19	4.91	5.58	6.36	7.60	8.79
0.55	1.485	2.19	2.87	3.57	4.24	4.95	5.64	6.43	7.69	8.91
0.50	1.500	2.21	2.90	3.61	4.27	5.01	5.70	6.50	7.77	9.02
0.45	1.515	2.23	2.93	3.65	4.32	5.07	5.76	6.58	7.85	9.13
0.40	1.530	2.25	2.97	3.68	4.37	5.13	5.84	6.65	7.95	9.24
0.35	1.550	2.28	3.00	3.72	4.42	5.19	5.91	6.73	8.04	9.35
0.30	1.570	2.31	3.04	3.77	4.48	5.25	5.98	6.81	8.15	9.48

Corrective Factors for Power for Other Than Basic Conditions of Design

Suppose the Basic Conditions of any given propeller be
$I.H.P.$ = Basic Indicated Horsepower,
$E.H.P.$ = Basic Effective (tow rope) Horsepower,
S = Basic Apparent slip for $S.B.C.$ of hull to which propeller was applied,
V = Basic Speed in knots.

Suppose further that this propeller working behind the given hull delivered a speed
v = Actual speed in knots.
$I.H.P._d$ = Actual Indicated horsepower,
A = Actual percentage mechanical efficiency of the engine,
K = Thrust deduction factor for type of hull and position of propeller,
$I.H.P._p = I.H.P._d \times \frac{A}{0.92} \div K$ = Actual power delivering net effective horsepower in propelling the vessel.
$e.h.p.$ = Net effective horsepower required for speed v.
$e.h.p._1$ = Gross effective horsepower corresponding to

$$I.H.P._d \times \frac{A}{0.92}.$$

(1) Being given $I.H.P._d$ to find $e.h.p.$ and R_d, where R_d = Estimated revolutions per minute. $10^s = I.H.P. + I.H.P._p$, and $e.h.p + E.H.P.$ = Power percentage corresponding to Z from Table 12,
$e.h.p. = E.H.P. \times (e.h.p + E.H.P.)$ = net effective horsepower used in propulsion.

$$10^s = I.H.P. + \left(I.H.P._d \times \frac{A}{0.92}\right).$$

$e.h.p._1 + E.H.P.$ = Power percentage corresponding to Z_1 from Table 12'
$e.h.p._1$ = Gross effective horsepower for which water must be supplied to the propeller.
$e.h.p._1$ must never be so great that $e.t._1 > 1.15\ E.T.$, where

$$e.t._1 = \frac{e.h.p._1 \times 33{,}000}{v \times 101.33}.$$

while

$$E.T. = \frac{E.H.P. \times 33{,}000}{V \times 101.33}.$$

This requirement is necessary to guard against cavitation of the suction to the propeller.

To guard against breaking down or dispersion of the thrust column, the values of $e.t.$ where $e.t. = \frac{e.h.p. \times 33{,}000}{v \times 101.33}$, should never exceed certain values which values correspond with the values of $v + V$ and $e.h.p. + E.H.P.$ given in Table 11.

In designing, the propeller should always, if possible, have such characteristics that the value of $v + V$ for any load $e.h.p. + E.H.P.$ will be greater

than the value of $v \div V$ given in Table 11 for the same value of *e.h.p.* ÷ *E.H.P.*

Corrective Equation for Revolutions

The revolutions to be expected with any power and speed with any hull, the propeller working under other than basic conditions of resistance, can be obtained as follows:

$$\text{Apparent slip under actual conditions} = s = S \times \frac{\text{I.H.P.}_d \times V^v}{\text{I.H.P.} \times v^v}.$$

Where $I.H.P._d$ and v are the actual power and speed while *I.H.P.*, V and S are the power, speed and apparent slip under basic conditions.

This equation may also be written in terms of shaft horsepower or in terms of K and Z, thus

$$s = S \frac{S.H.P._d \times V^v}{S.H.P. \times v^v} = S \frac{K \times V^v}{10^z \times v^v},$$

but the latter form should only be used where there is no danger of cavitation or of dispersal of thrust columns.

V^v and v^v are written as A_V and A_v, and Table 13 gives the values of Log A_V in all cases, and of Log A_v in cases of vessels of Classes 1, 2, 3, 4, 5, 6, 7, 8 on the surfaces, and 10, these cases being those given under *S.B.C.*

Table 11

$\frac{e.h.p.}{E.H.P.}$	.05	.075	.1	.15	.2	.25	.3	.35
$\frac{v}{V}$	.305	.36	.4	.46	.507	.545	.578	.605
$\frac{e.h.p.}{E.H.P.}$	.4	.45	.5	.55	.6	.65	.7	.75
$\frac{v}{V}$	.628	.65	.666	.68	.69	.7	.709	.716
$\frac{e.h.p.}{E.H.P.}$	.8	.85	.9	.95	1.0	1.05	1.10	1.15
$\frac{v}{V}$	.721	.74	.782	.825	.868	.912	.956	1.0
$\frac{e.h.p.}{E.H.P.}$	1.2	1.25	1.3	1.35	1.4	1.45	1.5	1.55
$\frac{v}{V}$	1.042	1.088	1.132	1.175	1.218	1.262	1.304	1.347
$\frac{e.h.p.}{E.H.P.}$	1.6	1.65	1.7	1.75	1.8	1.85	1.9	
$\frac{v}{V}$	1.392	1.435	1.478	1.522	1.567	1.61	1.652	

Table 12. Corrective Power Factor where

$$Z = -1.0414 \text{ Log } \left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right) \cdot \quad \text{Log I.H.P.}_p = \text{Log I.H.P.} + Z$$

$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z
.025	−1.668	.55	−.2704	1.15	+.0632	1.75	+.2531
.05	−1.355	.6	−.231	1.2	+.0825	1.80	+.2658
.075	−1.1715	.65	−.1948	1.25	+.1009	1.85	+.2782
.100	−1.0414	.7	−.1613	1.3	+.1186	1.9	+.2903
.15	− .858	.75	−.1301	1.35	+.1357	1.95	+.3023
.2	− .7279	.8	−.1009	1.4	+.1521	2.0	+.3135
.25	− .627	.85	−.0735	1.45	+.1681	2.05	+.3247
.3	− .5445	.9	−.0477	1.5	+.1834	2.10	+.3356
.35	− .4748	.95	−.0232	1.55	+.1982	2.15	+.3462
.4	− .4144	1.0	.0	1.6	+.2126	2.20	+.3566
.45	− .3611	1.05	+.0221	1.65	+.2265	2.25	+.3668
.5	− .3135	1.10	+.0431	1.7	+.2400	2.30	+.3767

For vessels of Class 8 submerged and of Class 9, both surface and submerged, the values of Log Av are given in Table 14.

Table 13. Logs Av for Vessels all Classes and Log A_v for Classes 1, 2, 3, 4, 5, 6, 7, 8 on the Surface and 10

Knots V and v.....	3	4	5	6	7	8	9	10
Log A.....	1.94	2.17	2.36	2.53	2.66	2.79	2.91	3.01
Knots V and v.....	11	12	13	14	15	16	17	18
Log A.....	3.12	3.23	3.33	3.43	3.52	3.60	3.68	3.76
Knots V and v.....	19	20	21	22	23	24	25	26
Log A.....	3.83	3.89	3.95	4.0	4.05	4.1	4.14	4.18
Knots V and v.....	27	28	29	30	31	32	33	34
Log A.....	4.22	4.25	4.28	4.31	4.34	4.36	4.38	4.4
Knots V and v.....	35	36	37	38	39	40	41	42
Log A.....	4.42	4.44	4.46	4.47	4.48	4.49	4.50	4.51
Knots V and v.....	43	44	45	46	47	48	49	50
Log A.....	4.52	4.53	5.54	5.55	4.56	4.57	4.58	4.59
Knots V and v.....	51	52	53	54	55	56	57	58
Log A.....	4.60	4.61	4.62	4.63	4.64	4.65	4.66	4.67
Knots V and v.....	59	60	61	62	63	64	65	66
Log A.....	4.68	4.69	4.70	4.71	4.72	4.73	4.74	4.75

CAVITATION

In discussing this subject, it must be divided into **two conditions**, as follows:

1. Suppose the propeller be designed for one particular condition of resistance of the vessel, then the performance curves of Power-speed and Revolutions-speed will all pass through the design point without cavitation. As the speed increases, however, the resistance increases with it, but in a greater

Table 14. Log A_s for Vessels of Class 8 Submerged, and of Class 9 both Surface and Submerged

Knots v	3	4	5	6	7	8	9	10
Log A	1.16	1.55	1.9	2.18	2.4	2.57	2.72	2.85
Knots v	11	12	13	14	15	16	17	18
Log A_s	2.96	3.07	3.17	3.26	3.35	3.44	3.52	3.6
Knots v	19	20	21	22	23	24	25	26
Log A_s	3.68	3.74	3.8	3.85	3.9	3.95	4.0	4.05
Knots v	27	28	29	30	31	32	33	34
Log A_s	4.08	4.09	4.15	4.18	4.21	4.24	4.26	4.28
Knots v	35	36	37	38	39	40	41	42
Log A_s	4.30	4.32	4.33	4.34	4.35	4.36	4.37	4.38
Knots v	43	44	45	46	47			
Log A_s	4.39	4.40	4.41	4.42	4.43			

degree, unless the vessel planes, until finally the necessary flow of water to the propeller to supply the thrust required becomes greater than that the combined atmospheric and hydraulic head on the water can produce. The result is that the propeller, instead of traveling smoothly along the suction column of the propeller, begins to cut across it and break the column apart, consuming power and producing heavy vibrations. This phenomenon is designated as **"Cavitation"** or **"Cavitation of the Suction Column."**

2. Suppose this same vessel be loaded down gradually so that with constant power the speed will gradually be reduced. This reduction in speed will be accomplished without any reduction in propulsive efficiency until a certain effective thrust is attained, after which any further reduction in speed will be accompanied by a reduction in propulsive efficiency. This phenomenon may be designated as **"Cavitation or Dispersal of the Thrust Column,"** and is accompanied by loss in efficiency and more or less vibrations, depending upon the amount of reduction of speeds below those corresponding to the limiting thrusts.

Both of these conditions for "Cavitation" are shown by Fig. 4, where the curve of critical (limiting) thrusts is shown, marked *C*, Curve of Critical Thrusts, *E.T.*

The well authenticated limits of safety are between the curves A and C, although values of $v + V$ above A may be used with probable safety, but to use such values in designing usually produces low values of $P + D$ and is therefore to be avoided. The curve A gives thrusts 30 per cent lower than those of C, while Curve B corresponds to thrusts 15 per cent lower than C. Curve D at values of $e.h.p. + E.H.P.$ up to 0.4 corresponds to conditions of maximum efficiency obtained by any vessels of which records are available, while the upper part of the curve from 0.9 upward gives these same conditions for records at these high loads. This curve approximates closely to the curve corresponding to "Powers varying as the cubes of the speeds." Actual performance curves, however, approximate very closely to Curves A, D, C and the curves of $M = \left(\frac{e.t.}{E.T.}\right)^x$ shown, in their characteristics. The point corresponding to $e.h.p. + E.H.P.$ = unity, and $v + V$ = unity, is the point of Basic Condition of the propeller.

Application of Fig. 4, to Condition (1)

Suppose a propeller be of such proportions for the work it has to perform as to plot in its performance well above the curve of critical thrusts. As the speed increases, the value of $e.t. + E.T.$ will increase, but so long as there is no abnormal increase in $e.t. + E.T.$ as compared with the increase in v, the performance curve extended will still plot in the safety zone until the approximate value $e.t. + E.T. = 1.15$ is reached. After passing this value, the combined hydraulic and atmospheric heads producing hydraulic flow to the propeller becomes insufficient and the propeller commences to cut across the water forming the suction supply column, cutting this column off in thin layers, so that there are a succession of these thin layers following each other through the length of the propeller. This discontinuity of the water supply produces the phenomenon commonly spoken of as "Cavitation." It produces a loss in power, gradually increasing in amount as the value of $e.t. + E.T.$ increases above 1.15, and vibration occurs increasing as the value $e.t. + E.T.$ increases.

Cavitation will have the following effects upon estimate of power where effective power is given, upon estimate of effective power where indicated or shaft horsepower is given, and upon the equation for apparent slip.

(*a*) **e.h.p. Given to Find I.H.P.$_p$ or S.H.P.$_d$.**

Suppose $e.h.p. + E.H.P. = 1.02$, and $v + V = 0.8$, $K = 1.1$. The value of M for these values of $e.h.p. + E.H.P.$ and $v + V$ is 1.111.

$$Z \text{ for } e.h.p. + E.H.P. = + 0.011 \text{ (Table 12).}$$

$$\text{Log } I.H.P._p = \text{Log } I.H.P. - Z,$$

$$\text{Log } I.H.P._d \text{ (without cavitation)} = \text{Log } I.H.P._p \times \text{Log } K,$$

$$\text{Log } I.H.P._d \text{ (with cavitation)} = \text{Log } I.H.P._p + \text{Log } K + \text{Log } M.$$

(*b*) **I.H.P.$_d$ Given to Find e.h.p.**, $v + V$ = known value, $e.h.p. + E.H.P.$ unknown. Calling the corrective Power Factor for that value of $e.h.p. + E.H.P.$ at the intersection of the value $v + V$ with the Critical Thrust Curve Z_1, and Z the value for the actual value of the $e.h.p. + E.H.P.$ delivered;—

$Z = Z_1 - \text{Log } M$, and M will have the value that satisfies this equation and the equation for Log $I.H.P._d$ as given below.

(*c*) **Either e.h.p. or I.H.P.$_d$ Given to Find Revolutions.**

$$s = S\frac{I.H.P._d \times Av}{I.H.P. \times A_s} = S\frac{M \times K \times Av}{10^x \times A_s},$$

where Z_1 = the value of Z corresponding to that value of $e.h.p. \div E.H.P.$ where $v + V$ crosses the curve of Critical Thrusts, and where Z for the actual value of $\frac{e.h.p.}{E.H.P.}$ being delivered is expressed by $Z = Z_1 + \text{Log } M$.

Application of Fig. 4 to Condition 2

Water in passing through a propeller has its direction of flow gradually changed from the direction with which it originally enters the propeller at the leading edges of the propeller blades, until on leaving the propeller it passes off in the direction of the generatrices of an infinite number of right hyperboloids of revolution, the outer limiting one having for its base the propeller disc. The efficiency of propulsion realized, will, among other causes, depend upon the distance of the minimum diameter of this limiting hyperboloid abaft the propeller and upon the area of this minimum cross-section; the more nearly the limiting hyperboloid approaches to a cylinder, the greater will be the efficiency of propulsion. As the resistance of a ship is increased by excessive loading, foul bottom or head winds and seas, the slower will be the speed realized for any given power and the closer to the propeller and the smaller in area will become the minimum belt area of the thrust hyperboloid. The efficiency of propulsion will remain practically constant throughout quite a large range of speeds until the speed becomes such that the thrusts for any given speed and effective horsepower plots the propeller upon the Curve of Critical Thrusts, *E.T.* Any further decrease in speed will be accompanied by a decrease in efficiency exactly as in *Condition* 1, and the same corrective formulas as given in the preceding section will apply.

GENERAL FORMULÆ FOR CORRECTION OF POWER FOR "CAVITATION" AND "DISPERSAL OF THE THRUST COLUMN"

$\frac{e.h.p.}{E.H.P.}$ = Actual Load Factor.

$\frac{e.h.p._1}{E.H.P.}$ = Load Factor corresponding to point on Critical Thrust Curve where $v + V$ crosses that curve.

Z = Power Exponent corresponding to $\frac{e.h.p.}{E.H.P.}$.

Z_1 = Power Exponent corresponding to $\frac{e.h.p._1}{E.H.P.}$.

Z and Z_1 regarded as positive.

(1) $\frac{e.h.p.}{E.H.P.} > 1$, $\frac{e.h.p._1}{E.H.P.} > 1$.

$\text{Log } M = Z - Z_1$.

$\text{Log } I.H.P._d = \text{Log } I.H.P. + \text{Log } K + 2Z - Z_1$.

(2) $\frac{e.h.p.}{E.H.P.} > 1$, $\frac{e.h.p._1}{E.H.P.} = 1$.

$\text{Log } M = Z$.

$\text{Log } I.H.P._d = \text{Log } I.H.P. + \text{Log } K + 2Z$.

(3) $\frac{e.h.p.}{E.H.P.} > 1$, $\frac{e.h.p._1}{E.H.P.} < 1$.

$\text{Log } M = Z + Z_1$.

$\text{Log } I.H.P._d = \text{Log } I.H.P. + \text{Log } K + 2Z + Z_1$.

(4) $\frac{e.h.p.}{E.H.P.} = 1, \quad \frac{e.h.p._1}{E.H.P.} < 1.$

$\text{Log } M = Z_1.$

$\text{Log } I.H.P._d = \text{Log } I.H.P. + \text{Log } K + Z_1.$

(5) $\frac{e.h.p.}{E.H.P.} < 1, \quad \frac{e.h.p._1}{E.H.P.} < 1.$

$\text{Log } M = Z_1 - Z.$

$\text{Log } I.H.P._d = \text{Log } I.H.P. + \text{Log } K + Z_1 - 2Z.$

ANALYSIS AND DESIGN OF PROPELLERS

By use of the following formulæ it is possible, having given the characteristics of a propeller, including form of projected area, and the general characteristics of the hull behind which the propeller is to work, including the curve of *e.h.p.* – Speed, to reduce the propeller to standard conditions and to make a very close estimate of the power and revolutions for any given speed under the conditions of resistance represented by the curve of *e.h.p.*

Conversely, having the general characteristics of a hull and the curve of *e.h.p.* – Speed, either from model tank trials or estimated, it is possible to design the propeller and estimate the power and revolutions for any required speeds, or having a fixed power and revolutions, to estimate the speed which will be obtained.

Forms for Analysis of Propeller

Form 1, Curve of effective (tow-rope) horsepower available.
Propeller blades of Standard Projected Area Ratio Form.

(1) D = Diameter of Propeller, in feet.

(2) P = Pitch of Propeller, in feet.

(3) $P.A. + D.A.$ = Total Projected Area Ratio of Propeller outside of 0.2 D circle.

(4) $P.A. + D.A.$ = Basic Projected Area Ratio = $\frac{1}{4}$ $(P.A. + D.A.)$ for 4-bladed propeller = $\frac{1}{3}$ $(P.A. + D.A.)$ for 2-bladed propeller.

(5) $T.S.$ = Basic Tip Speed of Propeller, in feet per minute for $P.A. + D.A.$ (Table 8).

(6) $P \times R = (T.S. \times P.) + \pi D.$

(7) $S.B.C.$ = Slip Block coefficient of hull of vessel to which propeller is applied; derived from equation for type of hull and relative position of propeller.

(8) $1 - S = 1 -$ Apparent Basic Slip, taken from Table 8, for $P.A. + D.A.$ and $S.B.C.$

(9) $V = \{P \times R \times (1 - S)\} + 101.33$ = Basic Speed, in knots per hour.

(10) $I.T._D.$ = Basic Indicated Thrust, in pounds per square inch of disc area, including the hub. Taken from Table 8, for $P.A. + D.A.$

(11) $I.H.P. = (D^2 \times I.T._D \times P \times R) \div C$ = Basic Indicated Horsepower,

where $C = \begin{cases} 389 & \text{– for 2-bladed propellers.} \\ 291.8 & \text{– for 3-bladed propellers.} \\ 252.4 & \text{– for 4-bladed propellers.} \end{cases}$

(12) $P.C.$ = Basic Propulsive Coefficient for $P.A. + D.A.$

(13) $E.H.P. = I.H.P. \times P.C.$ = Basic Effective (Tow-rope) Horsepower delivered by the propeller.

(14) $S.H.P. = I.H.P. \times 0.92$ = Basic Shaft Horsepower delivered to the propeller.

To Find Power for Any Desired Effective (Tow-rope) Power

(15) v = Any desired speed of ship.
(16) *e.h.p.* = Total net effective (tow-rope) horsepower to be delivered by one propeller for speed v.
(17) *e.h.p.* ÷ *E.H.P.* = Load factor.

(18) $z = -1.0414 \log\left(\frac{e.h.p.}{E.H.P}\right) = 1.0414 \log\left(\frac{E.H.P.}{e.h.p.}\right)$ = Power corrective factor for *e.h.p.* ÷ *E.H.P.* (Table 12).
(19) *I.H.P.*$_p$ or *S.H.P.*$_p$ = (*I.H.P.* or *S.H.P.*) ÷ 10^z = Indicated or Shaft horsepower required to deliver *e.h.p.* without *Thrust Deduction*.
(20) K = Thrust Deduction factor for *S.B.C.* as explained under section on Thrust Deduction and relative position of propeller.
(21) *I.H.P.*$_d$ or *S.H.P.*$_d$ = $K \times$ (*I.H.P.*$_p$ or *S.H.P.*$_p$) = Gross Indicated or Shaft horsepower required to deliver *e.h.p.*
(22) *p.c.* = *e.h.p.* ÷ *I.H.P.*$_d$ = Final propulsive coefficient.

To Find Revolutions

(23) Log Av = Speed Factor for V (Table 13).
(24) Log A_v = Speed Factor for v (Table 13).

(25) $s = S\frac{I.H.P._d \times Av}{I.H.P. \times A_v} = S\frac{S.H.P._d \times Av}{S.H.P. \times A_v}$ = when no cavitation exists,

$$S\frac{K \times Av}{10^z \times A_v} = \text{Apparent slip for speed } v.$$

(26) $R_d = \frac{101.33 \times v}{P \times (1 - s)}$ = Estimated Revolutions for

$$\text{Power} \left\{\begin{matrix} I.H.P._d \\ S.H.P._d \end{matrix}\right\} \text{and speed } v.$$

Variations from Above Form

(a) *Engine to be designed for a given power;* to find estimated speed and revolutions at that speed.
(15) *I.H.P.*$_d$ or *S.H.P.*$_d$ = Gross designed Power of Engine (or on one shaft).
(16) K = *Thrust Deduction* factor for *S.B.C.* and relative position of propeller.
(17) *I.H.P.*$_p$ or *S.H.P.*$_p$ = (*I.H.P.*$_d$ or *S.H.P.*$_d$) ÷ K = Net Power of Engine (or on one shaft) actually producing propulsive thrust.
(18) Z = (Log *I.H.P.* or Log *S.H.P.*) − (Log *I.H.P.*$_p$ or Log *S.H.P.*$_p$) = Power corrective factor.

(19) (*e.h.p.* ÷ *E.H.P.*) = $\frac{1}{10^z + 1.0414}$ = Load Factor. (Table 12.)

(20) $e.h.p. = E.H.P. \times (e.h.p. \div E.H.P.)$ = Net effective (tow-rope) horsepower delivered by one propeller.

(21) N = Number of Propellers.

(22) $N \times e.h.p.$ = Total effective horsepower delivered.

(23) v = Speed corresponding to $N \times e.h.p.$ from curve of effective tow-rope horsepower.

Using $I.H.P._d$ or $S.H.P._d$ and $I.H.P.$ or $S.H.P.$ and Logs Av, log A_v (the latter for the v of (23)), find s and R_d as before.

(*b*) *Engine to be designed for a given power.* Speed estimated at a given value, but no effective horsepower for this speed given. To find effective horsepower which will be delivered at this speed and the corresponding revolutions.

(15) $I.H.P._d$ or $S.H.P._d$ = Gross designed Power of Engine (or on shaft).

(16) K = Thrust Deduction factor for *S.B.C.* and relative position of propeller.

(17) $I.H.P._p$ or $S.H.P._p = (I.H.P._d$ or $S.H.P._d) \div K$ = Net Power of Engine (or on shaft) actually producing propulsive thrust.

(18) Z = (Log $I.H.P.$ or Log $S.H.P.$) − (Log $I.H.P._p$ or Log $S.H.P._p$) = Power corrective factor.

(19) $e.h.p. \div E.H.P.$ = Load factor corresponding to Z. (Table 12.)

(20) $e.h.p. = E.H.P. \times (e.h.p. \div E.H.P.)$ = Net Effective (tow-rope) horsepower delivered by one propeller.

(21) v = Given designed speed.

From the data of $I.H.P._d$ or $S.H.P._d$ and v, find s and R_d as before.

It is evident that the values of s and R_d will depend entirely upon the correctness of the estimates of power, slip block coefficient and speed. Should the power estimate be too low for the speed, the revolutions and speed obtained with the designed power of the engine will be lower than the designed revolutions and the estimated speed of the ship, and *vice versa*. Should the slip-block coefficient used be too heavy, the estimated revolutions will be too low.

The above forms are those to be used where blades are of Standard Form of Projected Area Ratio and where no cavitation of the suction column to nor dispersal of the thrust column from the propeller exists.

It will now be necessary to give the characteristics of the Standard Forms referred to.

Standard Forms of Projected Area Ratios

The projected area forms of all propellers, no matter of what contour, may be swung around in circular arcs until the areas are symmetrically arranged on each side of a centre line drawn perpendicular to the axis of the propeller shaft, and the original area of the projection and its radial distribution of surface will be retained. If a form of blade of constant percentage radial distribution of projected surface is adopted, no matter what percentage of the disc area represents the projected area, the relative distribution of projected surface will be a constant and therefore one of the most frequent variables met with in propeller design, a variable which seriously affects efficiency and revolutions, will be eliminated. To eliminate another variable to as great an extent as possible, the blade sections on the circular arcs through the projections have also been made of standard form as nearly as possible.

Standard Projected Area Forms

The Standard Projected Area Ratio Form adopted is that used for a large number of propellers in the United States Navy, whose records of per-

formances averaged very high and, in general, exceeded those given by any other forms.

These characteristics are set forth in Table 15, where with a propeller radius of unity, the chords of the arcs between the middle line of the projected area and either leading or following edge of the projection at each of the values 0.2, 0.3, 0.4, 0.5, etc., of the radius are given for each value of Projected Area Ratio of a **3-bladed propeller**, from $P.A. + D.A. = 0.15$ to $P.A. + D.A. = 0.7$.

Table 15. Table of Chords of Arcs between Middle Radial Line of Projection of Blade and Bounding Edge

P.A.+D.A.	Decimal parts of radius with chords below											
	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	0.925	0.95	0.975	0.9875
.15	.042	.064	.087	.107	.120	.128	.126	.105	.094	.08	.059	.042
.20	.055	.084	.114	.140	.159	.171	.168	.139	.125	.108	.078	.058
.25	.069	.107	.142	.173	.198	.212	.208	.174	.157	.133	.097	.069
.30	.082	.128	.170	.207	.236	.253	.250	.210	.190	.161	.120	.088
.35	.097	.150	.199	.242	.274	.295	.293	.245	.222	.192	.142	.105
.40	.109	.170	.227	.277	.316	.338	.333	.278	.251	.214	.158	.118
.45	.123	.191	.253	.308	.352	.378	.374	.313	.284	.243	.178	.127
.50	.136	.210	.279	.341	.389	.417	.415	.346	.312	.269	.195	.143
.55	.148	.230	.306	.372	.425	.459	.455	.382	.348	.293	.216	.165
.60	.159	.249	.332	.405	.463	.499	.497	.413	.377	.322	.238	.184
.65	.172	.270	.359	.437	.497	.539	.537	.445	.403	.343	.255	.200
.70	.184	.288	.382	.466	.533	.578	.575	.481	.437	.375	.280	.220

For a **4-bladed propeller** the standard form corresponding to the total projected area ratio will be that for three-fourths of that ratio, while for a 2-bladed propeller it would be that corresponding to three-halves of the total projected area ratio.

In laying down the projected area form for any desired projected area ratio it must be borne in mind that all projected area ratios are outside of the $0.2R$ circle, and any variation from this in the actual blade due to variation of hub diameter may be neglected, as may also any slight widening of the blade inside of the $0.7R$ circle which may be made in order to reduce the thickness of the blade near the hub by increasing the width.

The lengths of the chords of the half arcs for any diameter propeller, D = diameter, will equal length of chord as given in the Table for each fraction of the radius and the particular projected area ratio desired multiplied by the radius of the propeller.

Thus, suppose the diameter of the propeller to be 10 feet and the desired projected area ratio to be 0.6, the length of the chord of the half arc at $0.7R$ will be $0.499 \times 5 = 2.495$ feet.

The greatest circular width of the projected area form of the Standard Form is at the $0.7R$ radius.

Standard Form of Blade Sections

The Standard blade section used is that having the driving face a true helical surface generated by the generatrix of the pitch surface of the propeller, while the back of the propeller is formed by a circular arc, the edges of the blades being made as sharp as possible yet being thickened up sufficiently to give durability.

The approximate ratio of thickness to width of developed circular arc at $0.2R$ from the centre of the hub for blades of manganese bronze of 60,000 pounds tensile strength should be approximately according to Table 16.

Table 16. Table of Blade Thicknesses

P.A. + *D.A.* (3 Blades).....	.15	.2	.25	.3	.35	.4
Thickness + Width.....	.19	.2	.197	.193	.187	.18
P.A. + *D.A.* (3 Blades).....	.45	.5	.55	.6	.65	
Thickness + Width.....	.171	.16	.15	.14	.127	

Table 17. Projected Area and Corresponding Developed Area Ratios for Different Values of Pitch Divided by Diameter

P + *D*	0.8	1.0	1.2	1.4	1.6	1.8	2.0
*P.A.** / *D.A.*	Values of *H.A.* + *D.A.*						
.1	.11	.117	.125	.132	.140	.148	.156
.15	.165	.175	.186	.197	.209	.222	.235
.2	.221	.235	.250	.266	.282	.298	.315
.25	.277	.296	.315	.334	.353	.372	.390
.3	.334	.356	.378	.400	.422	.444	.465
.35	.390	.415	.440	.465	.490	.515	.540
.4	.445	.473	.500	.528	.555	.583	.610
.45	.497	.527	.557	.587	.617	.647	.678
.5	.550	.583	.615	.648	.680	.713	.745
.55	.600	.634	.669	.703	.738	.772	.807
.6	.650	.686	.722	.758	.794	.830	.866
.65	.700	.738	.775	.813	.850	.888	.927
.7	.75	.79	.83	.87	.91	.95	.99

* 3 Blades.

Variations of Projected Area Forms from Standard

Blade forms of projected area may vary in three different manners from the Standard Forms, as follows:

1. Projected area form widening gradually as the radius increases. Circumferential bounding edge circular with corners rounded. Will be known as **Fan-shaped Blades.**

2. Projected area form having greatest half arc chord at greater than 0.7*R* from the centre of the hub. Will be known as **Broad-tipped Blades.**

3. Projected area form having greatest half arc chord at less than 0.7*R* from the centre of the hub. Will be known as **Narrow-tipped Blades.**

The three types are shown in Figs. 1, 2 and 3. In each case the actual propeller can be referred to a Basic propeller having, in Case 1, a projected area outside the 0.2*R* equal to the projected area of the actual propeller outside of this same limiting circle. The Basic propeller of diameter D_1 must be found by assuming various diameters and laying down the projected area forms for those values of projected area ratios which will coincide closely

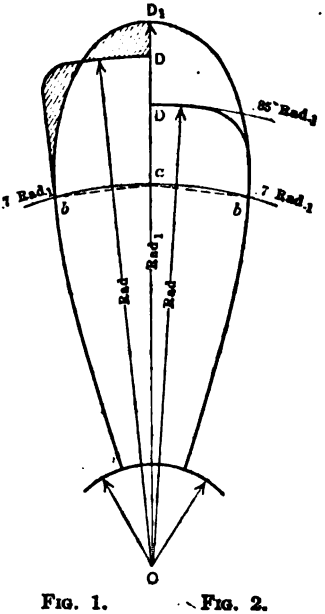


FIG. 1. FIG. 2.

with the actual rectified form of projected area for the greater part of the blade and whose area outside the tip of the actual blade, shown hatched, will equal the hatched area of the actual blade. In Cases 2 and 3, the greatest length of chord of the half arcs through projected area should be ascertained. The actual radius from the centre of the hub of this arc, multiplied by two and divided by 0.7 will give the diameter of the Basic propeller. Dividing the length of the chord by the radius of the Basic propeller will give the standard length of chord at $0.7R$ for unity radius, and the corresponding value of

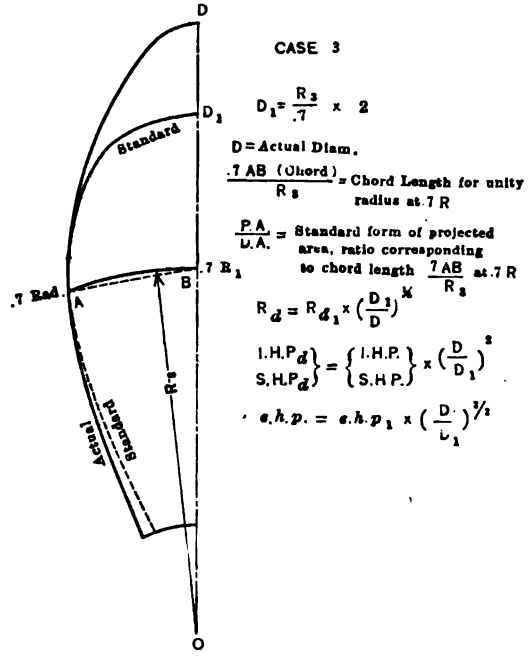


FIG. 3.

P.A. + *D.A.*, for the Basic 3-bladed propeller can then be picked out or interpellated from Table 15. Slight departures of the standard form from the actual form between the $0.7R$ and the hub, as shown by Fig. 3, may be neglected.

Analysis of Non-standard Projected Area Form Propellers, Fan-shaped Blades

Let *e.h.p.* = Effective (tow-rope) horsepower of the vessel.
v = Speed of vessel corresponding to *e.h.p.*
S.H.P.$_d$ or *I.H.P.*$_d$ = Power required to deliver *e.h.p.* with the Basic Propeller.

$S.H.P._d$ or $I.H.P._d$ = Actual power required by Actual Propeller to deliver $e.h.p.$

R_{d1} = Revolutions of Basic Propeller corresponding to $I.H.P._{d1}$ or $S.H.P._{d1}$, $e.h.p.$ and v.

R_d = Revolutions of Actual Propeller corresponding to $I.H.P._d$ or $S.H.P._d$, $e.h.p.$ and v.

D_1 = Diameter of Basic Propeller.

D = Diameter of Actual Propeller.

Then
$$s_1 = S \times \frac{(I.H.P._{d1} \text{ or } S.H.P._{d1}) \times Av}{(I.H.P. \text{ or } S.H.P.) \times A_v}.$$

$$R_{d1} = \frac{v \times 101.33}{P \times (1 - s_1)}.$$

$$R_d = R_{d1} \times \left(\frac{D_1}{D}\right)^{1/4}.$$

$$(I.H.P._d \text{ or } S.H.P._d) = (I.H.P._{d1} \text{ or } S.H.P._{d1}) \times \frac{D_1}{D}.$$

and the propulsive coefficients will vary as the diameters.

Broad-tipped and Narrow-tipped Blades

Let the diameter of the circular arc of greatest projected area length = D_0.

D = Diameter of Actual Propeller.

D_1 = Diameter of Basic Propeller = $D_0 + 0.7$.

$e.h.p._1$ = Effective horsepower delivered by Basic Propeller.

$e.h.p.$ = Actual effective to be delivered by Actual Propeller.

$S.H.P._{d1}$ or $I.H.P._{d1}$ = Power of Basic Propeller to deliver $e.h.p._1$, at speed v.

$S.H.P._d$ or $I.H.P._d$ = Actual Power of Actual Propeller to deliver $e.h.p.$ at speed v.

R_{d1} = Revolutions of Basic Propeller with $S.H.P._{d1}$ or $I.H.P._{d1}$ at speed v.

R_d = Revolutions of Actual Propeller with $S.H.P._d$ or $I.H.P._d$ and v.

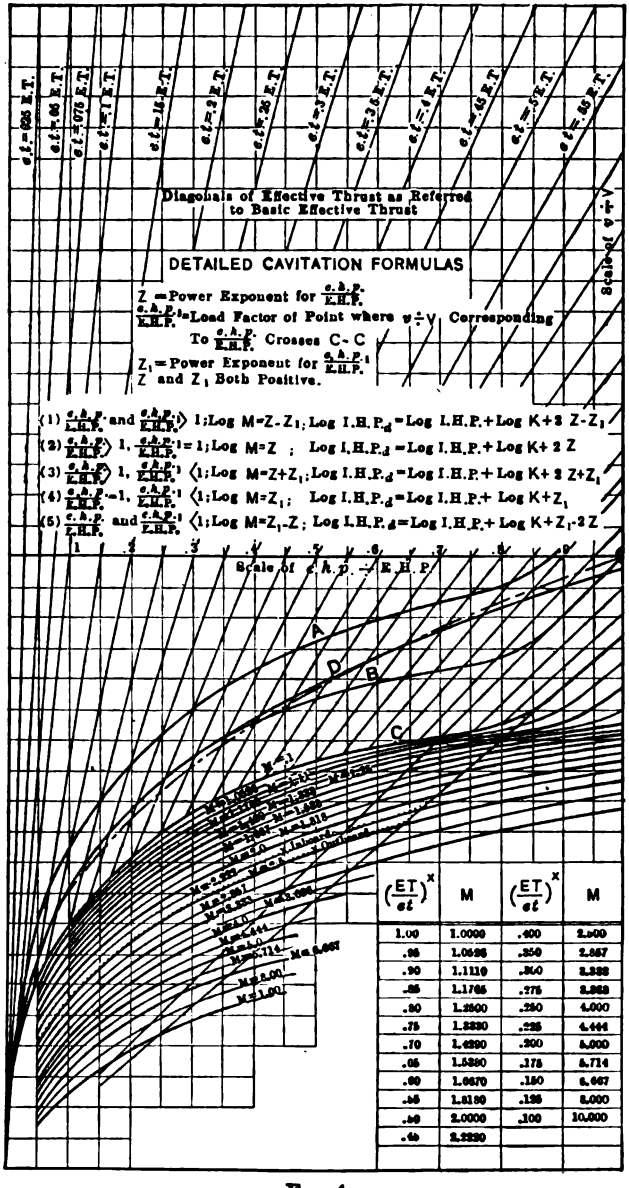
$S.H.P._d$ or $I.H.P._d$ = $(S.H.P._{d1}$ or $I.H.P._{d1}) \times \left(\frac{D}{D_1}\right)^2$. This factor varies apparently from $\left(\frac{D}{D_1}\right)^2$ to $\left(\frac{D}{D_1}\right)^{3/4}$, the latter giving the greater values of $I.H.P._d$ or $S.H.P._d$, and therefore giving a greater lee-way for safety in the estimate.

$$e.h.p. = e.h.p._1 \times \left(\frac{D}{D_1}\right)^{3/4}.$$

$$s_1 = S \frac{(I.H.P._{d1} \text{ or } S.H.P._{d1} \times Av}{(I.H.P. \text{ or } S.H.P.) \times A_v}.$$

$$R_{d1} = \frac{101.33\ v}{P \times (1 - s_1)}.$$

$$R_d = R_{d1} \times \left(\frac{D_1}{D}\right)^{1/4}.$$


e.t.=.025 E.T.
e.t.=.06 E.T.
e.t.=.075 E.T.
e.t.=.1 E.T.
e.t.=.14 E.T.
e.t.=.2 E.T.
e.t.=.26 E.T.
e.t.=.3 E.T.
e.t.=.36 E.T.
e.t.=.45 E.T.
e.t.=.5 E.T.
e.t.=.65 E.T.
Diagonals of Effective Thrust as Referred to Basic Effective Thrust
DETAILED CAVITATION FORMULAS
Z = Power Exponent for e.A.P./E.H.P.
e.A.P./E.H.P.1 = Load Factor of Point where v ÷ V1 Corresponding To e.A.P./E.H.P. Crosses C-C
Z1 = Power Exponent for e.A.P.1/E.H.P.1
Z and Z1 Both Positive.
(1) e.A.P./E.H.P. and e.A.P.1/E.H.P.1 > 1; Log M=Z-Z1; Log I.H.P.d=Log I.H.P.+Log K+2 Z-Z1
(2) e.A.P./E.H.P. 1, e.A.P.1/E.H.P.1 = 1; Log M=Z ; Log I.H.P.d=Log I.H.P.+Log K+2 Z
(3) e.A.P./E.H.P. 1, e.A.P.1/E.H.P.1 < 1; Log M=Z+Z1; Log I.H.P.d=Log I.H.P.+Log K+2 Z+Z1
(4) e.A.P./E.H.P.=1, e.A.P.1/E.H.P.1 < 1; Log M=Z1; Log I.H.P.d=Log I.H.P.+Log K+Z1
(5) e.A.P./E.H.P. and e.A.P.1/E.H.P.1 < 1; Log M=Z1-Z; Log I.H.P.d=Log I.H.P.+Log K+Z1-2 Z
Scale of e.A.P. ÷ E.H.P.
A
D
B
C
Inboard
Outboard

$\left(\frac{ET}{et}\right)^x$	M	$\left(\frac{ET}{et}\right)^x$	M
1.00	1.0000	.400	2.500
.95	1.0526	.350	2.857
.90	1.1110	.300	3.333
.85	1.1765	.275	3.636
.80	1.2500	.250	4.000
.75	1.3330	.225	4.444
.70	1.4290	.200	5.000
.65	1.5380	.175	5.714
.60	1.6670	.150	6.667
.55	1.8180	.125	8.000
.50	2.0000	.100	10.000
.45	2.2220		

FIG. 4a.

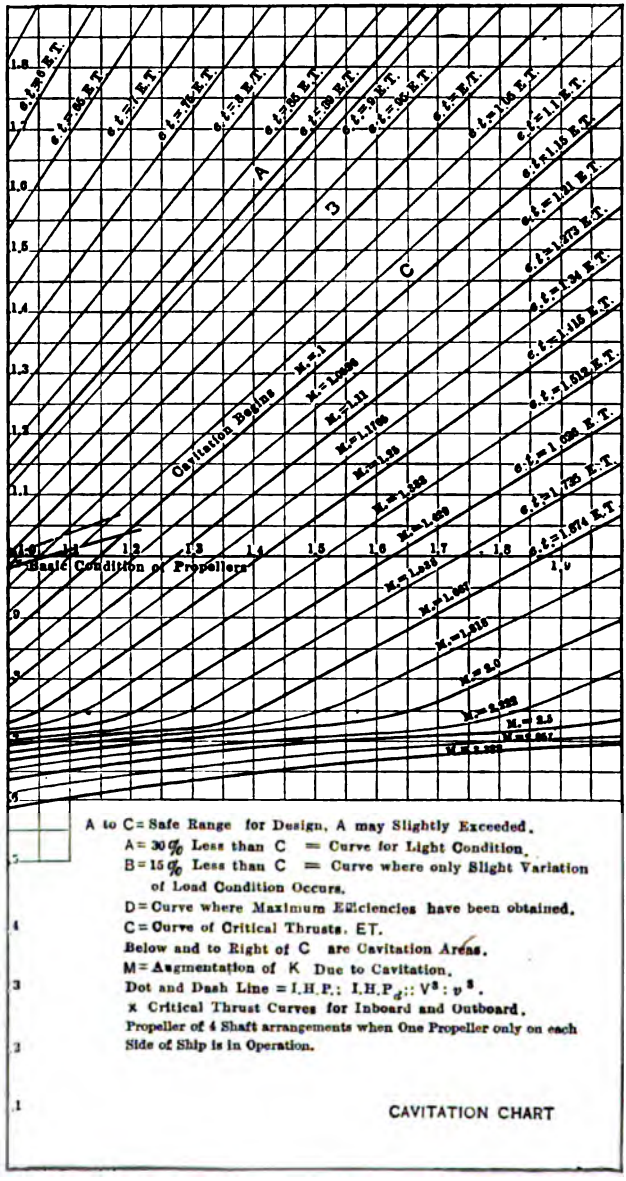

Basic Condition of Propellers
Cavitation Begins
A
B
C
A to C = Safe Range for Design, A may Slightly Exceeded.
A = 30% Less than C = Curve for Light Condition.
B = 15% Less than C = Curve where only Slight Variation of Load Condition Occurs.
D = Curve where Maximum Efficiencies have been obtained.
C = Curve of Critical Thrusts. ET.
Below and to Right of C are Cavitation Areas.
M = Augmentation of K Due to Cavitation.
Dot and Dash Line = I.H.P.: I.H.P.$_d$:: V^3 : v^3.
x Critical Thrust Curves for Inboard and Outboard.
Propeller of 4 Shaft arrangements when One Propeller only on each Side of Ship is in Operation.
CAVITATION CHART

FIG. 4b.

Should the Basic Results plot in the cavitating area of Fig. 4, the corrections for Power or for effective power delivered must be applied to the Basic Propeller estimates and the values for the actual propeller then be obtained from these corrected Basic values.

Facts Concerning the Production of Cavitation

(1) To lighten the load on a vessel with a given propeller delays cavitation and *vice versa*.

(2) To increase the projected area ratio of a propeller, pitch and diameter remaining constant, produces a delay in cavitation and in dispersal of the thrust column and *vice versa*.

(3) To increase the pitch of a given propeller tends to expedite cavitation and *vice versa*.

(4) To decrease the diameter of a given propeller, the pitch remaining constant, tends to produce earlier cavitation and dispersal of the thrust column.

(5) The existence of thrust deduction, in addition to increasing the power necessary for propulsion, decreases the speed and net effective thrust at which cavitation will occur.

(6) Where the hull of a vessel is of such underwater form as to produce a heavy wake, the speeds at which cavitation and dispersal of the thrust column occur will be higher than if no wake existed, on account of the "wake gain."

(7) Where insufficient tip clearance exists between the propeller and the hull, increases in effective horsepower and speed of vessel are accompanied by a gradual increase in the thrust deduction, which latter increase produces earlier cavitation.

(8) Where sections across propeller blades are of such form as to give a free flow of water past the sections with no tendency to form eddies, but where the sections are abnormally thick as compared with their width, the abnormal thickness produces an actual pitch considerably greater than the normal pitch and cavitation is expedited.

(9) Where blade sections are very thick as compared with their width and where the entering and leaving edges are of such form as to produce eddying of the water flowing past the sections of the higher speeds, the thrust per revolution will be increased due to the higher actual pitch produced by the thickness of the sections, and the power required per revolution will increase in greater proportion than the effective thrusts. As the blade speeds increase eddies begin to form, and this formation of eddies is accompanied by a still further exaggeration of power and all of the phenomena of cavitation, and this will occur at lower thrusts and speeds than would be the case were the sections normally fine.

DESIGN OF THE PROPELLER

Limitations of Load Factors *imposed by the S.B.C. of the vessel and by the speed*, as applied to all conditions of design are given in Table 18.

These limit values are only approximate and may be exceeded by about 25 per cent. The speed used should be the speed expected under conditions of maximum hull resistance. The propeller obtained will be the approximate smallest and most efficient propeller suited to meet these conditions.

Table 18. Limiting Maximum Values of e.h.p. + E.H.P. for Various Values of S.B.C. and of v

Slip block coefficient													
0.3		0.4		0.5		0.6		0.7		0.8		0.9	
Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$	Speed	$\frac{e.h.p.}{E.H.P.}$
12	0.015	6	0.009	0	.....	0	.....	0	.....	0	.....	0	.....
14	0.022	8	0.015	2	0.006	2	0.007	1	0.003	1	0.005	1	0.005
16	0.034	10	0.022	4	0.010	4	0.010	2	0.007	2	0.011	2	0.011
18	0.053	12	0.038	6	0.013	6	0.025	3	0.010	3	0.020	3	0.020
20	0.083	14	0.063	8	0.022	8	0.055	4	0.018	4	0.030	4	0.044
22	0.120	16	0.100	10	0.045	10	0.110	5	0.030	5	0.050	5	0.085
24	0.175	18	0.155	12	0.090	12	0.205	6	0.045	6	0.085	6	0.140
26	0.240	20	0.232	14	0.163	14	0.345	7	0.070	7	0.125	7	0.213
28	0.340	22	0.335	16	0.260	16	0.530	8	0.105	8	0.190	8	0.325
30	0.500	24	0.485	18	0.395	18	0.765	9	0.150	9	0.270	9	0.480
32	0.690	26	0.685	20	0.585	20	1.050	10	0.217	10	0.390	10	0.690
34	0.900	28	0.910	22	0.815	22	1.360	11	0.295	11	0.530	11	0.900
36	1.110	30	1.160	24	1.070	24	1.700	12	0.394	12	0.690	12	1.150
38	1.350	32	1.410	26	1.340	.....	.....	13	0.505	13	0.865	13	1.430
40	1.600	34	1.700	28	1.630	.....	.....	14	0.635	14	1.060	14	1.700

Limiting Values of v + V. These for any load factor *e.h.p. + E.H.P.*, should never be less than those corresponding to the *Critical Thrusts, E.T.* (Fig. 4) for this load factor. In cases where it may be absolutely necessary to use lower values of $v + V$, the augment of thrust deduction (M) must be brought into the computations as explained under the general subject of "Cavitation."

Problems in propeller Design

These may be classed under two general heads, namely: (1) **Problems of sufficient data** and (2) **Problems of insufficient data.**

Under the first head are grouped all problems in which full hull data and curve of *e.h.p.*–Speed are given. Designed power of engines and revolutions desired at this power may also be supplied or it may be desired to obtain an estimate of power with the best revolutions for any given speed. Such problems are those which a shipbuilder should recommend to his customer if he has the best interests of the customer at heart.

Under the second head are grouped all problems in which hull data, more or less full, is generally supplied, but in which no curve of *e.h.p.*–Speed is available. The designed power of the engines and their revolutions at this power are given together with the speed which it is expected to obtain with this power. Problems under this head are very unsatisfactory, as should the estimate of speed be wrong the revolutions actually obtained may vary widely from those desired, increasing should the speed be greater than the estimated, and decreasing should it be less, while should the actual speed be much less than the estimate, the danger zone below the curve of *Critical Thrusts*, Fig. 4, may be entered and a corresponding loss in propulsive efficiency be encountered.

Forms for Design Computations (Standard Blades)

Speed, effective horsepower, limiting maximum diameter given to find Projected Area Ratio, indicated or shaft horsepower, diameter, pitch,

revolutions and propulsive efficiency; percentage variation of loading of vessel small. This condition applies to Naval and Passenger Vessels.

(1) D Max. (2); (3) (4) decreasing values.

(2) $e.h.p.$ + $E.H.P.$ Max. allowed for $S.B.C.$ and v (Table 18).

(3) $e.h.p.$ Constant.

(4) $E.H.P. = e.h.p. + \frac{e.h.p.}{E.H.P.}$ Constant.

(5) v (knots) Constant designed speed.

(6) $v + V$ Constant for value of $\frac{e.h.p.}{E.H.P.}$ on Curve D, Fig. 4.

(7) $V = v \div \frac{v}{V}$ Constant.

(8) $E.T._p \times \frac{P.A.}{D.A.} = \frac{C \times E.H.P.}{D^2 \times V}$.

$$C = \begin{cases} 2.491 \text{ for 4-bladed screws.} \\ 2.88 \text{ for 3-bladed screws.} \\ 3.84 \text{ for 2-bladed screws.} \end{cases}$$

(9) $P.A. + D.A.$ (Basic and 3 blades, Total Proj. Area Ratio).

(10) $\begin{cases} \frac{4}{3} P.A. + D.A. \text{ (4 blades, Total Proj. Area Ratio).} \\ \frac{2}{3} P.A. + D.A. \text{ (2 blades, Total Proj. Area Ratio).} \end{cases}$

(11) $P.C.$ for $\left\{ \begin{array}{l} \text{(9) 3 blades} \\ \text{(10) 4 and 2 blades} \end{array} \right\}$ (Table 8).

(12) $I.H.P. = E.H.P. + P.C.$

(13) $S.H.P. = I.H.P. \times 0.92$.

(14) $T.S.$ for (9) (Table 8).

(15) $S.B.C.$ from equation for $S.B.C.$ = Constant.

(16) $1 - S$ for (9) and (15) (Table 8).

(17) $P = (\pi D \times V \times 101.33) \div \{T.S. \times (1 - S)\}$.

Estimate of Power

(18) Z for $\frac{e.h.p.}{E.H.P.}$ Table 12.

(19) $I.H.P._p = I.H.P. \div 10^z$.

(20) K for $S.B.C.$

(21) $I.H.P._d = I.H.P._p \times K$.

When $v + V$ for $\frac{e.h.p.}{E.H.P.}$ falls below or to the left of the Curve of Critical Thrusts (Fig. 4):

(22) M for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (Fig. 4) (Log $M = Z_1 - Z$ where Z_1 is value for $\frac{e.h.p.}{E.H.P.}$ where $\frac{v}{V}$ crosses $E.T.$).

(23) $I.H.P._d = I.H.P._p \times K \times M$.

Estimate of Revolutions

(24) Log Av for V (Table 13).

(25) Log A_s for v (Table 13 or 14).

(26) $s = S \frac{I.H.P._d \times Av}{I.H.P. \times A_s}$.

(27) $R_d = \frac{v \times 101.33}{P \times (1 - s)}$.

Should the revolutions obtained with the maximum diameter be higher than those considered desirable for the type of propelling engines to be fitted, the calculations should be continued, using the maximum diameter with decreasing values of $v \div V$ and $e.h.p. \div E.H.P.$, but retaining the value of $e.t. \div E.T.$ corresponding to the plotted point of the first calculation. With maximum diameter plotted on Fig. 4, at $e.h.p. \div E.H.P. = 0.3$, $v \div V = 0.67$, the value of $e.t. \div E.T.$ as shown by the diagonal line through this point is 0.45. In the new calculations this value is kept constant and points for the calculation are taken from Fig. 4. Should the curve of Critical Thrusts be reached and the revolutions still be too high, then follow along this curve with decreasing values of $e.h.p. \div E.H.P.$ for further calculations, but the change in revolutions along this curve will be small. Should the propeller finally obtained fall on the curve of Critical Thrusts, it will be in a precarious position, as any increase in resistance for a given speed will produce a corresponding decrease in efficiency of propulsion.

Indicated or Shaft Horsepower of Engines and designed revolutions given, also (a) effective horsepower curve (sufficient data), or (b) estimated speed (insufficient data), together with hull characteristics, location of propellers and maximum diameter of propeller that can be carried.

Let it be supposed in this case that the revolutions are low and the power comparatively high, which will necessitate the use of the maximum diameter of propeller possible. Further, let it be supposed that the vessel is one subject to heavy variations in load, as in a collier or freight steamer. Call the speed estimated for the light condition v_1 and that for the loaded condition v_2. Should the propeller be designed for v_1 the revolutions would be seriously lowered when speed is reduced by loading to v_2, also, in the case of reduction gear ships, the tooth pressures in the gears would be unduly increased. Should the propeller be designed for v_2, the revolutions will unduly increase in the light condition and the engines or turbines be dangerously overspeeded. It would appear to be preferable, therefore, to design the propeller for safety at the v_2 condition, and for designed revolutions at a speed $v = \frac{v_1 + v_2}{2}$.

Form for Calculation for Projective Area Ratio, Propulsive Coefficient and Effective Horsepower

(1) D = Max. Diameter of Propeller that can be carried.
(2) v_2 = Est. Min. Speed under maximum load condition, with designed power.

(3) $\frac{e.h.p.}{E.H.P.}$ = Safe load at v_2 for *S.B.C.* (Table 18).

(4) $I.H.P._d = (S.H.P._d \div 0.92)$ = Designed Power.
(5) K = Thrust Deduction Factor.
(6) $I.H.P._p = I.H.P._d \div K$ = Propulsive Horsepower.
(7) Z = Power Factor for $e.h.p. \div E.H.P.$ (Table 12).
(8) $I.H.P. = I.H.P._p \times 10^x$ = Basic Power.

(9) $v_2 \div V$ = That value corresponding to $\frac{e.h.p.}{E.H.P.}$ on Curve of Critical Thrusts (Fig. 4).

(10) $V = v_1 + (v_2 + V)$ = Basic Speed.

(11) $I.T._d + (1 - S) = \dfrac{C \times I.H.P.}{D^2 \times V}$.

Where $C = \begin{cases} 3.84 & \text{for 2-bladed screws.} \\ 2.88 & \text{for 3-bladed screws.} \\ 2.491 & \text{for 4-bladed screws.} \end{cases}$

(12) $P.A. + D.A.$ = Proj. Area Ratio for 3 blades and for Basic (Table 9).

(13) $\frac{2}{3}\,P.A. + D.A.$ = Proj. Area Ratio for 2 blades (Table 9).

(14) $\frac{4}{3}\,P.A. + D.A.$ = Proj. Area Ratio for 4 blades (Table 9).

(15) $P.C.$ = Basic Propulsive Coefficient, taken from Table 8 for (12), (13) or (14), depending upon number of blades.

(16) $E.H.P. = I.H.P. \times P.C.$ = Basic Effective Horsepower.

(17) $e.h.p. = E.H.P. \times \dfrac{e.h.p.}{E.H.P.}$ = Effective horsepower delivered by $I.H.P._d$ or $S.H.P._d$.

(18) v_1 = Speed for *e.h.p.* from effective horsepower curve, or estimated, for light condition.

(19) v_2 = Speed for *e.h.p.* from effective horsepower curve, or estimated, for heavy condition.

(20) $v = \dfrac{v_1 + v_2}{2}$ = Mean speed to use in estimating revolutions.

Calculations for Pitch

(21) $T.S.$ = Tip speed for (12) (Table 8).

(22) $1 - S = 1 -$ Basic Apparent Slip for $S.B.C.$ and (12) (Table 8).

(23) $P = \dfrac{\pi D \times V \times 101.33}{T.S. \times (1 - S)}$ = Pitch of Propeller in feet.

Calculations for Revolutions

(24) Log A_V for V (Table 13).

(25) Log A_v for v (Table 13 or 14, depending upon type of vessels).

(26) $s = S \dfrac{I.H.P._d}{I.H.P.} \times \dfrac{A_V}{A_v}$ = Apparent slip at speed v.

(27) $R_d = \dfrac{v \times 101.33}{P \times (1 - s)}$ = Revolutions for v.

Should the revolutions obtained be too great, at any value of *e.h.p.* + *E.H.P.* on the Critical Thrust Curve of Fig. 4, it will be necessary to resort to one of the methods of *Reduced Diameter*, namely to design either for *Fan-shaped* blades or for *Broad-tipped* blades, using the first where the reduction in revolutions is considerable, and the second where this reduction is slight.

The forms for such computations are shown on pages 1060 and 1061.

	Fan-shaped blades		Broad-tipped blades	
Given	$I.H.P._d$ R_d & v	$e.h.p.$, R_d & v	$I.H.P._d$ R_d & v	$e.h.p.$, R_d & v
(1) D = Diam. Max.	Constant	Constant	Constant	Constant
(2) D_1 = Diam. $> D$	1-2-3	1-2-3	1-2-3	1-2-3
(3) $D_1 + D$	1-2-3	1-2-3	1-2-3	1-2-3
(4) $(D_1 + D)^2$			1-2-3	
(5) $(D_1 + D)^{\frac{1}{2}}$	1-2-3	1-2-3		
(6) $(D_1 + D)^{\frac{3}{2}}$				1-2-3
(7) $(D_1 + D)^{\frac{1}{4}}$			1-2-3	1-2-3
(8) $I.H.P._d$	1-2-3		1-2-3	
(9) $I.H.P._{d1}$	$I.H.P._d + \frac{D_1}{D}$		$I.H.P._d \times \left(\frac{D_1}{D}\right)^2$	
(10) *K = Thrust deduction factor	1-2-3	1-2-3	1-2-3	1-2-3
(11) $I.H.P._{p1} = I.H.P._{d1} + K$	1-2-3		1-2-3	
(12) $e.h.p._1 + E.H.P._1$ (as safe for *S.B.C.* and v (Table 18).	1-2-3	1-2-3	1-2-3	
(13) Z for $e.h.p._1 + E.H.P.$	1-2-3	1-2-3	1-2-3	1-2-3
(14) $I.H.P._1 = I.H.P._{p1} \times 10^z$	1-2-3		1-2-3	
(15) $e.h.p.$ for v		1-2-3		1-2-3
(16) $e.h.p._1$		$e.h.p.$		$e.h.p. \times \left(\frac{D_1}{D}\right)^{\frac{3}{2}}$
(17) $E.H.P._1 = e.h.p._1 + \frac{e.h.p._1}{E.H.P._1}$		1-2-3		1-2-3
(18) v = Des. speed	Constant	Constant	Constant	Constant
(19) $v + V$ (Fig. 4)	Constant	Constant	Constant	Constant
(20) $V = v + \left(\frac{v}{V}\right)$	Constant	Constant	Constant	Constant
(21) $I.T._D \div (1 - S)$	$\frac{C \times I.H.P._1}{D_1^2 \times V}$		$\frac{C \times I.H.P._1}{D_1^2 \times V}$	
(22) $E.T._p \times \frac{P.A.}{D.A.}$		$\frac{C \times E.H.P._1}{D_1^2 \times V}$		$\frac{C \times E.H.P._1}{D_1^2 \times V}$
C =	3.84 for two-bladed screws.			
	2.88 for three-bladed screws.			
	2.491 for four-bladed screws.			
(24) $P.A. + D.A.$ for (21) (Table 9)	1-2-3		1-2-3	
(25) $P.A. + D.A.$ for (22) (Table 10)		1-2-3		1-2-3
(26) $\frac{1}{3}$ $P.A. + D.A.$ for two-bladed screws	1-2-3	1-2-3	1-2-3	1-2-3
(27) $\frac{1}{4}$ $P.A. + D.A.$ for four-bladed screws	1-2-3	1-2-3	1-2-3	1-2-3
(28) $P.C.$ for (24) or (25) three-bladed. (26) two-bladed screws... (27) four-bladed screws...	Table 8 for all columns.			

	Fan-shaped blades		Broad-tipped blades	
(29) $E.H.P._1 = I.H.P._1 \times$ (28)	1–2–3		1–2–3	
(30) $e.h.p._1 = E.H.P._1 \times \frac{e.h.p._1}{E.H.P._1}$	1–2–3		1–2–3	
(31) $I.H.P._1 =$ (29) + (17)		1–2–3		1–2–3
(32) $I.H.P._{p1} =$ (31) + $10^{g(13)}$		1–2–3		1–2–3
(33) $I.H.P._{d1} =$ (32) $\times$ (10)		1–2–3		1–2–3
(34) $e.h.p.$	$e.h.p._1$		$e.h.p._1 \times \left(\frac{D}{D_1}\right)^{3/2}$	
(35) $I.H.P._d$		$I.H.P._{d1} \times \left(\frac{D_1}{D}\right)$		$I.H.P._{d1} \div \left(\frac{D_1}{D}\right)^2$
Calculation for Pitch				
(36) $T.S.$ for (24) or (25)	1–2–3	1–2–3	1–2–3	1–2–3
(37) $1 - S$ for (24) or (25) and $S.B.C.$	1–2–3	1–2–3	1–2–3	1–2–3
(38) $P = \frac{\pi \times D_1 \times V \times 101.33}{T.S. \times (1 - S)}$	1–2–3	1–2–3	1–2–3	1–2–3
Calculations for Revolutions				
(39) Log Av for (20) (Table 13)	1–2–3	1–2–3	1–2–3	1–2–3
(40) Log A_v for (18) (Table 13 or 14)	1–2–3	1–2–3	1–2–3	1–2–3
(41) $s_1 = S \times \frac{I.H.P._{d1} \times A_v}{I.H.P._1 \times Av}$	1–2–3	1–2–3	1–2–3	1–2–3
(42) $R_{d1} = \frac{v \times 101.33}{P \times (1 - s)}$	1–2–3	1–2–3	1–2–3	1–2–3
(43) R_d	$R_{d1} \times \left(\frac{D_1}{D}\right)^{1/2}$	$R_{d1} \times \left(\frac{D_1}{D}\right)^{1/2}$		
(44) R_d			$R_{d1} \times \left(\frac{D_1}{D}\right)^{1/4}$	$R_{d1} \times \left(\frac{D_1}{D}\right)^{1/4}$

* In propellers of reduced diameter fitted to vessels of such type that the clearance between blade tips and hull has a governing effect on K, the value of K used should be that corresponding to the tip clearance of the Basic propeller or diameter D, and not the actual tip clearance. Reduced Diameter propellers enter and leave cavitation with the Basic propellers from which they are derived. The ratio of D to D_1 may be carried to as low as 0.80 with practical safety.

Materials Used in Construction of Propellers

Blades: 1. *Cast iron:* Advantages: Cheapness, weakness which protects shafting by blade breaking upon striking obstruction.

Disadvantages: Heavy corrosion in salt water, heavy blade sections, blunt edges of blades.

2. *Semi-steel:* Advantages: Stronger than cast iron, cheapness.

Disadvantages: Liability to carry hidden flaws rendering it unreliable, corrosion in sea water.

3. *Cast-steel:* Advantages: Stronger than 1 or 2.

Disadvantages: Liability of hidden flaws, roughness of castings, corrosion in sea water.

4. *Forged-steel:* Advantages: Stronger and more reliable than 3.

Disadvantages: Expense of fabrication, corrosion in sea water.

5. *Manganese* and other *Strong Bronzes:* Advantages: Great strength, low ratio of thickness to width, fine sections, thin edges, high polish of surfaces, easily cast, practically free from corrosion.

Disadvantages: High first cost, liability to produce corrosion of adjacent parts of hull unless zinc plate protectors are fitted to the latter.

6. *Monel Metal:* Advantages: Great strength and toughness, light and sharp blade sections, high polish and non-corrosive in sea water.

Disadvantages: High first cost, difficult to machine, warps heavily in cooling in the mold, thus necessitating large amount of waste metal in rough casting in order to insure machining to designed pitch, tendency to porosity around edges of blades, liability to produce corrosion of adjacent parts of the hull unless zinc protector plates are fitted to the latter.

The Hub. Hubs are usually made of cast-iron for semi-steel or cast-iron, semi-steel and cast-steel built-up propellers; of semi-steel for the cheaper classes of work, and of manganese bronze for the better classes, with manganese bronze built-up propellers; of manganese bronze or Monel-metal for Monel-metal blades; where the propeller hub and blades are cast as one, of course the hub is of the same material as the blades.

Thickness of Propeller Blade at Root

T = Thickness of blade, in inches, at and tangent to the hub, additional thickness due to fillets being neglected. See Table 16.
W = Width of blade tangent to hub, in inches.
A = (33,000 × I.H.P.$_d$) ÷ (2π × R × Number of blades) = 5252 $I.H.P._d$ ÷ (R × N).
B = 0.31 × Diameter of Screw in feet = 0.31D.
C = A ÷ B.
D = 12 × B − Radius of hub, in inches.
E = C × D.
F = (33,000 × $I.H.P._d$) ÷ (P × R × N)
G = 0.345 × Diameter of propeller in inches.
H = G − Radius of hub, in inches.
J = F × H.
K = Circumference of hub, in feet ÷ P, in feet.
L = Sine of Arc whose tangent is K.
M = Cosine Arc whose tangent is K.
N = L × J.
O = M × E.

Table 19. Values of f to be Used for Different Design Point Values of e.h.p. + E.H.P.

$\frac{e.h.p.}{E.H.P.}$	Values of f			
	Constant Torque		Reciprocating engines	
	Manganese bronze	Cast steel	Manganese bronze	Cast steel
0.1	4,800	4,300	4,800	4,300
0.2	5,500	4,900	5,400	4,850
0.3	6,300	5,750	6,000	5,500
0.4	7,150	6,650	6,550	6,050
0.5	8,000	7,500	7,150	6,650
0.6	9,000	8,450	7,750	7,250
0.7	9,800	9,300	8,300	7,800
0.8	10,750	10,250	8,900	8,400
0.9	11,600	11,100	9,450	8,950
1.0	12,500	12,000	10,000	9,500

$P = N + O$.

f = fiber stress as per values of *e.h.p.* + *E.H.P.* given in Table 19.

$$T = \sqrt{\frac{13.125\ P}{W \times f}}.$$

Fixing the maximum value of T at $0.2W$, T should never exceed $T = \sqrt[3]{\frac{P \times 2.625}{f}}$ for the strong bronzes.

For cast-iron and semi-steel, f = 2500 to 4000 for values of *e.h.p.* + *E.H.P.* not in excess of 0.4.

Centrifugal Force: Increase of Stress

W = Weight of blade, in pounds.

r = Distance of center of gravity of blade from center of hub.

g = Gravity = 32.

v = Velocity of center of gravity of blade, in feet per second,

$$C = \frac{W}{g} \times v^2, \text{ and the tension on the bolts} = \frac{W}{g} \times \frac{v^2}{r} = \text{centrifugal force,}$$

$\frac{v^2}{r}$ being the acceleration.

When a propeller is in motion, with the ship advancing, running at R revolutions per minute, $v = \frac{R}{60}\sqrt{P^2 + (2\pi r)^2}$.

To determine the bending moment at any distance d, from the center of gravity, due to C, and resisted by the section at the root of the blade $B.M. = C \times d$.

The bending moment in a plane at right angles to the axis of the screw at this distance d is $B.M. = \left(\frac{(2\pi \times r \times 60)}{R}\right)^2 \times \frac{W}{g} \times d$.

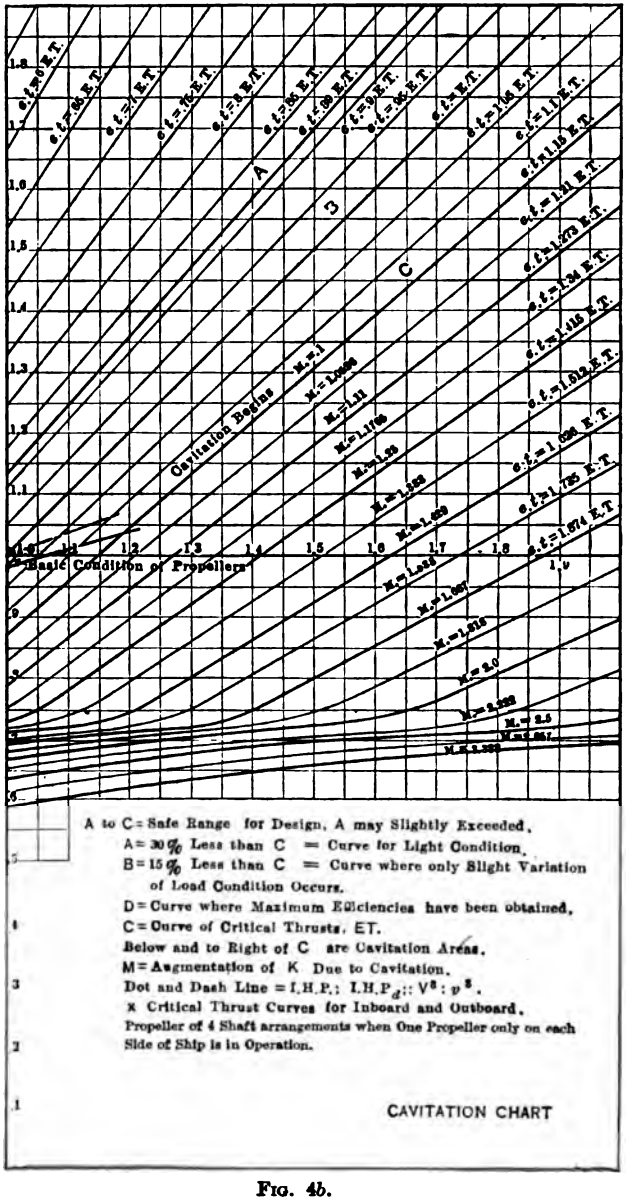

Cavitation Begins
Basic Condition of Propellers
A to C = Safe Range for Design, A may Slightly Exceeded.
A = 30% Less than C = Curve for Light Condition.
B = 15% Less than C = Curve where only Slight Variation of Load Condition Occurs.
D = Curve where Maximum Efficiencies have been obtained.
C = Curve of Critical Thrusts, ET.
Below and to Right of C are Cavitation Areas.
M = Augmentation of K Due to Cavitation.
Dot and Dash Line = I.H.P.; I.H.P.$_d$:: $V^3 : p^3$.
x Critical Thrust Curves for Inboard and Outboard.
Propeller of 4 Shaft arrangements when One Propeller only on each Side of Ship is in Operation.
CAVITATION CHART

FIG. 4b.

Should the Basic Results plot in the cavitating area of Fig. 4, the corrections for Power or for effective power delivered must be applied to the Basic Propeller estimates and the values for the actual propeller then be obtained from these corrected Basic values.

Facts Concerning the Production of Cavitation

(1) To lighten the load on a vessel with a given propeller delays cavitation and *vice versa*.

(2) To increase the projected area ratio of a propeller, pitch and diameter remaining constant, produces a delay in cavitation and in dispersal of the thrust column and *vice versa*.

(3) To increase the pitch of a given propeller tends to expedite cavitation and *vice versa*.

(4) To decrease the diameter of a given propeller, the pitch remaining constant, tends to produce earlier cavitation and dispersal of the thrust column.

(5) The existence of thrust deduction, in addition to increasing the power necessary for propulsion, decreases the speed and net effective thrust at which cavitation will occur.

(6) Where the hull of a vessel is of such underwater form as to produce a heavy wake, the speeds at which cavitation and dispersal of the thrust column occur will be higher than if no wake existed, on account of the "wake gain."

(7) Where insufficient tip clearance exists between the propeller and the hull, increases in effective horsepower and speed of vessel are accompanied by a gradual increase in the thrust deduction, which latter increase produces earlier cavitation.

(8) Where sections across propeller blades are of such form as to give a free flow of water past the sections with no tendency to form eddies, but where the sections are abnormally thick as compared with their width, the abnormal thickness produces an actual pitch considerably greater than the normal pitch and cavitation is expedited.

(9) Where blade sections are very thick as compared with their width and where the entering and leaving edges are of such form as to produce eddying of the water flowing past the sections of the higher speeds, the thrust per revolution will be increased due to the higher actual pitch produced by the thickness of the sections, and the power required per revolution will increase in greater proportion than the effective thrusts. As the blade speeds increase eddies begin to form, and this formation of eddies is accompanied by a still further exaggeration of power and all of the phenomena of cavitation, and this will occur at lower thrusts and speeds than would be the case were the sections normally fine.

DESIGN OF THE PROPELLER

Limitations of Load Factors *imposed by the S.B.C. of the vessel and by the speed*, as applied to all conditions of design are given in Table 18.

These limit values are only approximate and may be exceeded by about 25 per cent. The speed used should be the speed expected under conditions of maximum hull resistance. The propeller obtained will be the approximate smallest and most efficient propeller suited to meet these conditions.

Table 21. Dimensions of Standard Three-blade, Built-up Propeller; see Figs. 6, 7 and 8.

Formula for obtaining d

$$A = \frac{33{,}000 \times \text{S.H.P.} \times G}{N \times P \times \text{R.P.M.} \times n \times r \times f}$$

where A = Net area of one bolt, sq. in.
$G = a - B$.
a = Distance from center of shaft to center of thrust, in in.
= Diameter of propeller in in. × 0.345.
B = Distance from center of shaft to face of blade flange, in in.
N = Number of blades.
P = Pitch in ft.
n = Number of bolts in driving side of one blade.
r = Radius of pitch circle, in in.
f = 4,500 for manganese bronze or naval brass.

Formula for obtaining m $\quad m = \dfrac{126{,}048 \times \text{S.H.P.}}{\text{R.P.M.} \times D \times \text{Length of key} \times f_s}$

Depth of key is obtained from the same formula as m, substituting f_s for f_s.

D	B	C	D_1	D_2	F	$F_1°$	F_2	H	K	L	L_1	N	N_1	N_2
8¼	6⅞	4 13/16	12⅛	15	16 7/16	28	7 1/16	26	10 5/16	20½	10¼	8	9⅜	½
8¼– 9 inc.	7½	5¼	13¼	16 7/16	18 3/16	28	8⅜	28	11¼	21¾	10⅞	8⅜	9¾	½
9 – 9½	8	5⅜	14¼	17⅝	19⅜	28	9	30	12	23	11½	9⅛	10⅜	½
9½–10¼	8 9/16	6	15⅛	18¾	20⅝	28	9 9/16	32	12⅞	24½	12¼	9⅞	11½	½
10¼–10¾	9⅛	6⅜	16⅛	20	21¼	28	10⅛	34	13⅝	26	13	10¼	11⅞	½
10¾–11½	9⅝	6¾	17	21⅛	23 5/16	28	10¾	36	14½	27¾	13⅞	11	12⅝	½
11½–12	10 3/16	7⅛	18	22¼	24⅝	28	11⅜	38	15¼	29¼	14⅝	11⅝	13½	½
12 –12¾	10¾	7½	18 5/16	23½	25¾	28	12	40	16	30¾	15⅜	12	13⅝	½
12¾–13½	11¼	7⅞	19⅞	24⅝	27 5/16	28	12 9/16	42	16⅞	32¼	16⅛	12¾	14¾	½
13½–14	11 13/16	8¼	20 13/16	25 13/16	28 9/16	28	13⅛	44	17⅝	33¾	16⅞	13½	15½	½
14 –14¾	12⅜	8⅝	21¾	26 15/16	29 13/16	28	13¾	46	18½	35½	17¾	14¼	16⅜	½
14¾–15¾	12⅞	9	22¾	28 5/16	31 3/16	28	14⅜	48	19¼	37	18½	14⅝	16¾	½
15¼–15¾	13⅜	9⅜	23⅝	29⅜	32 7/16	28	14 15/16	50	20	38¼	19⅛	15⅜	17½	½
15¾–16½	13 15/16	9¾	24⅝	30½	33 11/16	28	15 9/16	52	20⅞	39¾	19⅞	15¾	18	½
16½–17¼	14½	10⅛	25½	31 13/16	35 1/16	28	16⅛	54	21⅝	41⅛	20¾	16⅜	18¾	½
17¼–18	15	10½	26½	32⅞	36 5/16	28	16¾	56	22½	43¼	21⅝	17⅛	19⅝	½
18 –18½	15 9/16	10⅞	27 7/16	34	37 9/16	28	17 5/16	58	23¼	44¾	22⅜	17⅝	20⅛	½
18½–19¼	16 1/16	11¼	28⅜	35 1/16	38⅞	28	17 13/16	60	24	46¼	23⅛	18⅜	21¼	½
19¼–19¾	16⅝	11⅝	29⅜	36 3/16	40 1/16	28	18½	62	24⅞	47¾	23⅞	19	21⅝	½
19¾–20½	17⅛	12	30⅛	37 9/16	41 7/16	28	19⅛	64	25⅝	49¼	24⅝	19¾	22½	½
20½–21	17 11/16	12⅜	31¼	38 11/16	42¾	28	19 11/16	66	26½	50¾	25⅜	20½	23¼	½
21 –21¾	18¼	12¾	32 3/16	39⅞	44⅛	28	20 5/16	68	27¼	52¼	26⅛	20⅞	23⅝	½
21¾–22½	18¾	13⅛	33⅛	41 1/16	45⅜	28	20 13/16	70	28	53¾	26⅞	21⅜	24⅞	½

Table 21.—(*Continued.*)

D	N_3	N_4	PD	PD_1	PD_2	R	R_1	R_2	S	T	T_1	T_2	T_3	W	W_1
8¼	4⅜	3 13/16	13¾	13½	Made to suit dia. D and thickness l	4¾	4 9/16	8 15/16	5⅛	1 15/16	2 3/16	2¾	2½	3⅜	1¾
8¼– 9 inc.	5	4	15	14¾		5 3/16	5	9¾	5¾	2⅛	2⅜	2⅝	2 13/16	3 11/16	1⅞
9 – 9½	5⅜	4 5/16	16	15¾		5 9/16	5⅜	10 7/16	6¼	2¼	2 9/16	2 13/16	2⅞	3 15/16	2
9½–10¼	5¾	4 11/16	17⅛	16 13/16		5 13/16	5 11/16	11⅛	6¾	2⅜	2 11/16	3	3 1/16	4 5/16	2⅛
10¼–10¾	6⅛	4⅞	18¼	17⅞		6 3/16	6 1/16	11 13/16	7	2 9/16	2⅞	3 5/16	3¼	4½	2¼
10¾–11½	6⅜	5 3/16	19¼	18 15/16		6 5/8	6 7/16	12½	7½	2 13/16	3 1/16	3⅜	3½	4¾	2 7/16
11½–12	6¾	5 9/16	20⅜	20		7	6¾	13¼	8	2 13/16	3¼	3 9/16	5	5	2 9/16
12 –12⅜	7	5 13/16	21 7/16	21 1/16		7⅜	7⅛	13 15/16	8¼	3	3⅜	3¾	3⅞	5¼	2 11/16
12¾–13½	7½	6 1/16	22½	22⅛		7¾	7½	14⅝	8¾	3¼	3 9/16	3 15/16	4 1/16	5½	2 13/16
13½–14	7⅞	6⅜	23 9/16	23⅛		8⅛	7⅞	15 5/16	9¼	3 5/16	3¾	4⅛	4¼	5 13/16	2 15/16
14 –14¾	8¼	6¾	24½	24 1/16		8½	8 1/16	16	9¾	3 7/16	3⅞	4 5/16	4 7/16	6 1/16	3 1/16
14¾–15¼	8½	6 15/16	25 1/16	25½		8⅞	8 9/16	16⅜	10	3 9/16	4 1/16	4½	4⅝	6 5/16	3¼
15¼–15¾	8¾	7¼	26¾	26 5/16		9¼	8 15/16	17 7/16	10½	3¾	4¼	4 11/16	4 13/16	6 9/16	3⅜
15¾–16½	9⅛	7 7/16	27⅞	27⅜		9⅝	9¼	18⅛	10¾	3⅞	4 7/16	4⅞	5	6 13/16	3½
16½–17¼	9½	7 13/16	28 15/16	28 7/16		10	9⅝	18 13/16	11¼	4	4 9/16	5 1/16	5 3/16	7⅛	3⅝
17¼–18	10	8⅛	30	29½		10⅜	10	19½	11¾	4 1/16	4¾	5¼	5⅜	7⅜	3¾
18 –18½	10⅜	8½	31 1/16	30½		10¾	10⅜	20 1/16	12¼	4 5/16	4 15/16	5 7/16	5 9/16	7⅝	3⅞
18½–19¼	10¾	8 13/16	32⅛	31 9/16		11 1/16	10 13/16	20⅞	12¾	4½	5 1/16	5⅝	5 13/16	7⅞	4
19¼–19¾	11⅛	9	33¼	32⅝		11 7/16	11 1/16	21 9/16	13	4⅝	5¼	5 13/16	6	8 1/16	4⅛
19¾–20½	11½	9⅜	34¼	33 11/16		11 13/16	11 7/16	22¼	13½	4 13/16	5 7/16	6	6⅛	8 7/16	4¼
20½–21	11⅞	9 11/16	35⅜	34¾		12 3/16	11¾	23	14	4 13/16	5⅝	6 1/16	6⅜	8 11/16	4 7/16
21 –21¾	12¼	9⅞	36 7/16	35 13/16		12 9/16	12⅛	23 13/16	14¼	5 1/16	5¾	6⅜	6 9/16	8 15/16	4 9/16
21¾–22½	12¾	10⅜	37½	36⅞		12 15/16	12½	24⅜	15	5¼	5 15/16	6 9/16	6¾	9 3/16	4 11/16

Table 21.—(*Continued.*)

D	*d*	d_1	d_2	d_3	d_4	d_5	d_6	d_7	d_8	d_9	*f*	f_1	*h*	h_1	*j*	*k*	k_1
8¼	See formula at head of Table 21	5/8	5/8	5/16	1.714*d*	1.143*d*	0.286*d*	5/16	1/2	1/4	1 3/4	1 3/8	0.357*d*	0.643*d*	1 1/4	1/2	7/32
8¼– 9 inc.		5/8	5/8	5/16	1.714*d*	1.143*d*	0.286*d*	5/16	9/16	9/16	1 1/4	1 7/16	0.357*d*	0.643*d*	1 1/4	1/2	1/4
9 – 9½		5/8	5/8	3/8	1.714*d*	1.143*d*	0.286*d*	3/8	9/16	5/8	1 1/4	1 9/16	0.357*d*	0.643*d*	1 3/8	1/2	1/4
9½–10¼		5/8	5/8	3/8	1.714*d*	1.143*d*	0.286*d*	3/8	5/8	5/8	1 1/4	1 11/16	0.357*d*	0.643*d*	1 3/8	1/2	9/32
10¼–10¾		5/8	5/8	3/8	1.714*d*	1.143*d*	0.286*d*	3/8	5/8	5/8	1 1/4	1 3/4	0.357*d*	0.643*d*	1 1/2	5/8	5/16
10¾–11½		5/8	5/8	7/16	1.714*d*	1.143*d*	0.286*d*	7/16	5/8	5/8	1 1/4	1 7/8	0.357*d*	0.643*d*	1 5/8	5/8	5/16
11½–12		5/8	5/8	7/16	1.714*d*	1.143*d*	0.286*d*	7/16	11/16	3/4	1 1/4	2	0.357*d*	0.643*d*	1 5/8	3/4	11/32
12 –12¾		5/8	5/8	7/16	1.714*d*	1.143*d*	0.286*d*	7/16	11/16	3/4	1 1/4	2 1/8	0.357*d*	0.643*d*	1 3/4	3/4	11/32
12¾–13½		5/8	3/4	1/2	1.714*d*	1.143*d*	0.286*d*	1/2	3/4	7/8	1 1/4	2 3/16	0.357*d*	0.643*d*	1 7/8	7/8	3/8
13½–14		5/8	3/4	1/2	1.714*d*	1.143*d*	0.286*d*	1/2	3/4	7/8	1 1/4	2 5/16	0.357*d*	0.643*d*	2	7/8	3/8
14 –14¾		5/8	3/4	1/2	1.714*d*	1.143*d*	0.286*d*	1/2	13/16	7/8	1 1/4	2 3/8	0.357*d*	0.643*d*	2	1	13/32
14¾–15¼		3/4	3/4	9/16	1.714*d*	1.143*d*	0.286*d*	9/16	13/16	1	1 1/4	2 1/2	0.357*d*	0.643*d*	2 1/4	1	13/32
15¼–15¾		3/4	3/4	9/16	1.714*d*	1.143*d*	0.286*d*	9/16	7/8	1	1 1/4	2 5/8	0.357*d*	0.643*d*	2 1/4	1 1/8	7/16
15¾–16½		3/4	7/8	9/16	1.714*d*	1.143*d*	0.286*d*	9/16	7/8	1	1 1/4	2 3/4	0.357*d*	0.643*d*	2 1/4	1 1/8	15/32
16½–17¼		3/4	7/8	9/16	1.714*d*	1.143*d*	0.286*d*	9/16	15/16	1	1 1/4	2 7/8	0.357*d*	0.643*d*	2 1/4	1 1/8	15/32
17¼–18		3/4	7/8	5/8	1.714*d*	1.143*d*	0.286*d*	5/8	15/16	1	1 1/4	2 15/16	0.357*d*	0.643*d*	2 1/2	1 1/4	1/2
18 –18½		3/4	7/8	5/8	1.714*d*	1.143*d*	0.286*d*	5/8	1	1	1 1/4	3	0.357*d*	0.643*d*	2 1/2	1 1/4	1/2
18½–19¼		3/4	7/8	5/8	1.714*d*	1.143*d*	0.286*d*	5/8	1	1 1/8	1 1/4	3 1/8	0.357*d*	0.643*d*	2 1/2	1 3/8	17/32
19¼–19¾		3/4	7/8	5/8	1.714*d*	1.143*d*	0.286*d*	5/8	1 1/16	1 1/8	1 1/4	3 1/4	0.375*d*	0.643*d*	2 3/4	1 3/8	9/16
19¾–20½		3/4	7/8	3/4	1.714*d*	1.143*d*	0.286*d*	3/4	1 1/16	1 1/8	1 1/4	3 3/8	0.357*d*	0.643*d*	2 3/4	1 1/2	9/16
20½–21		3/4	7/8	3/4	1.714*d*	1.143*d*	0.286*d*	3/4	1 1/8	1 1/8	1 1/4	3 7/16	0.357*d*	0.643*d*	3	1 1/2	19/32
21 –21¾		3/4	1	3/4	1.714*d*	1.143*d*	0.286*d*	3/4	1 1/4	1 1/8	1 1/4	3 9/16	0.357*d*	0.643*d*	3	1 5/8	19/32
21¾–22½		3/4	1	3/4	1.714*d*	1.143*d*	0.286*d*	3/4	1 5/16	1 1/8	1 1/4	3 5/8	0.357*d*	0.643*d*	3	1 3/8	5/8

Table 21.—(*Continued.*)

D	k_2	k_3	k_4	k_5	k_6	k_7	l	m	r	r_1	r_2	l	l_1	l_2
8¼	1 3/16	7/16	1 1/8	7/16	0.857d	0.286d	Shaft diam. D above: 15 " inc. l = 1 7/8" ; 17½" inc. l = 1"; 20 " inc. l = 1½"	See formula at head of Table 21	3 5/16	1 3/16	9/16	1 3/8	7/32	1/8
8¼– 9 inc.	1 5/16	1/2	1 1/4	1/2	0.857d	0.286d			3 3/8	7/8	5/8	1 1/2	1/4	1/8
9 – 9½	1	1/2	1 5/16	1/2	0.857d	0.286d			3 7/8	7/8	5/8	1 5/8	1/4	1/8
9½–10¼	1	9/16	1 3/8	9/16	0.857d	0.286d			4 1/8	1	3/4	1 3/4	9/32	5/32
10¼–10¾	1 1/8	9/16	1 1/2	9/16	0.857d	0.286d			4 3/8	1	3/4	1 7/8	5/16	5/32
10¾–11½	1 1/8	5/8	1 9/16	5/8	0.857d	0.286d			4 5/8	1	3/4	1 15/16	5/16	5/32
11½–12	1 1/4	5/8	1 11/16	5/8	0.857d	0.286d			4 7/8	1 1/8	1 3/16	2 1/16	11/32	3/16
12 –12¾	1 1/4	1 5/16	1 3/4	1 5/16	0.857d	0.286d			5 1/8	1 1/8	1 3/16	2 3/16	1 1/32	3/16
12¾–13½	1 3/8	3/4	1 7/8	3/4	0.857d	0.286d			5 3/8	1 3/16	7/8	2 1/4	3/8	3/16
13½–14	1 7/16	3/4	1 15/16	3/4	0.857d	0.286d			5 5/8	1 3/16	7/8	2 3/8	3/8	3/16
14 –14¾	1 1/2	1 3/16	2	1 3/16	0.857d	0.286d			6	1 5/16	1 3/16	2 1/2	13/32	7/32
14¾–15¼	1 9/16	7/8	2 1/8	7/8	0.857d	0.286d			6 3/16	1 5/16	1 5/16	2 9/16	13/32	7/32
15¼–15¾	1 5/8	7/8	2 1/4	7/8	0.857d	0.286d			6 7/16	1 3/8	1	2 11/16	7/16	7/32
15¾–16½	1 11/16	1 3/16	2 5/16	1 5/16	0.857d	0.286d			6 11/16	1 3/8	1	2 13/16	15/32	7/32
16½–17¼	1 3/4	1 5/16	2 3/8	1 5/16	0.857d	0.286d			6 15/16	1 1/2	1 5/16	2 7/8	15/32	1/4
17¼–18	1 13/16	1	2 5/16	1	0.857d	0.286d			7 1/4	1 1/2	1 5/16	3	1/2	1/4
18 –18½	1 7/8	1	2 9/16	1	0.857d	0.286d			7 1/2	1 5/8	1 5/8	3 1/8	1/2	1/4
18½–19¼	1 15/16	1 5/16	2 5/8	1 5/16	0.857d	0.286d	Shaft diam. D above: 7½"–10 " inc. l = 1½" ; 10 "–12½" inc. l = 1 5/8" ; 12½"–15 " inc. l = 1 7/8"		7 3/4	1 5/8	1 5/8	3 1/4	17/32	1/4
19¼–19¾	2	1 5/16	2 3/4	1 5/16	0.857d	0.286d			8	1 13/16	1 3/4	3 3/8	9/16	9/32
19¾–20½	2	1 3/8	2 13/16	1 3/8	0.857d	0.286d			8 1/4	1 13/16	1 3/4	3 7/16	9/16	9/32
20½–21	2 1/8	1 3/16	2 7/8	1 3/16	0.857d	0.286d			8 1/2	1 13/16	1 5/16	3 9/16	19/32	9/32
21 –21¾	2 3/16	1 5/16	3	1 5/16	0.857d	0.286d			8 3/4	1 13/16	1 5/16	3 5/8	19/32	9/32
21¾–22½	2 1/4	1 1/4	3 1/16	1 1/4	0.857d	0.286d			9	1 7/8	1 3/8	3 3/4	5/8	5/16

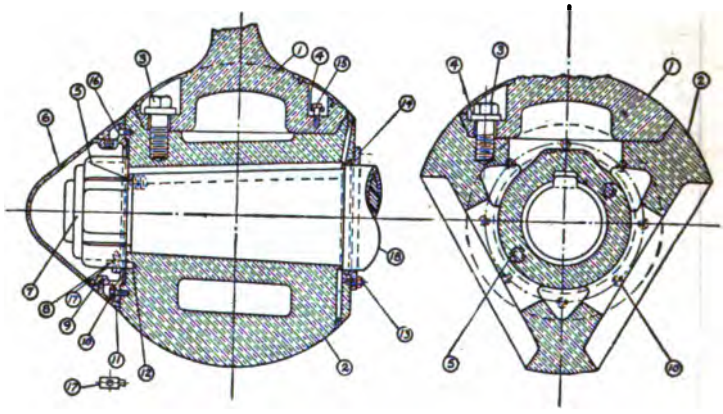


FIG. 6.—Standard three-blade, built-up propeller; for details see Figs. 7 and 8.

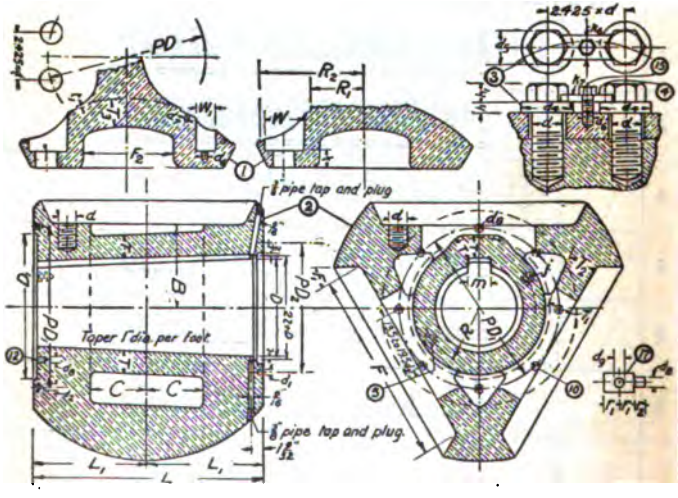


FIG. 7.—Details of three-blade, built-up propeller shown in Fig. 6.

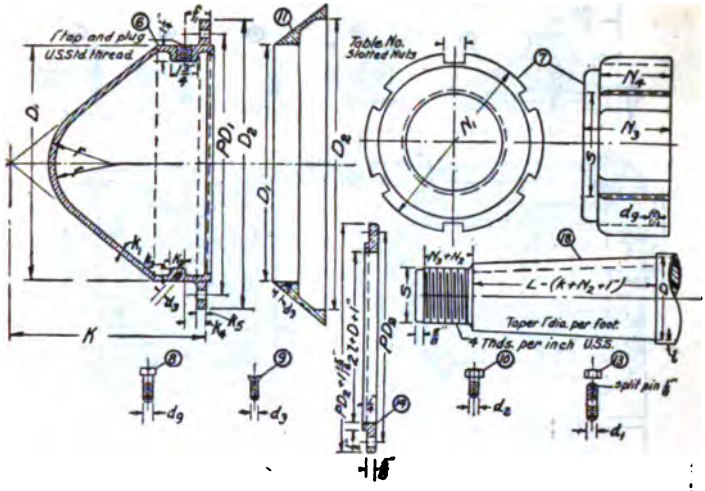


FIG. 8.—Details of three-blade, built-up propeller shown in Fig. 6.

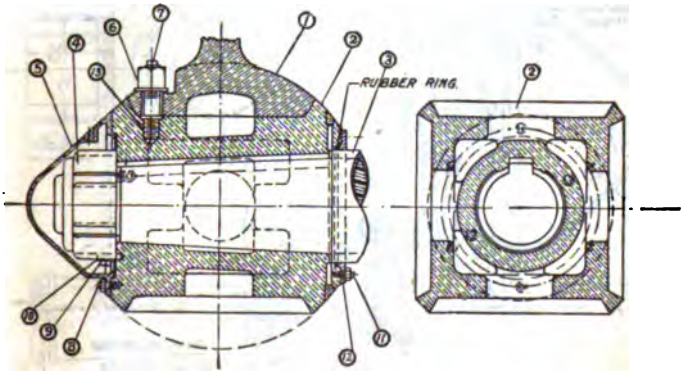


FIG. 9.—Standard four-blade, built-up propeller; for details see Figs. 10 and 11.

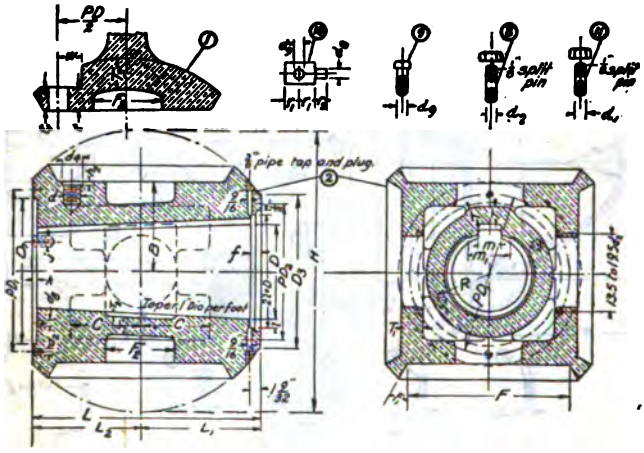


FIG. 10.—Details of four-blade, built-up propeller shown in Fig. 9.

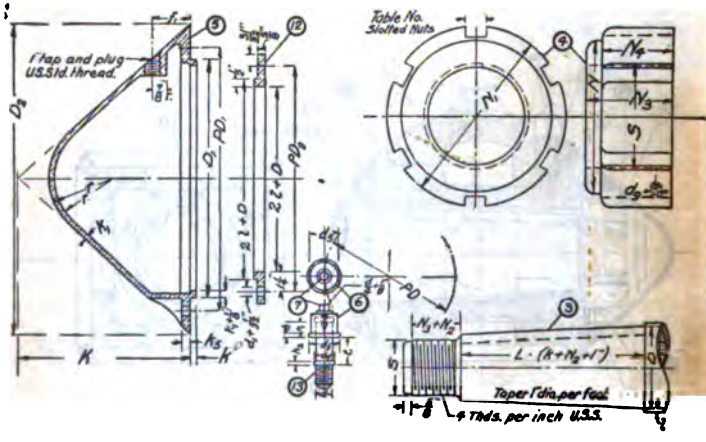


FIG. 11.—Details of four-blade, built-up propeller shown in Fig. 9.

22. Dimensions of Standard Four-blade, Built-up Propeller; see Figs. 9, 10 and 11.

D	B	C	D_1	D_2	D_3	F	F_1[0]	F_2	H	K	L	L_1	L_2	N
8¼	7 9/16	6	12 1/4	15 3/16	13 1/8	14 1/8	28	5 7/8	24	9 1/16	20 1/2	10 7/16	10 1/16	8
8¼– 9 inc.	8¼	6½	13¼	17 1/8	14 1/4	15 9/16	28	6 3/8	26	9 13/16	21¾	11¼	10½	8 3/8
9 – 9½	8 7/8	7	14¼	18 7/16	15 3/16	16½	28	6 7/8	28	10 9/16	23	12 3/8	10 7/8	9 1/8
9½–10¼	9½	7½	15¼	19¾	16 3/16	17 1/16	28	7 3/8	30	11 5/16	24½	13	11¼	9 7/8
10¼–11	10 1/8	8	16 5/16	21 1/16	17½	18 7/8	28	7 7/8	32	12 1/16	26 1/8	13 7/8	11½	10¼
11 –11½	10¾	8½	17 5/16	22 3/8	18 9/16	20	28	8 3/8	34	12 13/16	27¾	14¾	13	11
11½–12¼	11 3/8	9	18 5/16	23 11/16	19 5/8	21 3/16	28	8 7/8	36	13 9/16	29 7/16	15 5/8	13 13/16	11 5/8
12¼–13	12	9½	19 3/8	25	20¾	22 3/8	28	9 5/16	38	14 5/16	31	16 7/16	14 9/16	12 3/8
13 –13¾	12 5/8	10	20 3/8	26 5/16	21 7/8	23 9/16	28	9 13/16	40	15 1/8	32 11/16	17 3/8	15 9/16	13 1/8
13¾–14¼	13¼	10½	21 3/8	27 5/8	22 15/16	24 3/4	28	10 5/16	42	15 13/16	34 5/16	18 5/16	16 1/8	13½
14¼–15	13 15/16	11	22 7/16	28 15/16	24 1/16	25 15/16	28	10 13/16	44	16 5/8	35 15/16	19 1/16	16 7/8	14¼
15 –15¾	14 9/16	11½	23 7/16	30¼	25 1/8	27 1/8	28	11 5/16	46	17 3/8	37 9/16	19 15/16	17 5/8	15
15¾–16½	15 3/16	12	24 7/16	31 9/16	26 1/4	28 11/16	28	11 13/16	48	18 1/8	39¼	20 7/8	18 3/8	15¾
16½–17	15 13/16	12½	25½	32 7/8	27 5/16	29 7/8	28	12¼	50	18 7/8	40 13/16	21 11/16	19 1/8	16 1/8
17 –17¾	16 7/16	13	26½	34¼	28 7/16	30 1/8	28	12 13/16	52	19 5/8	42½	22 9/16	19 15/16	16¾
17¾–18½	17 1/16	13½	27½	35 5/16	29 5/16	31 15/16	28	13¼	54	20 3/8	44 1/8	23 7/16	20 11/16	17½
18½–19¼	17 13/16	14	28¼	36 7/8	30 5/8	33	28	13¾	56	21 1/8	45¾	24 5/16	21 7/16	18¼
19¼–20	18 5/8	14½	29 1/16	38 1/16	31 11/16	34 5/16	28	14¼	58	21 7/8	47 3/8	25 1/8	22¼	19
20 –20½	19	15	30 1/16	39½	32 13/16	35 5/8	28	14¾	60	22 5/8	49	26	23	19¾
20½–21¼	19 5/8	15½	31 5/8	40 13/16	33 7/8	36 1/16	28	15¼	62	23 3/8	50 5/8	26 7/8	23¾	20 1/8
21¼–22	20¼	16	32 5/8	42 1/8	35	37 5/16	28	15¾	64	24 1/8	52¼	27¾	24½	20 7/8
22 –22½	20 7/8	16½	33 5/8	43 7/16	36 1/16	38 3/8	28	16¼	66	24 7/8	53 15/16	28 5/8	25 5/16	21½

Table 22.—*(Continued.)*

D	N_1	N_2	N_3	N_4	PD	PD_1	PD_2	R	S	T	T_1	T_2	T_3	W
8¼	9⅜	½	4⅜	3 13/16	11¾	13⅜		4¾	5½	1½	1 15/16	¼	2 13/16	2
8¼– 9 inc.	9¼	½	5	4	12⅜	14⅛		5⅛	5¾	1⅝	2⅛	¼	2 15/16	2 3/16
9 – 9½	10⅛	½	5⅜	4 5/16	13 1/16	15⅛		5 9/16	6¼	1¾	2¼	¼	3⅛	2⅜
9½–10¼	11½	½	5¾	4 13/16	14 13/16	16⅞		5 15/16	6¾	1⅞	2 7/16	9/16	3⅜	2½
10¼–11	11⅞	½	6⅛	4⅞	15 13/16	17⅝		6 5/16	7	2	2⅝	9/16	3 9/16	2 13/16
11 –11½	12⅝	⅝	6⅝	5 3/16	16⅝	19		6¾	7½	2⅛	2¾	9/16	3 13/16	2⅞
11½–12¼	13½	¾	6¾	5 9/16	17⅞	20⅛		7⅛	8	2¼	2 15/16	⅝	4 1/16	3
12¼–13	14¼	¾	7¼	5⅞	18 13/16	21 1/16		7½	8½	2⅜	3⅛	⅝	4¼	3 3/16
13 –13¾	15⅛	¾	7⅝	6¼	19 15/16	22 5/16		7 15/16	9	2½	3¼	⅝	4½	3⅜
13¾–14¼	15½	¾	7⅞	6⅜	20 7/16	23 1/16		8 5/16	9¼	2⅝	3 7/16	7/16	4¾	3½
14¼–15	16⅜	⅞	8¼	6¾	21 7/16	24 1/16		8 13/16	9¾	2¾	3 9/16	7/16	4 15/16	3 13/16
15 –15¾	17⅜	1	8 13/16	7 1/16	22⅝	25 7/16		9⅛	10¼	2⅞	3¾	7/16	5 1/16	3⅞
15¾–16½	18	1	9⅛	7⅜	23½	26 5/16		9½	10¾	3	3 15/16	½	5⅛	4
16½–17	18¾	1⅛	9½	7 13/16	24½	27 5/16		9⅞	11¼	3⅛	4 1/16	½	5⅜	4 3/16
17 –17¾	19¼	1⅛	9¾	7 15/16	25⅝	29		10 5/16	11½	3¼	4¼	½	5 13/16	4⅜
17¾–18½	20	1⅛	10 3/16	8 1/16	26 9/16	30⅛		10 11/16	12	3⅜	4⅜	9/16	6 1/16	4½
18½–19¼	20¾	1⅛	10⅝	8⅜	27 9/16	31¼		11 1/16	12½	3½	4 9/16	9/16	6 5/16	4 13/16
19¼–20	21⅝	1⅛	11⅛	9	28⅝	32¼		11½	13	3⅝	4¾	9/16	6½	4⅞
20 –20½	22½	1¼	11½	9 7/16	29⅝	33½		11⅞	13½	3¾	4⅞	⅝	6¾	5
20½–21¼	22⅞	1¼	11 11/16	9⅝	30¾	34⅝		12¼	13¾	3⅞	5 1/16	⅝	6 15/16	5 9/16
21¼–22	23⅝	1¼	12⅛	9⅞	31⅝	35¾		12⅝	14⅛	4	5¼	⅝	7 3/16	5⅝
22 –22½	24½	1¼	12½	10 5/16	32 5/16	36⅞		13 5/16	14¾	4⅛	5⅜	1 1/16	7⅜	5½

Made to suit diameter D and thickness T

Table 22.—(*Continued.*)

D	d	d_1	d_2	d_3	d_4	d_5	d_6	d_7	f	f_1	h	h_1	h_2
8¼	See formula at head of Table 21	5/8	5/8	1.9d	1.2d	1.5d+1/8	1/2	1/2	1 1/4	1 7/8	0.233d	0.766d	0.166d
8¼- 9 inc.		5/8	5/8	1.9d	1.2d	1.5d+1/8	9/16	9/16	1 1/4	2 1/16	0.233d	0.766d	0.166d
9 - 9½		5/8	5/8	1.9d	1.2d	1.5d+1/8	9/16	5/8	1 1/4	2 1/4	0.233d	0.766d	0.166d
9½-10¼		5/8	5/8	1.9d	1.2d	1.5d+1/8	5/8	5/8	1 1/4	2 3/8	0.233d	0.766d	0.166d
10¼-11		5/8	5/8	1.9d	1.2d	1.5d+1/8	5/8	5/8	1 1/4	2 1/2	0.233d	0.766d	0.166d
11 -11½		5/8	5/8	1.9d	1.2d	1.5d+1/8	5/8	5/8	1 1/4	2 13/16	0.233d	0.766d	0.166d
11½-12¼		5/8	5/8	1.9d	1.2d	1.5d+1/8	11/16	3/4	1 1/4	2 7/8	0.233d	0.766d	0.166d
12¼-13		5/8	5/8	1.9d	1.2d	1.5d+1/8	3/4	3/4	1 1/4	3	0.233d	0.766d	0.166d
13 -13¾		3/4	3/4	1.9d	1.2d	1.5d+1/8	3/4	7/8	1 1/4	3 3/16	0.233d	0.766d	0.166d
13¾-14¼		5/8	3/4	1.9d	1.2d	1.5d+1/8	3/4	7/8	1 1/4	3 3/8	0.233d	0.766d	0.166d
14¼-15		5/8	3/4	1.9d	1.2d	1.5d+1/8	13/16	7/8	1 1/4	3 1/2	0.233d	0.766d	0.166d
15 -15¾		3/4	3/4	1.9d	1.2d	1.5d+1/8	7/8	1	1 1/4	3 5/8	0.233d	0.766d	0.166d
15¾-16½		3/4	3/4	1.9d	1.2d	1.5d+1/8	7/8	1	1 1/4	3 13/16	0.233d	0.766d	0.166d
16½-17		3/4	7/8	1.9d	1.2d	1.5d+1/8	15/16	1	1 1/4	4	0.233d	0.766d	0.166d
17 -17¾		3/4	7/8	1.9d	1.2d	1.5d+1/8	15/16	1	1 1/4	4 1/8	0.233d	0.766d	0.166d
17¾-18½		3/4	7/8	1.9d	1.2d	1.5d+1/8	1	1	1 1/4	4 1/4	0.233d	0.766d	0.166d
18½-19¼		3/4	7/8	1.9d	1.2d	1.5d+1/8	1	1	1 1/4	4 7/16	0.233d	0.766d	0.166d
19¼-20		3/4	7/8	1.9d	1.2d	1.5d+1/8	1 1/16	1 1/16	1 1/4	4 9/16	0.233d	0.766d	0.166d
20 -20½		3/4	7/8	1.9d	1.2d	1.5d+1/8	1 1/16	1 1/16	1 1/4	4 3/4	0.233d	0.766d	0.166d
20½-21¼		3/4	7/8	1.9d	1.2d	1.5d+1/8	1 1/16	1 1/16	1 1/4	4 7/8	0.233d	0.766d	0.166d
21¼-22		3/4	1	1.9d	1.2d	1.5d+1/8	1 1/8	1 1/8	1 1/4	5 1/16	0.233d	0.766d	0.166d
22 -22½		3/4	1	1.9d	1.2d	1.5d+1/8	1 1/8	1 1/8	1 1/4	5 1/4	0.233d	0.766d	0.166d

Table 22.—(*Continued.*)

D	j	k	k_1	k_2	l	m	m_1	r	r_1	r_2	t	t_1
8¼	1⅛	½	7/32	7/16	Above shaft diam. { 7½"–10 " incl. l = ½" ; 10 "–12½" incl. l = 5/8" ; 12½"–15 " incl. l = ¾"	See formula at head of Table 21	2m	2 11/16	1 3/16	9/16	1 7/8	2 1/16
8¼– 9 inc.	1¼	½	¼	½			....	2 15/16	⅝	⅝	2 1/16	2¼
9 – 9½	1⅜	½	¼	9/16			....	3⅛	⅝	⅝	2¼	2⅜
9½–10¼	1⅜	½	9/32	9/16			....	3⅜	1	¾	2⅜	2 9/16
10¼–11	1½	⅝	5/16	⅝			....	3⅜	1	¾	2½	2¾
11 –11½	1⅝	⅝	5/16	⅝			....	3 13/16	1	¾	2 11/16	2⅞
11½–12¼	1¾	¾	11/32	11/16			....	4 1/16	1⅛	13/16	2⅞	3 1/16
12¼–13	1¾	¾	11/32	¾			....	4¼	1 3/16	⅞	3	3¼
13 –13¾	1⅞	⅞	⅜	¾			....	4½	1 3/16	⅞	3 3/16	3 5/16
13¾–14¾	2	⅞	⅜	13/16			....	4¾	1 3/16	⅞	3⅜	3 9/16
14¾–15	2	1	13/32	13/16			....	5	1 5/16	15/16	3½	3¾
15 –15¾	2¼	1	13/32	⅞			....	5 1/16	1⅜	1	3⅝	3 15/16
15¾–16½	2¼	1⅛	7/16	15/16			....	5 7/16	1⅜	1	3 13/16	4⅛
16½–17	2½	1⅛	15/32	15/16			....	5⅝	1½	1 1/16	4	4¼
17 –17¾	2½	1¼	15/32	1			....	5⅞	1½	1 1/16	4⅛	4 7/16
17¾–18½	2½	1¼	½	1			....	6 1/16	1⅝	1⅛	4¼	4⅝
18½–19¼	2¾	1⅜	½	1 1/16			....	6 5/16	1⅝	1⅛	4 7/16	4¾
19¼–20	2¾	1⅜	17/32	1⅛			....	6½	1 11/16	1¼	4 9/16	4 15/16
20 –20½	2¾	1½	17/32	1⅛			....	6¾	1 11/16	1¼	4¾	5⅛
20½–21¼	3	1½	9/16	1 1/16			....	7	1 11/16	1¼	4⅞	5 5/16
21¼–22	3	1⅝	9/16	1¼			....	7¼	1 13/16	1 5/16	5 1/16	5 5/16
22 –22½	3	1⅝	19/32	1¼			....	7 5/16	1 13/16	1 5/16	5¼	5⅝

SECTION 10

AUXILIARY MACHINERY

BY

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CONTENTS

EVAPORATORS AND DISTILLERS
BY RUSSELL C. JONES

	PAGE
Evaporators:	
Marine Service	1079
Types of Plants	1079
Capacity, Space, Weight	1080
Types of Evaporators	1080
Materials of Construction	1081
Accessories	1085
Methods of Installation	1086
Regenerative Compressor	1091
Piping Arrangement	1093
Operation	1094
Distillers:	
Marine Service	1096
Commercial Types	1097
Evaporator Feed Heaters	1098

FEED WATER HEATERS
BY JOSEPH PRICE

Types and Materials	1103
Installation	1104
Size Required	1107
Design	1108
Operation	1110

OIL COOLERS
BY JOSEPH PRICE

	PAGE
Design	1111
Commercial Types	1113
Operation	1114

CENTRIFUGAL PUMPS
BY M. L. KATZENSTEIN

Characteristic Curves	1115
Construction	1117
Methods of Driving	1122
Circulating Pumps	1123
Boiler Feed Pumps	1127

RECIPROCATING PUMPS
BY M. L. KATZENSTEIN AND M. SPILLMAN

Types	1131
Construction	1133
Commercial Specifications	1133
Tables of Commercial Sizes	1134
Cargo Oil Pumps	1149
Performance Factors	1151
Design Factors	1152
Valve Service	1153

CENTRIFUGAL FANS
By LOUIS LANYI
AND RAINES KESSLER
Definitions, Formulæ and Types of Fans..... 1163
Types of Fan Impellers..... 1170
Parallel Operation..... 1175

HEATING AND VENTILATING Ash Handling Machinery, Miscellaneous
By FRANK W. STERLING
Heating..... 1177
Newport News S. B. & D. D. Co., Practice..... 1178
U. S. Naval Specifications..... 1180
Ventilation..... 1181
Ventilators..... 1184
Combined Heating and Ventilation. 1189
Thermofan System..... 1189
Sirocosone System..... 1191
Ash Handling Machinery..... 1192
Whistle and Siren..... 1196

AIR COMPRESSORS
By FRANK W. STERLING
Properties of Air..... 1198
Flow of Air..... 1199
Work of Compression..... 1200
Air Compressors..... 1201
Testing Air Compressors..... 1203
Operation and Maintenance..... 1204

DECK AUXILIARIES
By FRANK GENTLES
Steering Gear..... 1206
Automatic Follow up Gear for Steering Engine..... 1218
Rudder Torque..... 1218
Size of Steering Gears..... 1219
Steering Gear Control..... 1222
Windlasses..... 1230
Cargo Winches..... 1234
Electric Winches..... 1241
Warping Winches..... 1242
Capstans..... 1242

EVAPORATORS AND DISTILLERS

BY

RUSSELL C. JONES

EVAPORATORS

Marine Service. Evaporators are the **largest steam users** of all marine auxiliaries and require careful attention. They are used to supply boiler feed make-up, cooking, drinking and wash water. The production of fresh water from sea water is accomplished by the vaporization of sea water by means of the condensation of steam in coils that are surrounded by the sea water in a shell. The vapor is usually condensed in a separate condenser, called a **distiller.** In multiple effect plants, the vapor produced is carried along to serve as steam in the coils of a second evaporator or **effect.**

As evaporation proceeds, the water level of the brine in the shell must be maintained by the continuous introduction of sea water as feed. This increases the salinity of the brine and this salinity must be kept below certain limits by the continuous discharge of a small amount of brine (continuous blow-down) or by the periodic discharge of the entire volume of brine when the salinity reaches a certain limiting value. For a complete discussion of blow-down, see "Salt Water Evaporators" by W. L. DeBaufre, Journal of A. S. N. E.. Vol. 27.

Types of Evaporating Plants. There are two general types of marine evaporating plants:

1. The **Low Pressure** Multiple Effect Type is designed to be operated by auxiliary exhaust steam and with vapor under a vacuum. This type is generally installed on large vessels, distilling vessels, transports, repair vessels and on other vessels where its weight is not prohibitive, and space is available and where economy is essential.

2. The **High Pressure** Type, both single and multiple effect, is designed to be operated by boiler steam at reduced pressure as necessary. It is generally installed on destroyers, small vessels, auxiliary vessels, and in cases where the necessary capacity of the plant is low.

Capacity and Test. The U. S. Navy specifications call for the following capacity and test requirements:

Low Pressure Arrangement. The plant will have the **specified capacity** when operating in multiple effect using steam at 10 pounds pressure per gage. The plant under these conditions must evaporate sea water into potable water containing not more than 2 grains chlorine per gallon with an expenditure of not more than 0.35 pound of steam per pound of water, the steam used for operating evaporating plant auxiliaries being excluded. Evaporators will not be rated higher than 60 gallons per day per square foot of heating surface.

The plant will have ½ **overload capacity** when operating in double effect, using steam at 70 pounds pressure, and when distilling water with less than 2 grains of chlorine per gallon the expenditure of steam to be not more than 0.65 pound per pound of water, steam used by auxiliaries being excluded.

The plant will be capable of being operated at least 1000 hours in service at above 70 per cent. of specified capacity without scaling.

High Pressure Arrangement. Plants fitted for operating in **double effect** will have the *specified capacity* when operated in double effect using steam at 70 pounds pressure per gauge, and under these conditions must evaporate sea water into potable water containing not more than 2 grains of chlorine per gallon with an expenditure of not more than 0.7 pound of steam per pound of water distilled, the steam used for driving auxiliaries being excluded. Evaporators will not be rated higher than 60 gallons per day per square foot of heating surface.

For **single-effect** plants the test requirement will be 1.2 pounds steam per pound of water under the above conditions.

The **maximum capacity** of evaporators, when clean, will be at least 40 per cent in excess of the *rated capacity specified*.

The Maximum Evaporator Capacity Required for different classes and sizes of vessels, American practice, is given in Table 1. The ratings for Naval vessels are based on double effect operation. In an emergency a **double effect plant can be run as two single effects,** increasing the maximum capacity of the entire plant about 80 per cent with reduced economy in operation. In general, increasing the number of effects of an evaporator plant decreases the steam required in the evaporator coils per pound of distilled water made. The **Merchant Marine** regularly use single effect evaporators; the larger **passenger vessels** are equipped for double effect duty; **Naval vessels** are usually equipped for double effect duty. In selecting size of evaporator consideration should be given to the number of crew and passengers. (See page 1097 for recommendations for cooking, drinking and wash water allowances.)

Table 1. Evaporator Practice in the United States. Net Tons per 24 Hours, Max. Capacity

S.H.P. of vessel	Navy capital ships	Navy Destroyer and Scout classes	Passenger vessels	Cargo vessels reciprocating engines	Cargo vessels turbines	Sea-going tugs
700	...	9*				
1,500	...	...	..	12		
2,000	...	...	..	12-20	15	15
2,500	...	...	..	30-70†	20-24	
3,000	...	...	..	12		
3,500	...	...	..	24		
10,000	...	...	70	..		
25,000	146	44	...	..		
30,000	219	61	150	32		
90,000	...	219				
180,000	417					

*Cutters. †Oil tankers.

Space Occupied. Fig. 1 graphically shows the **net volume** occupied (without lagging) of all sizes of the principal American makes of evaporators.

Weights of Evaporators. Fig. 2 graphically shows the **net dry weights** of all sizes of the principal American makes of Evaporators.

Commercial Types. Fig. 3 shows the **straight-tube, horizontal evaporator,** steel shell, cast hollow door (Curve *A*, Figs. 1 and 2).

Figs. 4 and 4*a* show the **U-tube vertical evaporator,** steel shell, cast iron door frame and door (Curve *B*, Figs. 1 and 2).

Fig. 5 shows the **vertical unit coil evaporator,** submerged heating surface, steel shell (Curve *C*, Figs. 1 and 2).

Fig. 6 shows the **vertical unit coil evaporator,** submerged heating surface, cast iron shell (Curve *D*, Figs. 1 and 2).

Figs. 7 and 7*a* show the **vertical unit coil evaporator,** submerged heating surface, bronze shell with sheet copper dome (Curve *E*, Figs. 1 and 2, single effect and Curve *F*, Figs. 1 and 2 for double effect duty).

Fig. 8 shows the **vertical unit coil evaporator,** partially submerged heating surface, steel shell (Curve *G*, Figs. 1 and 2).

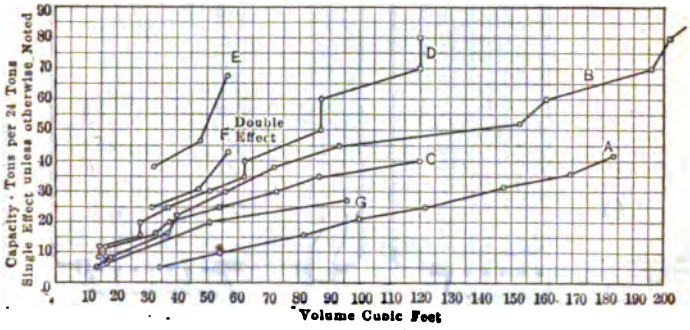


FIG. 1.—Capacities vs. volumes of various types evaporators.

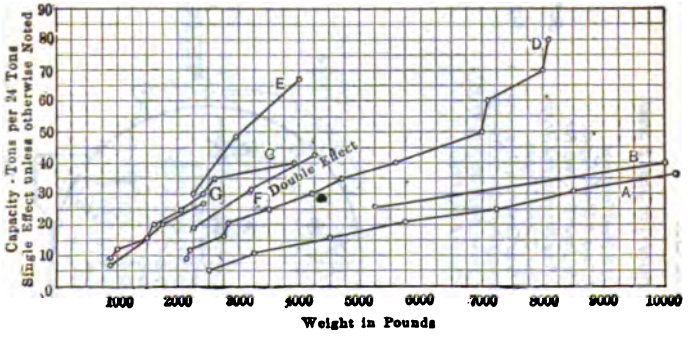


FIG. 2.—Capacities vs. weights of various types evaporators.

Selection of Type and Materials of Construction. Shells are made of cast iron, steel plate, cast bronze, sheet copper or sheet brass. **Steam headers** are made of cast iron or bronze. The **heating surface** is brass or copper tubing, sometimes required tinned. **European practice** is cast iron shells, cast iron manifolds and brass tubing. Cast iron weighs more but resists well the corrosive effect of hot salt water. **Galvanizing** a steel plate shell evaporator is of little value. A steel plate shell eva orator may be made to last indefinitely with careful operation if the heating surface is fully submerged in the brine, preventing the superheating of vapor and the formation

MATERIAL LIST.

Shell Heads and Washers of Class B Steel Boiler Plate.
Main Braces of Class B Steel Forgings.
Angle Stiffeners of Class B Steel Shapes.
Evaporator Supports, Inner Brackets for Rails, Zinc and Dry Pipe Hangers and Strap around Tubes of Class C Steel Shapes.
Dry Pipe and Front Head Castings, Plugs and Return Bends of Compositions, 88 Cu. 10 Sn. 2 Zn.
Tubes of Seamless Drawn Brass 60-70 Cu. 40-30 Zn.
Studs and Bolts of Class A Steel Rods.

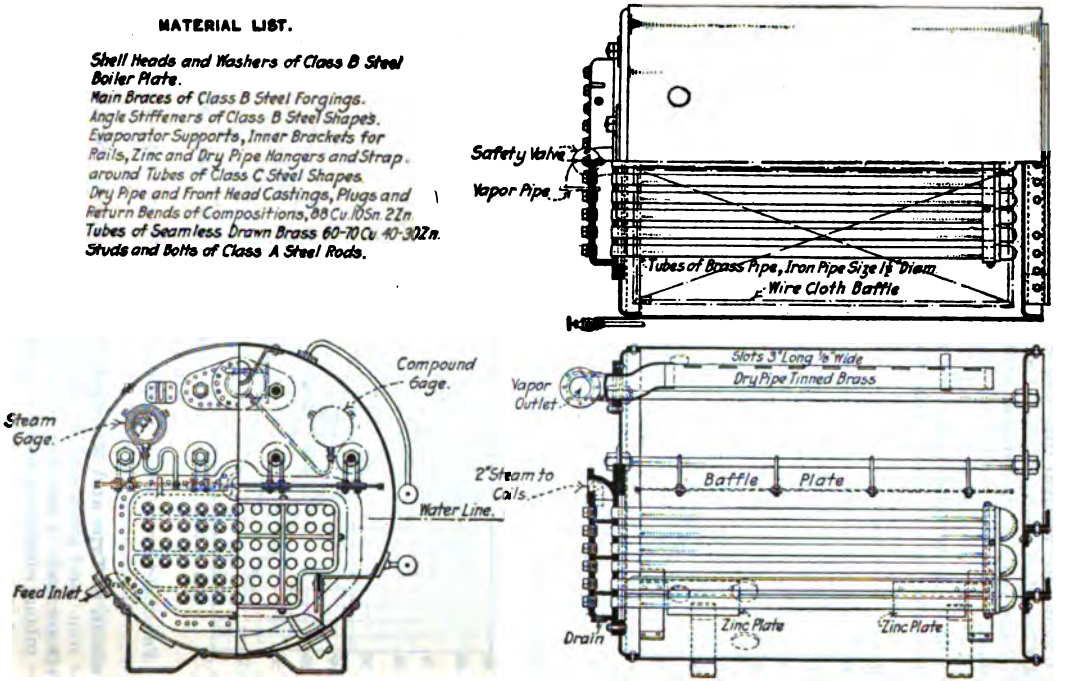


FIG. 3.—Straight tube, horizontal evaporator.

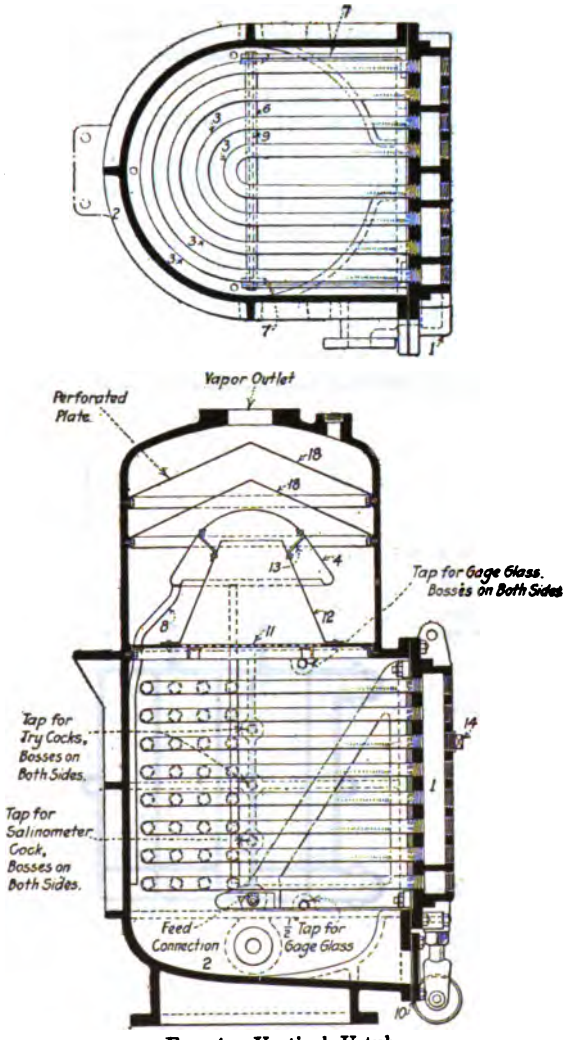

Vapor Outlet
Perforated Plate
Tap for Gage Glass. Bosses on Both Sides
Tap for Try Cocks, Bosses on Both Sides.
Tap for Salinometer Cock, Bosses on Both Sides.
Feed Connection
Tap for Gage Glass
2

FIG. 4.—Vertical U-tube.

of corrosive chlorine gases which attack the upper part of the shell. The following is quoted from *The Babcock and Wilcox Company's "Marine Steam"*: "Salt water is known to be a solvent of iron and steel, and when boiled under

NOMENCLATURE

MARK	NAME OF PART	MATERIAL
1	MANIFOLD	CAST IRON
2	SHELL	CAST IRON
3	TUBES 1"O.D. NO 14 B.W.G.	COPPER, 99.5% PURE
4	BAFFLE CAP	COPPER, 99.5% PURE
5	JACK SCREWS	STEEL, CLASS A CASE HARDENED
6	DISTANCE PIECES	COPPER
7	TUBE SUPPORTS	CAST IRON
8	DRAIN TUBES	COPPER
9	TIE ROD	BRONZE
10	SLUDGE DOOR	CAST IRON
11	PERFORATED PLATE	STEEL, GALVANIZED
12	BAFFLE	COPPER, 99.5% PURE
13	CLIPS	BRASS
14	PIPE PLUGS	BRASS
15	BEARING	CAST STEEL
16	ROLLERS	CAST IRON
17	PINS	STEEL
18	PERFORATED PLATE	COPPER, 99.5% PURE

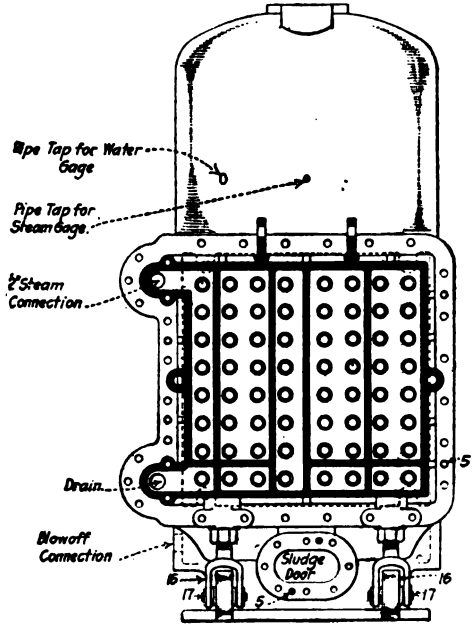
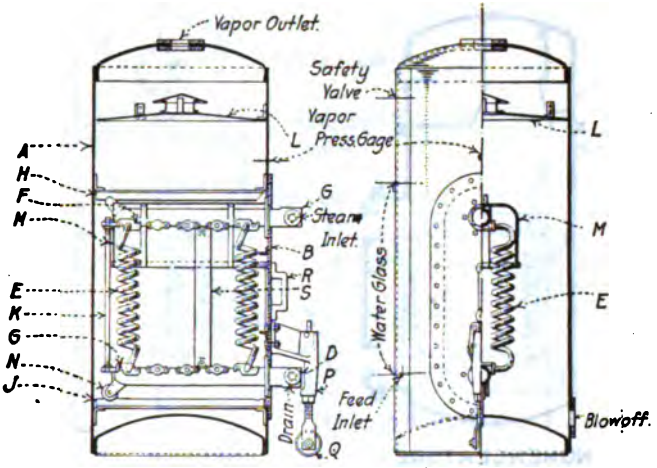


FIG. 4a.—Evaporator.

high pressure the magnesium chloride, about 250 grains of which are contained in every gallon, becomes highly corrosive." If the water level is carried too low in an evaporator, the superheating effect of the dry tube surface above the water level may break up the magnesium chloride into hydrochloric acid

and magnesia, causing unnecessary corrosion of the evaporator shell and vapor piping.

Evaporators should be designed for minimum deck space and accessibility of all parts of the heating surface for cleaning, inspection and replacement. Evaporators are often built with too much heating surface in the shell. There should be an ample vapor space above the water level. If interior



NOMENCLATURE.

MARK	NAME OF PART.	MATERIAL
A	SHELL - INCLUDING TOP AND BOTTOM HEADS, DOOR FRAME, CLIPS ON SHELL, ETC.	STEEL
B	DOOR	"
C	TOP MANIFOLD	CAST IRON
D	BOTTOM MANIFOLD	"
E	COIL COMPLETE	BRASS
F	TAPER NUTS FOR COIL STUDS	"
G	STUDS FOR COIL FLANGES	"
H	MANIFOLD GUIDE	STEEL
J	BOTTOM TRACK	STEEL
K	MANIFOLD BRACE	BRASS
L	CROWN BAFFLE	SHEET IRON
M	ENVELOPING BAFFLE	BRASS
N	MANIFOLD ROLLER WITH PIN & WASHERS	BRONZE
P	BRACKET FOR DOOR ROLLER	CAST STEEL
Q	DOOR ROLLER WITH FORK	STEEL
R	DOOR HANDLE	"
S	ZINC BAR	ZINC

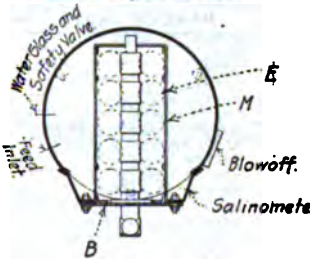
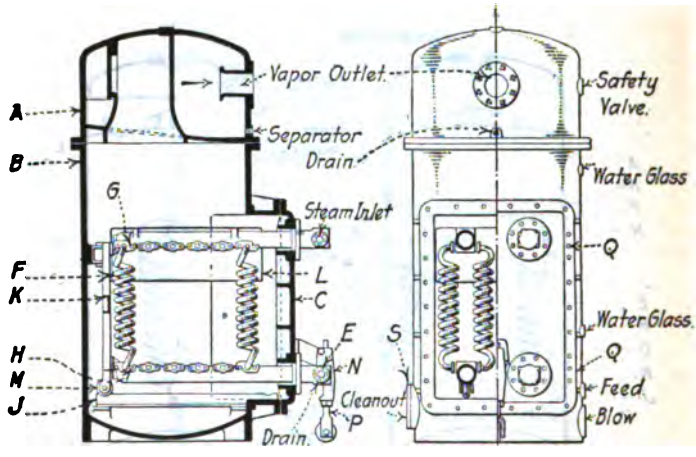


FIG. 5.—Vertical unit coil evaporator, submerged type, steel shell, brass coils.

baffles are employed they should be of suitable material to withstand corrosion. Certain patented forms of heating surface are designed to crack off scale without hand-cleaning. Straight tube evaporators can be cleaned in this manner only to a limited extent.

Accessories Necessary to Good Operation. Every evaporator should be fitted with safety valve on shell, set at 10 lbs. gage for single effect evaporators (U. S. Steamboat Inspection Rules), water gage glass with upper con-

nection well above the maximum water level, high pressure steam gage, compound pressure gage on evaporator shell, *continuous discharge* steam trap to drain condensate from heating surface, air pet-cock on drain line to steam trap, salinometer pot direct connected to shell of evaporator. A set of salinometers should be kept on board at all times. Sea water analysis apparatus is desirable.

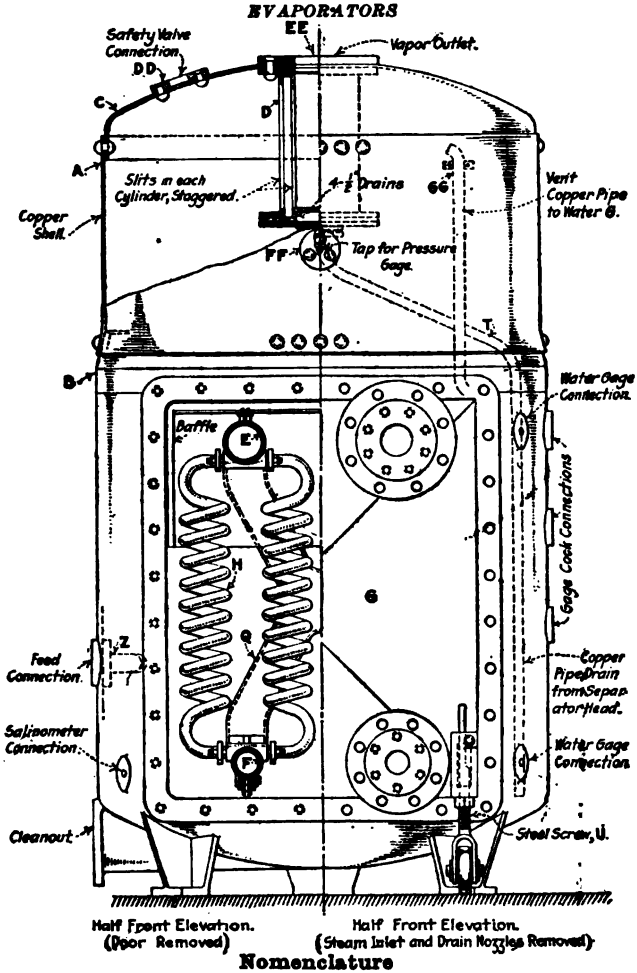


NOMENCLATURE.

MARK	NAME OF PART	MAT'L
A	SEPARATOR HEAD	C.I.
B	SHELL	C.I.
C	DOOR	C.I.
D	TOP MANIFOLD	C.I.
E	BOTTOM MANIFOLD	C.I.
F	COIL COMPLETE	BRASS
G	TAPER NUTS FOR COIL STUDS	"
H	STUDS FOR COIL FLANGES	"
J	BOTTOM TRACK	STEEL
K	MANIFOLD BRACE	BRASS
L	ENVELOPING BAFFLE	BRASS
M	MANIFOLD ROLLER WITH PIN	BRONZE
N	BRACKET FOR DOOR ROLLER	CAST ST.
P	DOOR ROLLER WITH FORK	STEEL
Q	JACK SCREW	"
R	BOLTS AND NUTS FOR A-B	"
S	CLEANOUT COVER	"

FIG. 6.—Vertical unit coil evaporator, double manifold, submerged type, cast iron shell, brass coils.

Methods of Installation. (1) Single effect discharging to main condenser through back pressure valve, Fig. 9. (2) Single effect discharging to main condenser through back pressure valve, with by-pass to small distiller between evaporator and back pressure valve, Fig. 10. (3) Single effect discharging direct to distiller only, Fig. 11. (4) Double effect evaporator plant using high pressure steam, with second effect discharging to distillers above atmospheric pressure, Fig. 12. (5) Double effect utilising low pressure steam with



Nomenclature

Mark	Material	Description
A	Copper	Upper shell
B	Comp. G	Lower shell
C	Copper	Top head
D	Comp. G	Separator
E	Comp. G	Top manifold
F	Comp. G	Bottom manifold
G	Comp. G	Door
H	Comp. A	Coil tinned inside and outside
L	Comp. G	Steam drain waterh'd casting
O	Comp. G	Trolley rail
Q	Comp. G	Manifold brace
R	Comp. G	Trolley wheel
T	Copper	Pipe drain from separator

Mark	Material	Description
U	Class "C" Bolt mat'l	Door roller fork and nut
V	Cast iron	Door roller
X	Class "B" Cast steel	Steam inlet nozzle
Y	Cast steel	Steam drain nozzle
Z	Comp. G	Feed inlet nozzle
BB	Copper	Baffle
DD	Comp. G	Pad for safety connection
EE	Comp. G	Pad for vapor outlet connections
FF	Comp. G	Pad for press. gage connection
GG	Copper	Pipe vent for water gage connection

[illegible] coil evaporator, submerged type, bronze shell

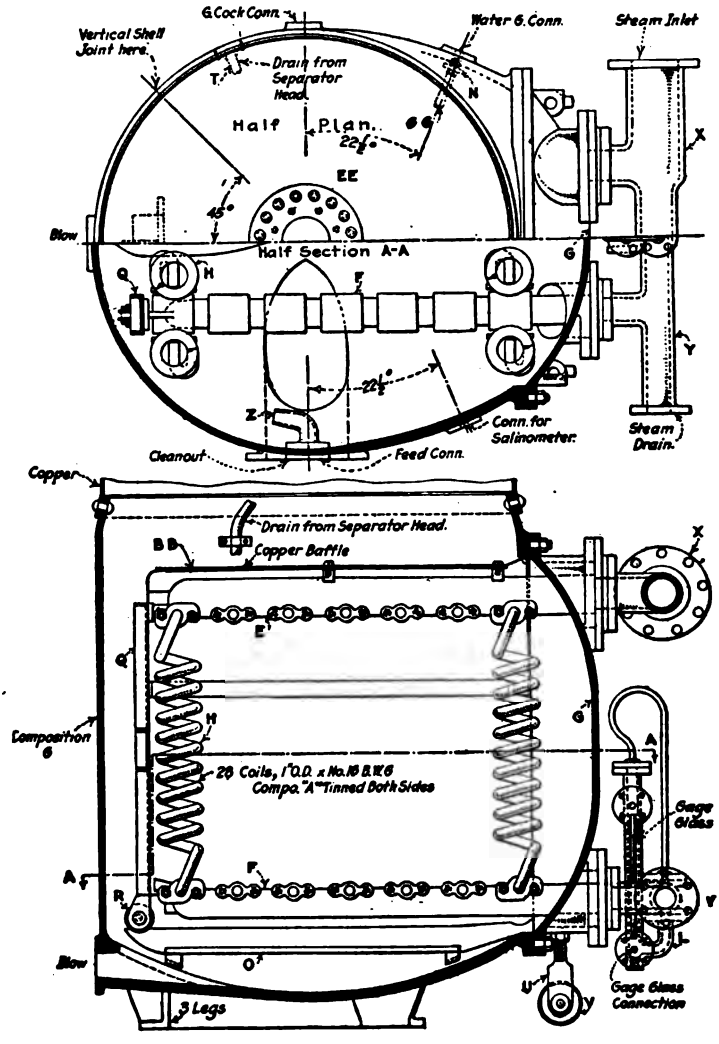


Fig. 7a.—Vertical unit coil evaporator, submerged type bronze shell.

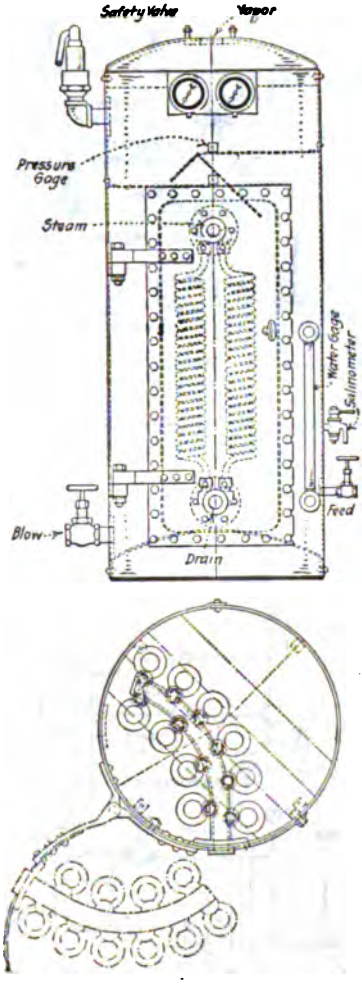


FIG. 8.—Vertical unit coil evaporator, partially submerged type, steel shell.

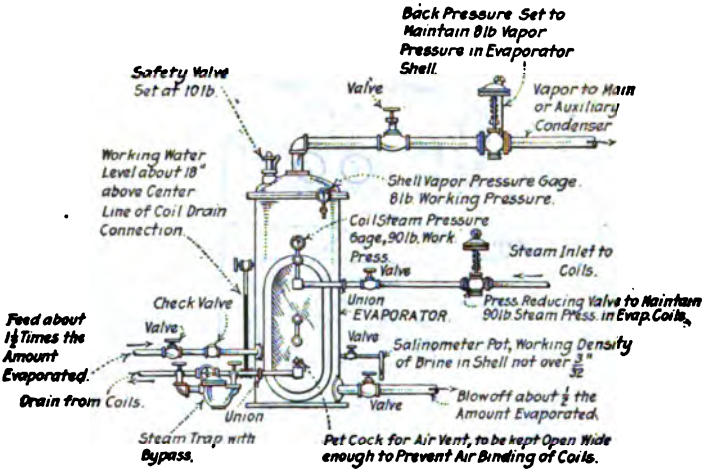

Back Pressure Set to Maintain 8lb Vapor Pressure in Evaporator Shell.
Safety Valve Set at 10lb.
Valve
Vapor to Main or Auxiliary Condenser
Working Water Level about 18" above Center Line of Coil Drain Connection.
Shell Vapor Pressure Gage. 8lb Working Pressure.
Coil Steam Pressure Gage, 90lb. Work. Press.
Steam Inlet to Coils.
Valve
Check Valve
Union
EVAPORATOR
Press. Reducing Valve to Maintain 90lb Steam Press. in Evap. Coils.
Feed about 1½ Times the Amount Evaporated.
Valve
Valve
Salinometer Pot, Working Density of Brine in Shell not over 3/32"
Drain from Coils.
Union
Valve
Blowoff about ½ the Amount Evaporated.
Steam Trap with Bypass.
Pet Cock for Air Vent, to be kept Open Wide enough to Prevent Air Binding of Coils.

FIG. 9.—Layout of evaporator for connection to main or auxiliary condenser.

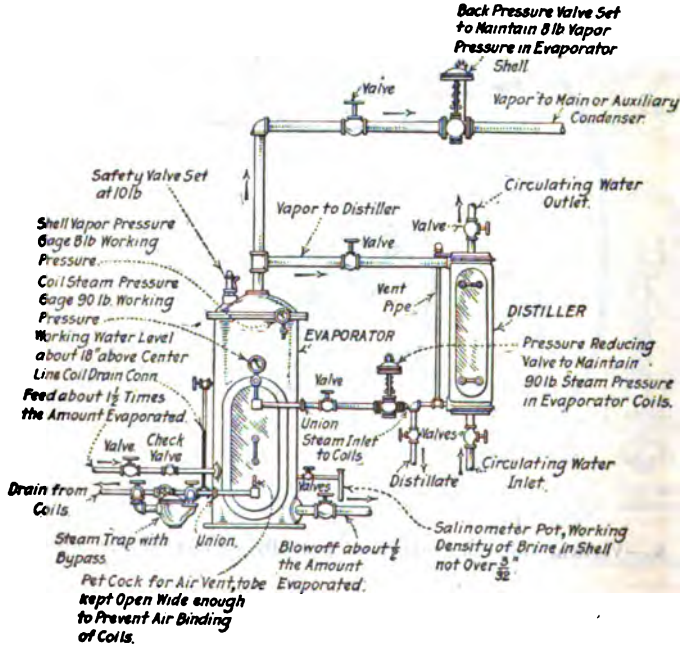

Back Pressure Valve Set to Maintain 8 lb Vapor Pressure in Evaporator Shell
Valve
Vapor to Main or Auxiliary Condenser
Safety Valve Set at 10lb
Circulating Water Outlet.
Vapor to Distiller
Valve
Valve
Shell Vapor Pressure Gage 8lb Working Pressure.
Coil Steam Pressure Gage 90 lb. Working Pressure
Vent Pipe
DISTILLER
EVAPORATOR
Working Water Level about 18" above Center Line Coil Drain Conn.
Pressure Reducing Valve to Maintain 90lb Steam Pressure in Evaporator Coils.
Feed about 1½ Times the Amount Evaporated.
Valve
Union
Steam Inlet to Coils
Valves
Valve
Check Valve
Circulating Water Inlet.
Distillate
Drain from Coils.
Valves
Salinometer Pot, Working Density of Brine in Shell not Over 3/32"
Steam Trap with Bypass
Union.
Blowoff about ½ the Amount Evaporated.
Pet Cock for Air Vent, to be kept Open Wide enough to Prevent Air Binding of Coils.

FIG. 10.—Layout of evaporator for connection to distiller and condenser.

second effect discharging to distillers under vacuum. (6) Regenerative single effect using expansive steam nozzle to recompress a portion of the distilled vapor for use as part of the heating steam, Fig. 13.

Regenerative Compressor. By fitting to an evaporator the regenerative compressor shown in Fig. 13, the amount of live steam required for the production of a given amount of fresh water is reduced since part of the vapor from the evaporator is compressed and commingled with the live steam supplied to the coils. By this means, an efficiency between 2 effect and 3 effect operation is attained in a single shell. The compressor is regularly supplied with a by-pass containing an orifice, thus permitting the evaporator to be operated in the usual manner. A specially graduated steam gage is

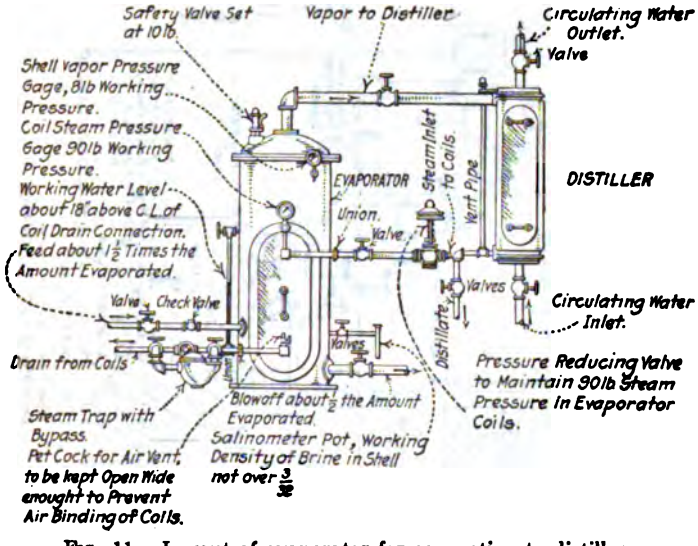

Safety Valve Set at 10lb.
Vapor to Distiller
Circulating Water Outlet.
Valve
Shell Vapor Pressure Gage, 8lb Working Pressure.
Coil Steam Pressure Gage 90lb Working Pressure.
Working Water Level about 18" above C.L. of Coil Drain Connection.
Feed about 1½ Times the Amount Evaporated.
EVAPORATOR
Steam Inlet to Coils.
Vent Pipe
DISTILLER
Union.
Valve
Valves
Valve
Check Valve
Circulating Water Inlet.
Drain from Coils
Valves
Distillate
Pressure Reducing Valve to Maintain 90lb Steam Pressure in Evaporator Coils.
Steam Trap with Bypass.
Pet Cock for Air Vent, to be kept Open Wide enough to Prevent Air Binding of Coils.
Blowoff about ½ the Amount Evaporated.
Salinometer Pot, Working Density of Brine in Shell not over 3/32

FIG. 11.—Layout of evaporator for connection to distiller.

also provided to indicate the steam consumption of the evaporator with either the compressor or the by-pass orifice in operation. This system is claimed to have the following advantages:

(1) The steam consumption for the production of a given amount of fresh water is reduced more than 50 per cent as compared with single effect operation.

(2) When applied to evaporators usually operated in double effect, each evaporator may be operated at over 2 effect efficiency independently of the other evaporator.

(3) The steam consumption of the plant is indicated at all times.

(4) The vapor compressor employed, having no moving parts, is not subject to wear and hence retains its initial efficiency indefinitely.

It must be borne in mind, however, that the capacity of an evaporator is reduced very considerably by using this device. This should not influence its desirability for merchant freighters where, due to the use of tankage the evaporators are seldom operated at more than 60 per cent of rating.

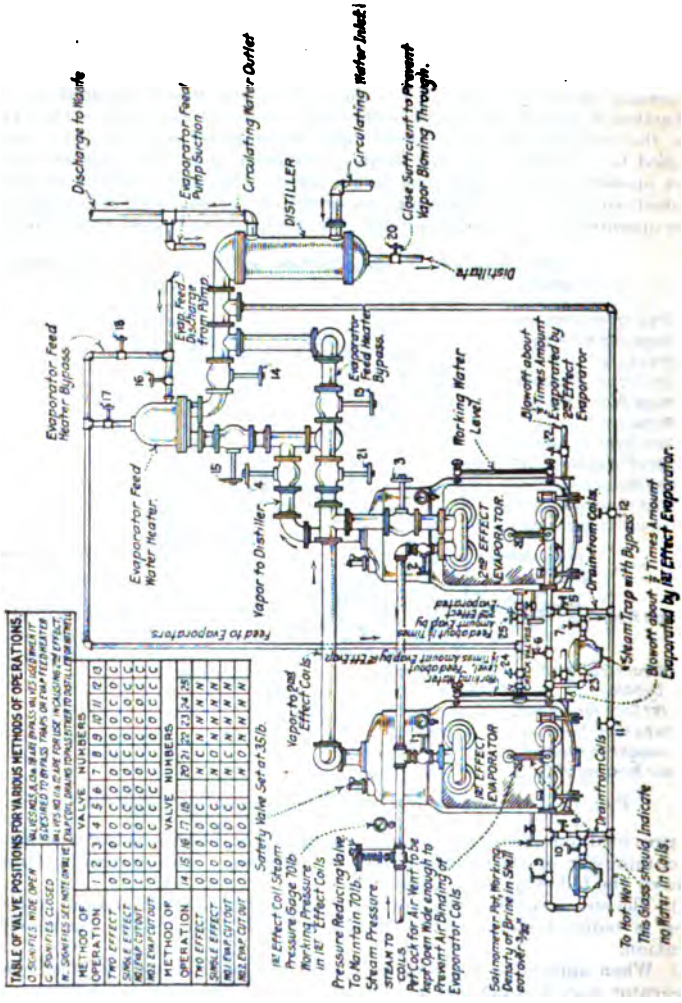


TABLE OF VALVE POSITIONS FOR VARIOUS METHODS OF OPERATIONS
O. SIGNIFIES VALVE OPEN
C. SIGNIFIES CLOSED
N. SIGNIFIES SET WITH HANDLE
VALVE NUMBERS
METHOD OF OPERATION
TWO EFFECT
SINGLE EFFECT
MULTIPLE EFFECT
MULTIPLE EFFECT
Discharge to Waste
Evaporator Feed Pump Suction
Circulating Water Outlet
DISTILLER
Circulating Water Inlet
Close Sufficient to Prevent Vapor Blowing Through.
Evaporator Feed Heater Bypass
Evaporator Feed Water Heater.
Evap. Feed Discharge From Pump
Evaporator Feed Heater Bypass
Vapor to Distiller
Feed to Evaporators
Working Water Level
Blowoff about 3 Times Amount Evaporated by 2d Effect Evaporator
1st EFFECT EVAPORATOR
2d EFFECT EVAPORATOR
Drain from Coils
Steam Trap with Bypass
Blowoff about 3 Times Amount Evaporated by 1st Effect Evaporator.
1st Effect Coil Steam Pressure Gage 70lb
Working Pressure in 1st Effect Coils
Pressure Reducing Valve To Maintain 70lb. Steam Pressure
STEAM TO COILS
Safety Valve Set at 35lb.
Vapor to 2nd Effect Coils
Pet Cock for Air Vent to be Kept Open Wide enough to Prevent Air Binding of Evaporator Coils
Salinometer Pet Working Density of Brine in Shell or Coil—32°
To Hot Well
This Glass should Indicate no Water in Coils.

FIG. 12.—Layout of double effect evaporating plant.

Piping Arrangement. The piping arrangement is **of great importance.** The **vapor pipe** from evaporator to distiller or condenser should have an ample rise in it to reduce entrainment of salt water to a minimum and this riser pipe should be so arranged as to positively drain toward the evaporator without any pockets in which salt may collect. The **steam trap** of continuous flow type must be below and very close to the drain connection from the heating surface. It is permissible for this trap to leak somewhat as this helps to purge the system of air. **Blow-off** connections should be not

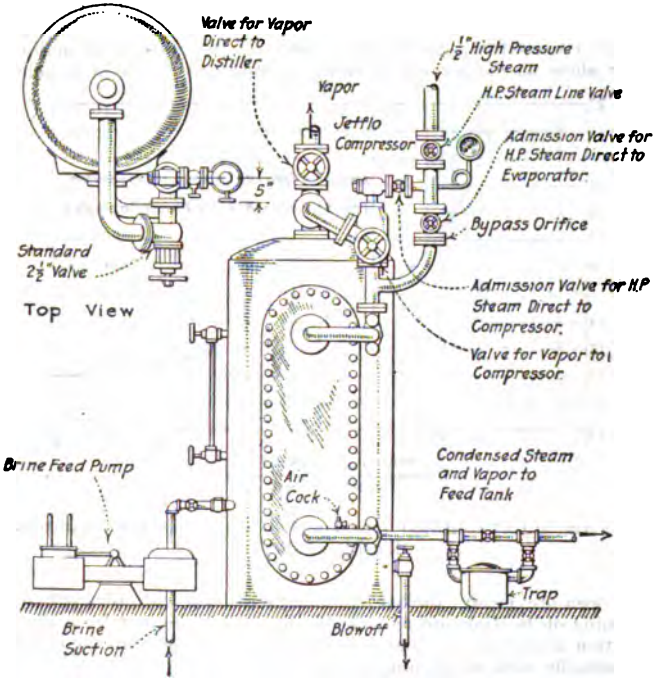


FIG. 13.—Evaporator fitted with regenerative compressor.

less than 2½ inches to facilitate passage of scale. An insufficient blow-down invariably produces impure water and limits the life of an evaporator. For each 1/32 increase in brine density the capacity of an evaporator decreases approximately 12 per cent, other conditions remaining the same.

Fig. 14 shows the necessary blow-down, with varying salinities.

Chemical Test for Salt in Distilled Water. The following **outfit** should be required: (1) Aqueous solution containing 4.101 grams per litre of c.p. crystals of silver nitrate; (2) 2 per cent solution of c.p. potassium chromate; (3) c.p. nitric acid (in glass stoppered bottle); (4) litmus

paper; (5) two burettes, capacity 50 c.c. graduated to 0.2 c.c.; (6) porcelain evaporating dish, capacity 250 c.c.; (7) glass stirring rod.

Test. 50 c.c. of water to be tested is measured from burette into evaporating dish. If this turns litmus blue add nitric acid until neutral or until litmus is turned faintly red. Add 10 drops of potassium chromate. Add silver nitrate from other burette slowly, until water turns faintly pink. Each c.c. of silver nitrate used indicates that water sample contains 1 grain of chlorine or 1.65 grains of sodium chloride per U. S. gallon.

Operation of Evaporators. Successful operation of an evaporator depends upon strict attention to the following rules: (1) Operating conditions must be maintained uniform; (2) density of salt water in evaporator should never exceed $\frac{5}{32}$, preferably a mean of $\frac{3}{32}$ should be maintained; (3) never allow an evaporator to stand without any water in it unless the

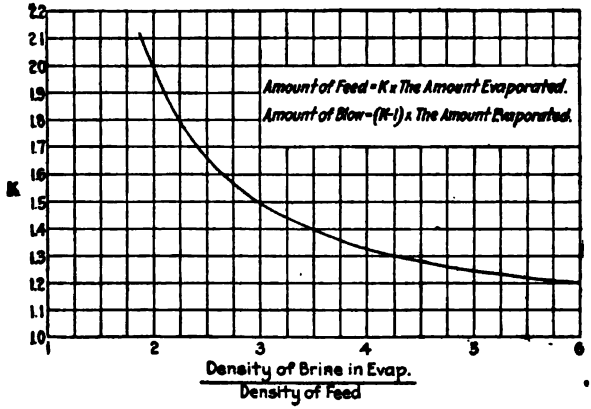


FIG. 14.—Amount of continuous feed and blow necessary to maintain fixed density in shell.

scale has been previously cracked off; this precaution will prevent hard scale from forming on heating surface; (4) bleed air continuously from the steam condensation drain line; (5) maintain as high a water level in the evaporator shell as possible without priming or boiling over.

Feed and Blow Regulation. There are, on the market, two devices for automatically maintaining the proper feed and blow of an evaporator, as well as the proper water level. These devices are not absolutely necessary to good operation of the evaporators but without them more or less constant attendance to the evaporators will be necessary.

One device consists essentially of a **combined feed and brine pump** actuated by a single steam cylinder, the brine and feed pump cylinders being so proportioned as to automatically maintain the desired ratio of feed to blow, and the steam to pump being controlled by a float actuated by water in the evaporator.

The other device consists essentially of **two Venturi tubes**, one for the blow and one for the feed, the difference in pressure at the throats and mouths

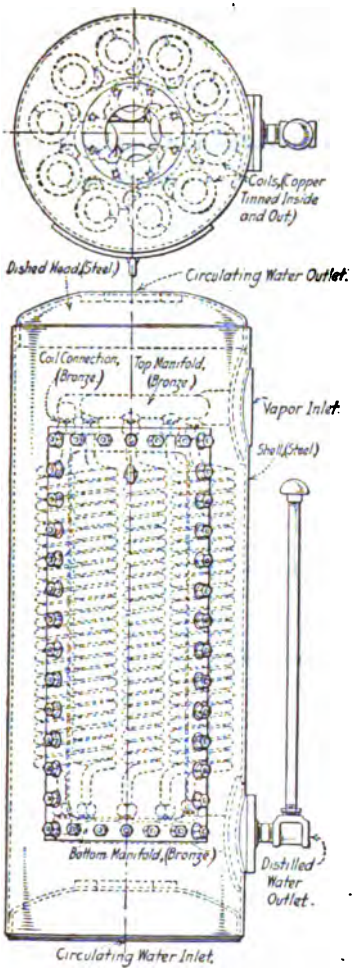


FIG. 15.—Vertical unit coil, steel shell, distiller.

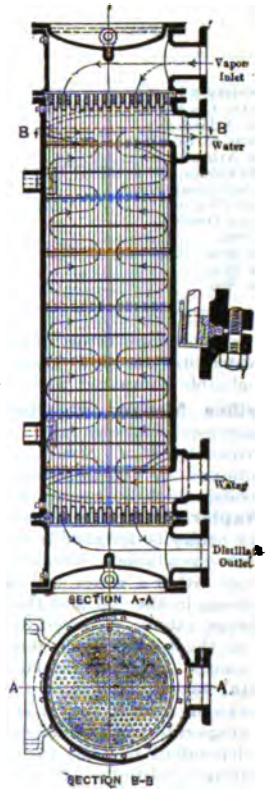


FIG. 16.—Vertical straight-tube, C.I. shell, distiller.

of the Venturi tubes being used to actuate a diaphragm which in turn controls a blow valve in such manner as to fix the ratio of feed to blow. A float actuated by the water level in the evaporator controls the feed valve and so fixes the water level in the evaporator.

Blow-down. Table 2 shows clearly the actual gallons of blow-down required.

Table 2. Gallons of Feed and Blow-down in Various Seas

Sea	Parts of salt per 1000	32ds density	Per 1000 gal. distilled water	
			Gal. feed	Gal. blow-down at 3/32 density
One thirty-second	31.25	1.00	1500	500
Atlantic Ocean, 41° N. 36° W.	38.40	1.23	1695	695
North Atlantic	42.60	1.36	1830	830
Atlantic at Equator	39.42	1.26	1727	727
South Atlantic	41.20	1.31	1782	782
Pacific Ocean, 25° S. 93° W.	35.20	1.12	1595	595
Mediterranean Sea	39.40	1.26	1725	725
British Channel	35.50	1.13	1605	605
German Ocean	32.80	1.05	1535	535
Irish Sea	33.76	1.08	1560	560
Baltic Sea	6.60	0.21	1076	76
Black Sea	21.60	0.69	1295	295
Arctic Sea	28.30	0.91	1435	435

It is a common error, in operating evaporators, to endeavor to save heat by restricting the blow-down. The amount of heat lost in blow-down water is negligible compared with the total steam required in an evaporator.

Orifice Method of Operation. An improved method of operating evaporators whereby any desired constant capacity may be produced is described by *M. C. Stuart* in the *Journal A.S.N.E.*, Feb., 1919. The result is obtained by the application of a **steam orifice** to the main steam line to the coils, which by reason of the law of constant flow of steam, as expressed by **Napier's formula**, effects a constant production of vapor.

The orifice is designed from the known quantity of steam required and the pressure available. The condensate is drained from the coils, as usual, through either a water seal or a trap. The temperature difference between the steam in the coil and the vapor in the shell is automatically adjusted to whatever value is required to transmit all the heat involved in the condensation of the steam which flows into the coils from the orifice. The orifice may consist of a **plate** inserted between two flanges, or it may be included in a **special valve** which is on the market. This method has shown under test remarkable results and is claimed to have the following **advantages:** (1) The evaporator may be operated continuously at any desired capacity, which is independent of scale formation, salinity in shell, water level and other operating conditions; (2) the liability of priming is reduced to a minimum; (3) the formation of scale is reduced and the amount of scale is indicated by the coil pressure; (4) steam consumption is definitely known; (5) operation is simplified and stabilized.

DISTILLERS

Marine Service of distillers is to condense all or a portion of the vapor from the evaporators to provide potable water. Table 3 gives general data on capacities required for various types of vessels.

Table 3. Size Distiller Required

Type of vessel	U. S. gal. per capita per 24 hours			
	Drinking and galley	Wash water	Total	Notes
U. S. Naval vessels	2.00	5-8	7-10	A
British Navy	2.35	...	...	B
Passenger vessels, Merchant Marine	3.00	2-5	5-8	C
Emigrant ships	1.00	2	3	
Freight vessels, Merchant Service	2.00	2	4	D
Yachts	4.00	10	14	D
Horses, additional capacity per animal	10.00	..	10	E

A. Recommendation of the Bureau of Steam Engineering, U. S. N. Includes water for paint work and scrubbing.
B. Macrow & Woollard's Pocket Book.
C. British Board of Trade estimates 3.75 Imperial Gallons per capita for first and second cabin passengers for all purposes.
D. Fresh water allowance per capita should be ample when figuring on a small number of persons.
E. Par. 299, U. S. Army Transport Service Regulations .

Space Occupied. Figs. 10, 11 and 12, pages 1090 to 1092, show the approximate relative size of distillers and evaporators. It should be noted that for a **single-effect** evaporator the distiller will be about twice the size that would be used for two of the same evaporators operating in **double effect,** provided steam pressures and final vapor pressures are the same, inasmuch as vapor from the first-effect evaporator is condensed in the coils of the second-effect evaporator.

Commercial Types. Distillers are of two general types: one has its cooling surface made up of a number of coils, and the other type has its cooling surface made up of a number of straight tubes. In either case the vapor is usually passed through the tubes with circulating water outside of tubes. **Vertical unit coil** distiller, steel shell, is shown in Fig. 15. **Vertical straight-tube** distiller, cast iron shell, is shown in Fig. 16. **Vertical unit coil** distiller, cast iron shell, is shown in Fig. 17. **Vertical straight-tube** distiller, copper or brass shell, shown in Fig. 18, is used extensively in Naval work.

Selection of Types. Shells are made of steel plate, cast iron, sheet copper or brass. **Vapor headers** are made of cast iron or composition. **Heating surface** is made of brass or copper tubing, tinned to prevent contamination of the distilled water by dissolved brass or copper. Cast iron vapor headers are sometimes galvanized, but present practice tends away from galvanizing. Cast iron shells weigh more than steel but resist well the corrosive action of salt water used as condensing medium. Copper and brass shells are used only in Naval practice.

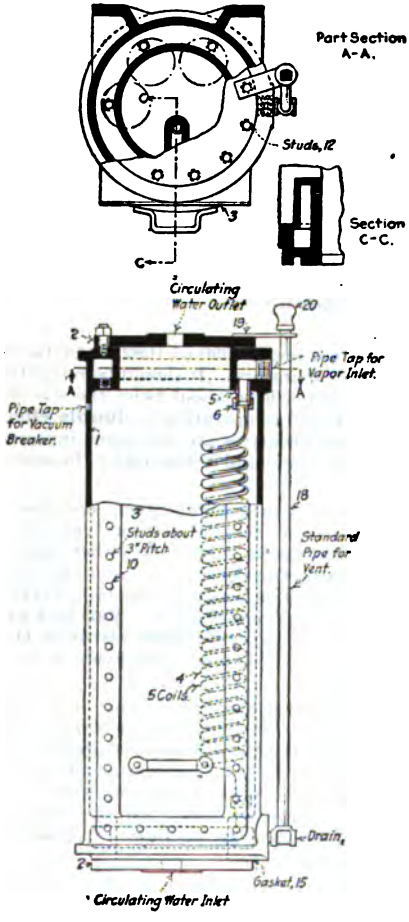
Accessories necessary to operation are vacuum breaker on shell, and vent pipe from distiller water main.

Methods of Installation. Figs. 9, 10, 11 and 12, on pages 1090, 1091 and 1092 show methods of installing distillers in connection with evaporators, condensers, etc.

For Chemical Test for Salt in distilled water, see page 1093.

EVAPORATOR FEED HEATERS

Marine Service of evaporator feed heaters is to preheat the evaporator feed water, utilizing exhaust vapor, usually the vapor from the last-effect evaporator, thus decreasing steam consumption in the evaporator coils.



NOMENCLATURE.

MARK.	MATERIAL.	DESCRIPTION.
1	CAST IRON	SHELL
2	" "	TOP & BOTTOM HEAD
3	STEEL PLATE	DOOR
4	COPPER	COILS
5	COMERC. BRONZE	COIL SPUDS
6	" "	COIL UNION NUTS
10	STEEL	DOOR STUDS
12	STEEL	COVER STUDS
15	RED RUBBER	HEAD GASKETS $\frac{3}{16}$" THICK
18	STD. ST. PIPE	VENT PIPE
19	STEEL PLATE	VENT PIPE BRACE
20	COMERC. BRONZE	VENT PIPE ROSE
	TOBIN BRONZE	COIL FERRULE
	STEEL	DOOR NUTS
	"	COVER NUTS
	RED RUBBER	DOOR GASKETS $\frac{3}{16}$" THICK
	STEEL	STUDS FOR COMC. WATER CONN'S

FIG. 17.—Vertical unit coil, C. I. shell, distiller.

This is the same service that feed-water heaters render to boilers. Evaporator feed heaters are usually omitted in merchant ship installations and are generally used in Naval work. In general this auxiliary is used to better economy of the evaporator plant but does not materially increase capacity.

Size. This equipment is usually built special by the evaporator manufacturer, to suit the conditions of the evaporator plant in question. To be most efficient the heater should provide ample storage space for hot feed water, as feeding of the evaporator is generally intermittent.

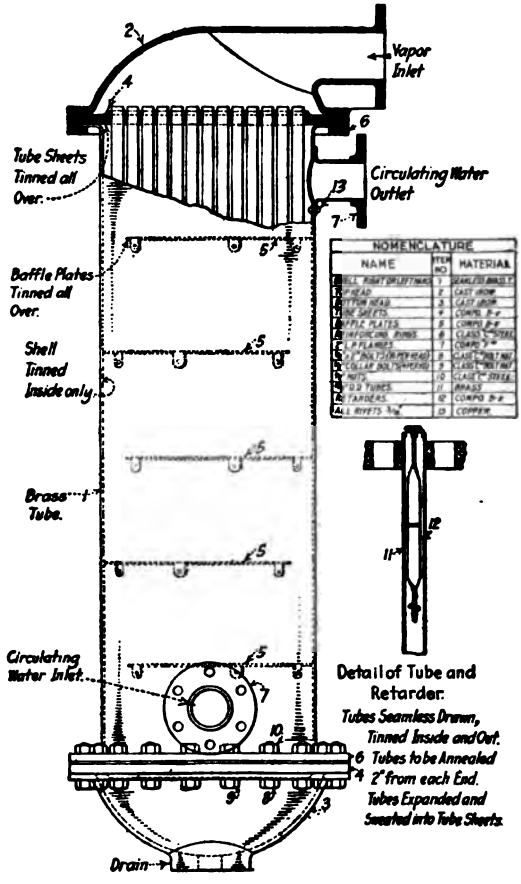


FIG. 18.—Vertical straight-tube, copper or brass shell, distiller.

Selection of Types. For Naval work shells are made of copper or composition, heating surface of brass or copper tubing. Heating surfaces are usually tinned, to prevent contamination of vapor used for heating, as this condensed vapor is returned to fresh water tank for drinking purposes.

Installation is made in the feed line to evaporator, water passing through shell and heated by vapor taken from vapor line to distiller, as shown in Fig. 12, page 1092.

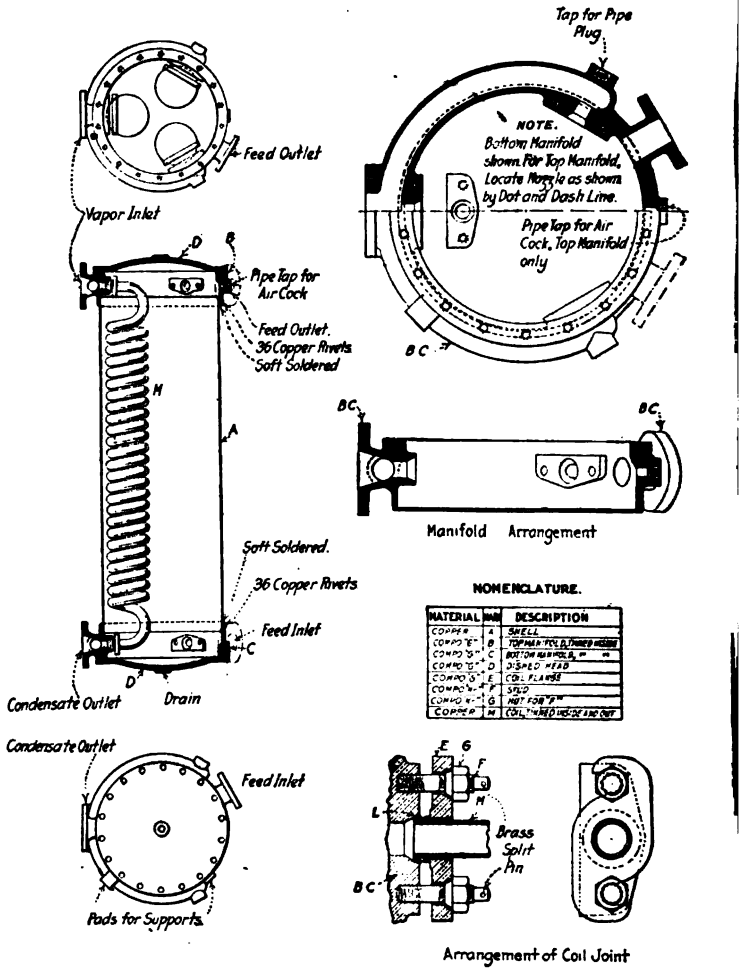


FIG. 19.—Evaporator feed heater, vertical unit coil.

Space and Weight. The following table gives the weights and cubical contents of the various size evaporator feed heaters of the principal makes

with the size evaporator plants with which each is used. Type B provides but little storage space for hot feed water.

Type of evaporator feed heater	Normal capacity of double-effect evap. plant—U. S. gallons per 24 hours	Cubical contents of evap. feed heater—cu. ft.	Weight of evap feed heater, lbs.
A	10,500	4.5	430
B	10,500	3	240

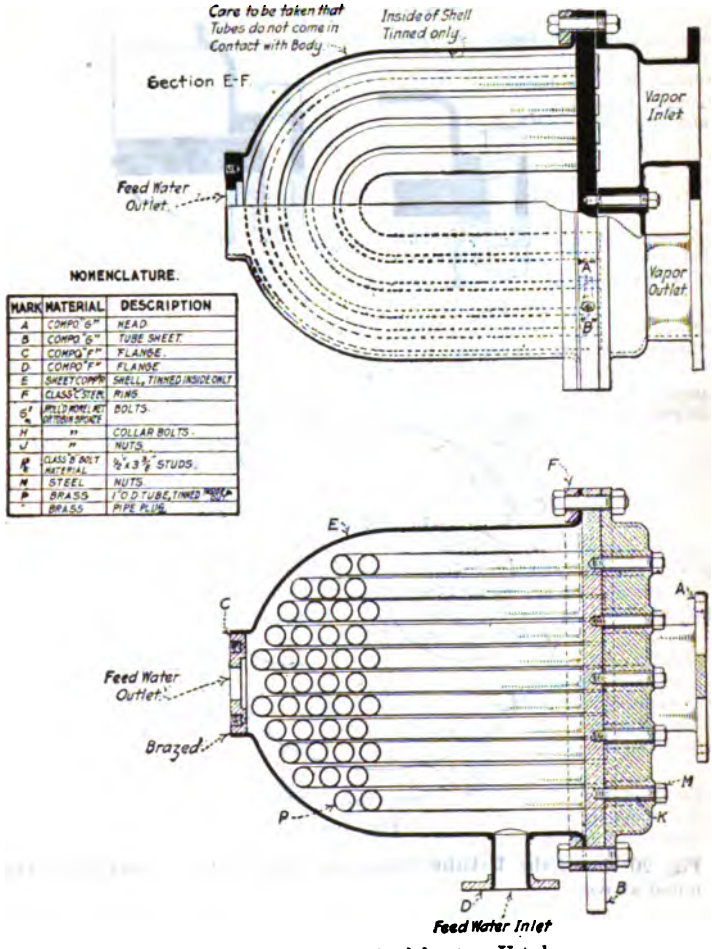
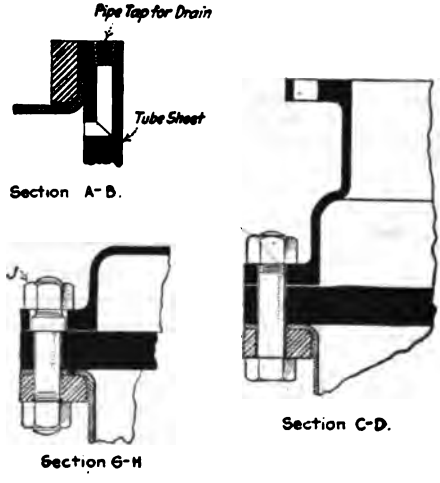


FIG. 20.—Evaporator feed heater, U-tube.

Commercial Types. Fig. 19 shows the **vertical unit coil evaporator** feed heater, copper shell, type A noted above.



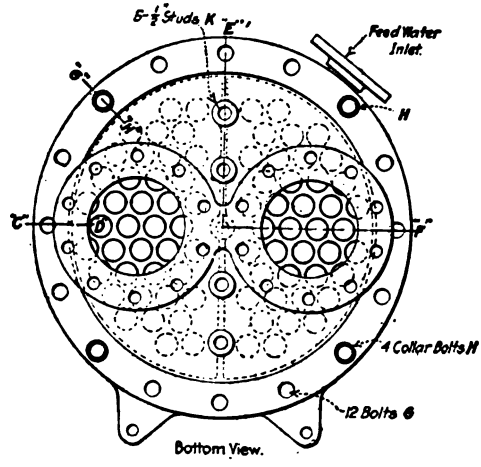


FIG. 20a.

Fig. 20 shows the U-tube evaporator feed heater, copper shell, type B noted above.

FEED WATER HEATERS

BY

. JOSEPH PRICE

Marine Service of feed water heaters is to preheat feed water to the boilers, utilising exhaust steam as the heating agent, thus effecting a saving in fuel used, and reducing boiler strains and distortion.

Saving Effected by Boiler Feed Heating. In general figures 1 per cent less coal is required, or the boiler capacity is increased 1 per cent, for every 11 deg. F. that the feed water is heated. In the average plant the total possible fuel saving, due to feed water heating, varies from 8 to 15 per cent, depending upon conditions. To compute the saving the following formula may be used:

$$\text{Per cent of Coal Saved} = \frac{T_2 - T_1}{T_S - T_1 + L} \times 100$$

Where T_2 = temperature in degrees Fahrenheit to which feed water can be heated, T_1 = hot-well temperature, degrees Fahrenheit, L = latent heat of steam at boiler pressure, and T_S = temperature of steam at boiler pressure in degrees Fahrenheit.

Types. There are various styles of feed water heaters on the market which may all be classed under two types—the **open-type** feed water heater, so called because the heating steam mixes directly with the water to be heated; and the **closed type**, so-called because the water passes through tubes surrounded by the heating steam. **Open feed water heaters are not used in American Marine practice**, but they are used to some extent in European Marine practice. The closed-type heater may be further divided into two general classes, one class having heating surface made up of **coils of brass or copper tubing**, and the other class having heating surface made up of **straight brass or copper tubes.**

Vertical Unit Coil feed water heater, **cast iron shell**, is shown in Fig. 1. **Vertical Unit Coil** feed water heater, **steel shell**, is shown in Fig. 2. **Bow-tube heater**, steel shell, the tubes being slightly bowed to take up expansion strains, is shown in Fig. 3. **Straight-tube** heater, steel shell, one tube sheet and header floating to allow expansion, is shown in Fig. 4.

Materials. Shells are made of steel plate or cast iron. **Heating surfaces** of copper or brass tubing. **Water manifolds** of cast iron or composition. Steel shells are preferable from standpoint of weight, but cast iron resists **corrosion** much better. In general, any heater should be designed so as to allow free expansion and contraction of tubes without danger of causing leaks.

Accessories necessary to good operation are safety valve, vacuum breaker, pressure gauge, water gauge glass and fittings, brass case thermometer. It is also very desirable to place back pressure valve on steam line, as shown in Figs. 5 and 6. The purpose of the back pressure valve is to maintain constant exhaust steam pressure in the heater.

Methods of Installation. Fig. 5 shows the method of installing direct flow feed water heater, (thoroughfare type). Heater should be placed in position convenient to feed pump and filter box with bottom of heater above surface of water in filter box. Feed line on delivery side of main feed pump should

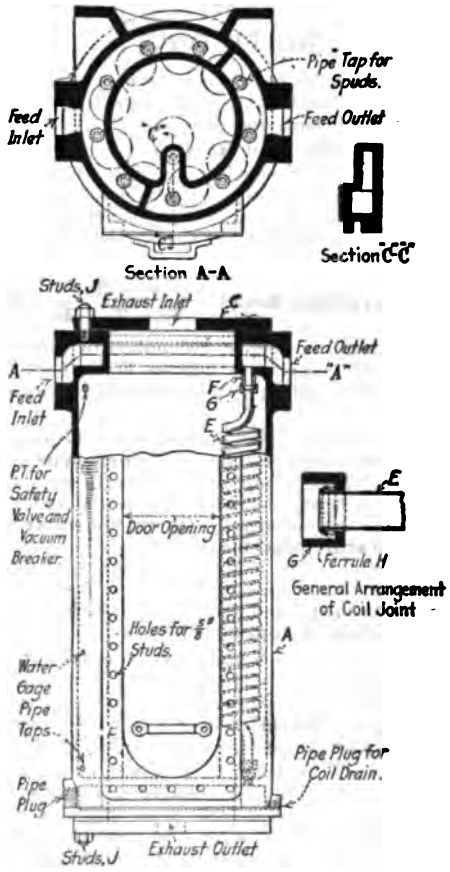


FIG. 1.—Feed-water heater, vertical unit coil, C.I. shell.

be connected to feed inlet of heater and pipe run from feed outlet of heater to boiler. By-pass heater as indicated. Auxiliary exhaust main is connected to top of heater and fitted with stop valve. Bottom of heater should be connected with main engine condenser or atmospheric connection with an automatic back pressure valve in this line. Condenser connection should

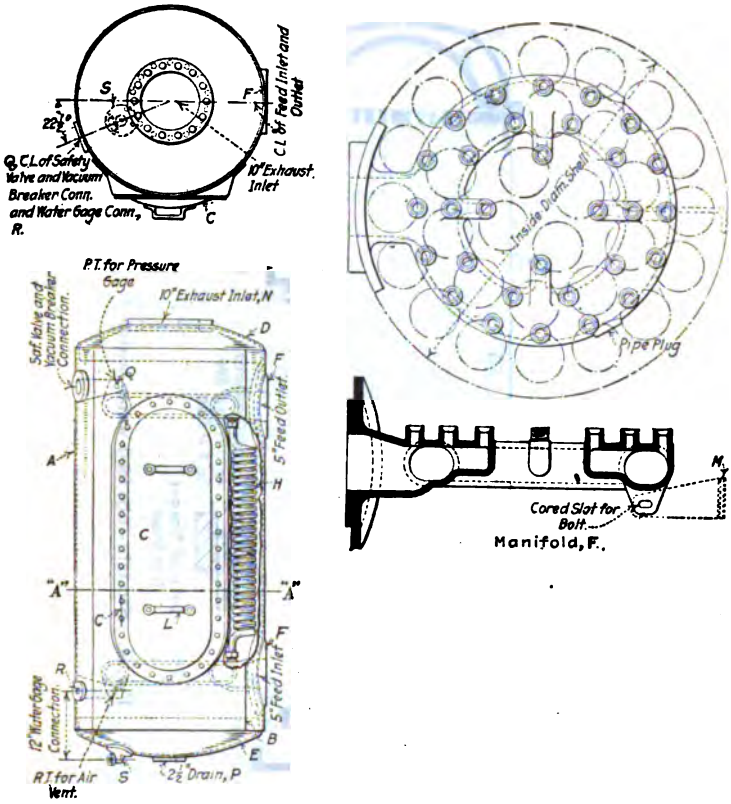


FIG. 2.—Feed-water heater, vertical unit coil, steel shell.

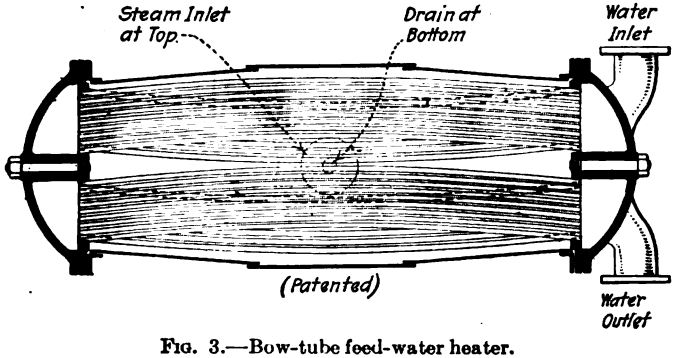


FIG. 3.—Bow-tube feed-water heater.

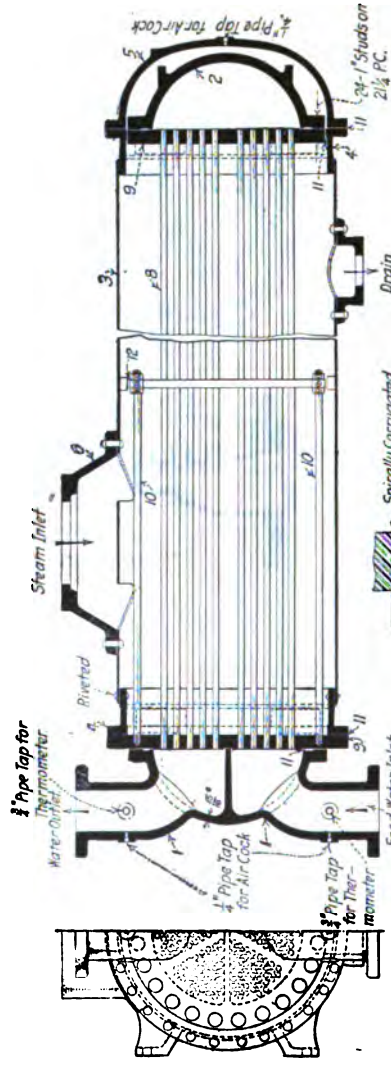

¾" Pipe Tap for Air Cock
24-1" Studs on 21¼ P.C.
Drain
Spirally Corrugated Agitator No. 22 B.W.G. ⅝" Wide Brass.
Detail of Tube and Tube Sheet.
Steam Inlet
Riveted
¾" Pipe Tap for Thermometer
Water Outlet
¾" Pipe Tap for Air Cock
¾" Pipe Tap for Thermometer
Feed Water Inlet.

FIG. 4.—Straight-tube feed-water heater

be fitted with stop valve. Pipe should be run from heater drain connection to filter box with stop valve or steam trap in line. This line is for purpose of removing condensation from heater shell.

Fig. 6 shows the **dead end** method of installing feed water heater. Heater is placed in position convenient to feed pump and filter box. Bottom of heater above surface of water in filter box. Connect feed line on delivery side of feed pump to feed inlet of heater and run pipe from feed outlet to

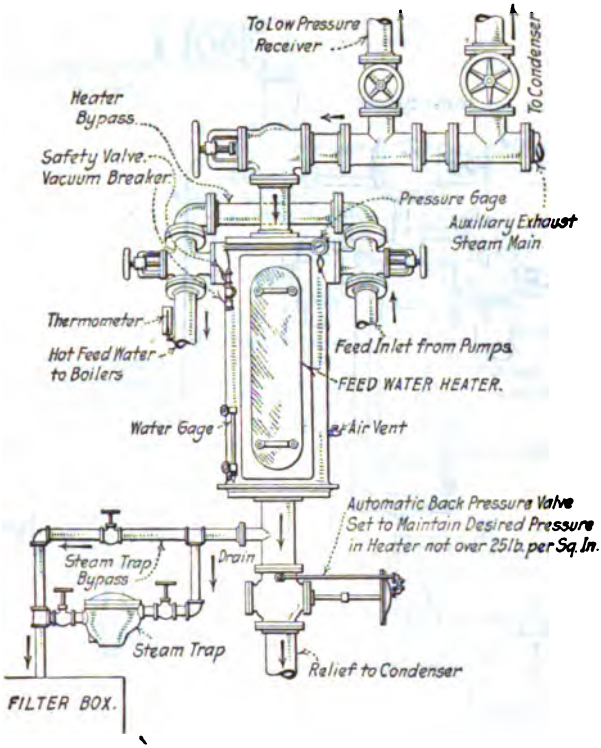


FIG. 5.—Direct flow installation of two-pass, closed type feed-water heater.

boilers. By-pass heater as indicated. Run branch line from auxiliary exhaust main to top of heater and fit same with stop valve. Install automatic back pressure valve in auxiliary exhaust main at point between branch line connection to heater and the condenser. Run pipe from bottom drain connection of heater to filter box with steam trap in line. This line is for purpose of removing condensation from shell.

Size Feed Water Heater Required. No hard and fast rule can be specified. The advisable size depends upon hot-well temperature, back pressure

of exhaust steam used for heating, amount of water to be heated, and amount of exhaust steam available for heating. It is apparent that feed water heaters cannot be rated on horsepower but must be separately computed for every case. In general, closed-type feed water heaters as used on shipboard are designed to heat the water to within 10 or 15 deg. F. of the temperature of the heating steam, provided there is sufficient steam available.

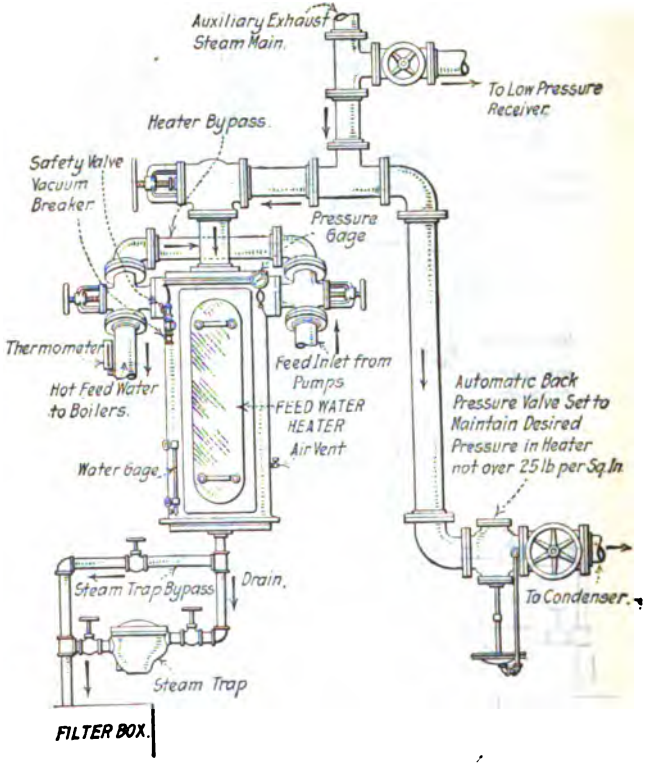


FIG. 6.—Dead end installation of two-pass, closed type feed-water heater.

The friction drop through a heater is generally limited by specifications to 12 or 15 lbs. per sq. in. Good modern marine practice requires about 2.5 sq. ft. of heating surface per 1000 lbs. per hr. of feed water.

Design of Closed Feed Water Heater. The following was abstracted by Mr. M. C. Stuart from his article in the Jour. A.S.N.E., Aug., 1917:

"A closed feed water heater is designed to heat a certain quantity of water through a given temperature range with a given steam temperature and with a limited loss of friction head in water flowing through the heater. In order

to produce these results in the most economical manner it is necessary to determine not only the amount of heating surface, but also the correct arrangement as regards tube length and number of tubes.

"In the design of heaters the following formulæ are used:

Notation:

$M.T.D.$ = mean temperature difference between water and steam.
K = heat transfer coefficient; B.t.u. transferred per hour, per sq. ft. heating surface, per degree mean temperature difference.
V = velocity, feet per second.
W = pounds of water per hour through heater.
S = total outside tube heating surface in heater, sq. ft.
N = number of unit tubes in heater.
L = length of unit tube, feet = total length of water passage through tubes surrounded by heating steam.
H = total water friction drop through heater, pounds per sq. in.
t_s = temperature of steam in heater, degrees Fahrenheit.
t_i = inlet water temperature, degrees Fahrenheit.
t_o = outlet water temperature, degrees Fahrenheit.
a, n = heat transfer coefficient constants determined from test of heater.
b, m = friction constants determined from test of heater or other sources.
C_1 = constant depending on inside cross-sectional area of tube used.
C_2 = constant depending on outside perimeter of tube used.
d_i = inside diameter of tube in inches.
d_o = outside diameter of tubes in inches

$$M.T.D. = \frac{t_o - t_i}{\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right)} \quad (1)$$

$$K = \frac{W}{S}\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right) \quad (2)$$

$$K = aV^n \quad (3)$$

$$H = bLV^m \quad (4)$$

$$L = \frac{H}{bV^m} \quad (5)$$

$$W = c_1VN \quad (6)$$

$$S = c_2LN \quad (7)$$

$$C_1 = \frac{\pi d_i^2 \times 62.3 \times 3600}{4 \times 144} \quad (8)$$

$$C_2 = \frac{\pi \times d_o}{12} \quad (9)$$

$$V^{1+m-n} = \frac{ac_2H}{bc_1\log_e\left(\frac{t_s - t_i}{t_s - t_o}\right)} \quad (10)$$

"The values of the constants for ⅝-in. standard tubes, No. 16 B.W.G. as determined from several tests of heaters are as follows:

Constant.....	n	m	a	b	c_1	c_2
Values.....	0.5 to 0.6	1.7 to 1.9	300 to 330	0.007 to 0.011	305.8	0.1636

"In designing a heater it is first necessary to determine the correct water velocity by substituting in formula 10 the values of the six constants and the values of the temperatures of water and steam and the friction loss. The tube length is then found from equation 5, the number of tubes is found from equation 6, and the total surface from equation 7.

"As an alternate method, a value of velocity V may be assumed, the value of K found from equation 3, and the surface found from equation 2. The number of tubes is then found from equation 6 and the tube length from equation 7. Finally the friction loss is determined from equation 4, and if this is too high, a lower velocity must be assumed and the computations repeated.

"The constants a and n must be determined from results of tests or from operation of a heater. The constants b and m are fixed by the laws of friction loss in tubes and may be determined from test results. The constants c_1 and c_2 depend only upon the inside and outside diameters of the tube."

Heat transfer rates or coefficients **vary with** the condition of the tubes. A slight film of scale, oil, or dirt on the tubes will cut down heat transfer rates, sometimes to considerable extent, depending on the nature and thickness of the film. The size of tube used and the design of tube layout also effect heat transfer rates.

The values of a and n given in the above table are apparently for fairly clean tubes and for good operation of the heater, the tubes being plain $\frac{5}{8}$ in. tubes with no retarders, or whirler strips therein.

Heat transfer rates **may be increased** considerably by inserting strips of twisted flat metal ribbon called whirler strips, **retarders**, or agitators, in the tubes, the effect being to increase water velocities and bring all particles of water into more intimate contact with the heating surface. The increase of heat transfer rate is, of course, accompanied by increased friction drop through the heater. Several patented forms of retarder are on the market.

Operation of Feed Water Heaters. Shell should be kept thoroughly drained of condensation. The water gage glass on the steam side of a heater is merely an indicator. No water should appear therein. Back pressure valve should be used on steam line to insure steam pressure in heater shell. Uniform operation of feed pump is essential to high average feed temperature. The steam space of heater must be kept vented of air especially for induction type installation. For this purpose tap hole in side of heater shell about 12 in. above bottom drain connection and keep air bleeding freely from same at all times. An automatic air vent is desirable.

From "Heat Transmission and Tube Length" by Leo Loeb in the Jour. A.S.N.E., May, 1915, "Water will absorb nearly 2 per cent by volume of air and this air will be carried over with the steam. At a given pressure and temperature of saturation air has a greater density than the steam with which it is associated, and it will gradually accumulate above the condensation in the bottom of the heater. Thus part of the surface is rendered inactive and the film resistance is increased to an extent that heat transfer is appreciably reduced. This blanketing effect will be greater with a film heater with horizontal tubes than if the tubes be vertical."

In a test cited in this same paper, which test was run without relieving any air from the steam space, "the rate of heat transmission decreased steadily from 661 B. t. u. per sq. ft. per deg. average temp. difference to 558 B.t.u., a net loss of 15.6 per cent. At the same time the hot feed temperature fell from 195.1 deg. to 181.8 deg."

OIL COOLERS

BY

JOSEPH PRICE

Marine Service. For turbine driven ships the lubrication of turbine bearing and reduction gears requires the positive cooling of the lubricating oil to dissipate the heat of friction and thereby maintain constant temperature and viscosity of oil, the same oil being used over and over again. This is accomplished by oil coolers using sea water as the cooling medium.

Duty and Size. For each design of turbine and reduction gear there is a best grade of oil for lubrication. One design might require a much more viscous oil than would another design. After the proper viscosity of the oil is decided upon, it is necessary to select an oil cooler which will **maintain an oil temperature corresponding to the desired viscosity when circulating an amount of oil necessary to proper lubrication.** Increase of oil temperature rapidly decreases viscosity.

Design. The oil cooler must be **designed** to remove all the heat generated in the gears and bearings by friction so as to prevent gradual increase of oil temperature as it is used over and over again in bearings and gears. Different designs of turbines and gears will differ as to the amount of heat generated by friction in the bearings and gears. For the average commercial marine turbine installation with **double reduction gears,** it is usually sufficient to assume that the heat generated in gears and bearings will be 7 per cent of the heat value of the total power of the turbine, 1 H.P. being equivalent to 2545 B.t.u. per hour. Undoubtedly the heat losses for average commercial turbines will be less than 7 per cent after bearings and gears have been in operation for a time.

For **single reduction gear** installations, it is sufficient to assume that 6 per cent of the total power will be lost as heat in bearings and gears.

Example: If the total power of a marine turbine of average design, equipped with double reduction gears, is 3000 H.P., then the equivalent heat value of the power is 3000 × 2545 = 7,635,000 B.t.u. per hour and 7 per cent of the total power is 534,450 B.t.u. per hour. The oil cooler should, therefore, be designed to absorb 534,450 B.t.u. per hour, or 8900 B.t.u. per minute from the oil when the oil is of the grade and at the temperature corresponding to the proper viscosity for lubricating the machinery under consideration, and when the amount of oil circulated is sufficient for proper lubrication.

Assume that the proper oil has a viscosity of about 150 Saybolt seconds at 130 deg. F. and that it is desired to maintain this average viscosity; and assume that an oil circulation of 175 gal. per min. or 1312 lbs. per min. is necessary. In order to absorb 8900 B.t.u. per min. the temperature drop of the oil in the cooler, assuming the specific heat of oil as 0.5, will have to be 8900 ÷ (1312 × 0.5) = 13.6 deg. F.

The cooler should, therefore, be designed to cool 175 gal. per min. of the specified oil from 137 deg. F. to 123 deg. F. when supplied with 85 deg. F. cooling water, the amount of cooling water being between 2 and 3 times the amount of oil. For colder circulating water the amount of water necessary will be much less, and circulating pumps need not be designed for continuous service of handling the larger amounts of 85 deg. F. water.

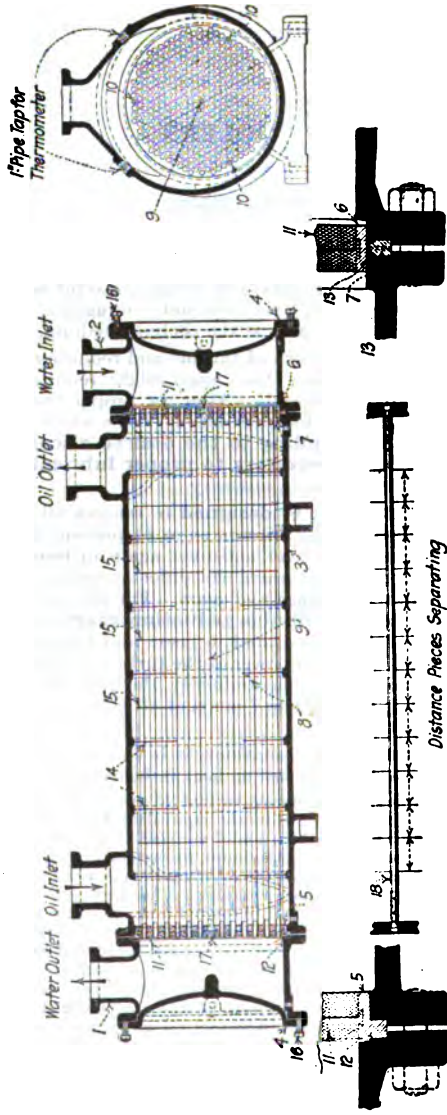


FIG. 1.—Straight-tube oil cooler with annular baffles.

The subject of heat transfer rates for oil coolers is exceedingly complex and it is impossible to use any set heat transfer rates in designing the coolers. For instance, if a given cooler will cool 100 gal. per min. of a certain grade of oil having a viscosity of 275 Saybolt seconds at 100 deg. F., the same cooler will cool only about five-eighths as much, or 63 gal. per min. of another grade of oil having a viscosity of about 800 Saybolt seconds at 100 deg. F., through the same temperature range and with the same water conditions. The heat transfer rate depends upon the viscosity of the oil and different designs of coolers will vary as regards heat transfer efficiency.

In the design of an oil cooler the following factors should be considered: (*a*) The velocity and friction drop of the oil, (*b*) the velocity and friction drop of the water, (*c*) the heat transfer coefficient. Increasing the velocity has the effect of increasing the friction drop of the oil and water and also increasing the heat transfer coefficient. Inasmuch as operation with a high heat transfer coefficient reduces the amount of surface required for a desired cooling, the design of any type of cooler should be made with as high velocities of oil and water as possible without causing excessive friction drop of the oil and water. The heat transfer coefficients found in the usual designs of oil coolers vary from 50 to 100 B.t.u. per hour per sq. ft. of surface per deg. F. mean temperature difference between the oil and the water. On account of the complex nature of the transfer of heat with various arrangements of surface and with various oils, the exact design of a cooler of a given type must be based upon test or performance results of a cooler of a similar type using an oil of similar viscosity.

Space and Weights. For the average commercial marine turbine installation of 3,000 H.P. the oil cooler will be about 14 in. diam. × 7 ft. long overall, depending entirely upon the efficiency of the design. The approximate dry weight of the cooler having cast iron shell will be 1,600 lbs.

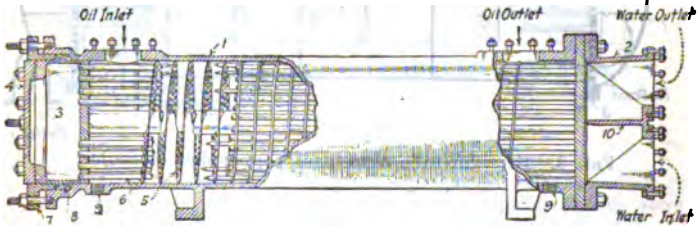


FIG. 2.—Straight-tube oil cooler with helical baffle.

Commercial Types. Fig. 1 shows the horizontal **straight-tube** cast iron shell type **with annular baffles,** circulating water passing through tubes and oil outside tubes.

Fig. 2 shows the **straight-tube** cast iron shell type **with helical baffle,** circulating water passing through tubes and oil outside tubes.

Fig. 3 shows the **straight-tube copper shell** type. In this instance the oil passes through the tubes and the circulating water outside the tubes.

Method of Installation. A complete lubricating oil system showing oil cooler location for marine turbine installation is shown on page 670. As the oil cooler is a very important unit (a geared turbine ship could not run

without it), it should be installed in duplicate, one cooler being available as a spare.

Operation of Oil Coolers. The hot lubricating oil is pumped through the coolers and thence distributed through the lubricating oil system to the bearings and gears, after which it is returned to the oil pump suction. Thermometers should be placed in the oil inlet and outlet of the cooler in order to keep careful check on the service and also to determine when the cooler needs cleaning.

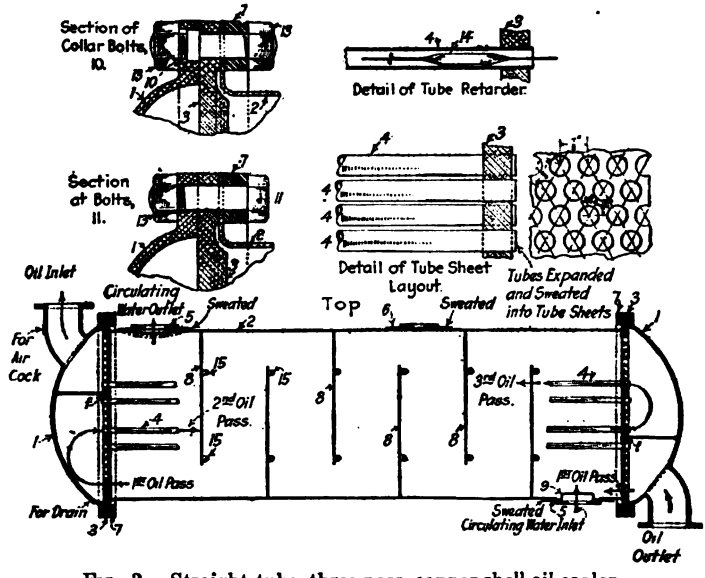


FIG. 3.—Straight-tube, three pass, copper shell oil cooler.

CENTRIFUGAL PUMPS

BY

M. L. KATZENSTEIN AND MAX SPILLMAN

Characteristic Curves

Centrifugal pumps are designed generally to deliver a given quantity of water against a stated total head at a given number of revolutions per minute. When working under these conditions, the ratio of useful work done to the power input measured at the shaft is called the efficiency of the pump A pump designed for a certain condition is capable also of working over a range of capacities and heads very much different without losing much efficiency. Such variation in working conditions may be obtained without change of speed, but this is within rather narrow range of limits. Greater variations

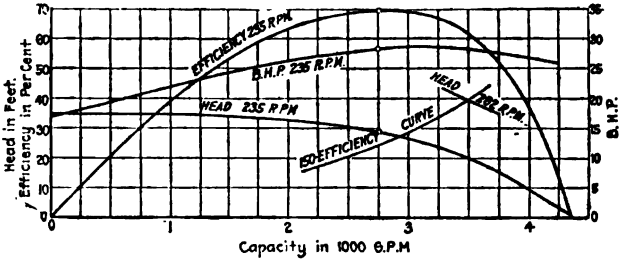


FIG. 1.

will be obtained by increasing or decreasing speeds. Characteristic curves are generally plotted for constant speeds and are useful as they show the economical range of capacities and heads.

Fig. 1 shows characteristic curve of an engine driven circulating pump having a most economical capacity of 2750 gal. per min. against 29 ft. head and requiring 28½ B.H.P. to drive it.

Fig. 2 shows the results of a test of a steam turbine driven circulating pump. In order to obtain best steam economies on the turbine, these pumps are often run at speeds which do not give the best pump efficiencies. This curve shows maximum efficiency at 4520 gal. per min. against a head of 60 ft., but the pump would operate over a range of heads from 20 to 30 ft. with efficiencies from 52 to 70 per cent.

Fig. 3 shows the results obtained with a boiler feed pump having three impellers in series with a most economical capacity between 900 and 1250 gal. per min., at constant speed.

Variation in speed causes variation in capacity as well as in head. A simple formula that will hold for points of equal efficiency at different speeds, is as follows:

$$\frac{N}{N_1} = \frac{Q}{Q_1} = \frac{\sqrt{H}}{\sqrt{H_1}} = \frac{\sqrt[3]{S.H.P.}}{\sqrt[3]{S.H.P._1}}$$

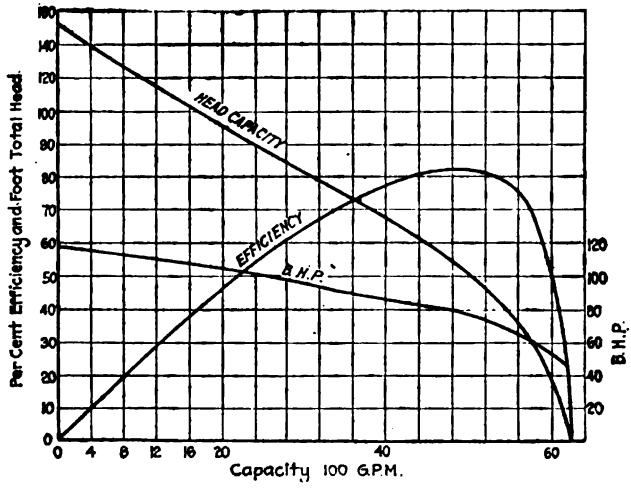


FIG. 2.

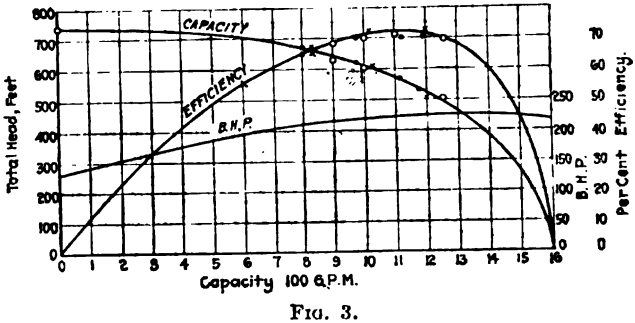


FIG. 3.

Where N = revolutions per min.; Q = capacities in gal. per min.; H = head in ft.; $S.H.P.$ = shaft horsepower.

It will appear, therefore, from Fig. 1, a capacity of 2925 gal. per min. is obtained at 28 ft. head with 235 R.P.M. and 69.7 per cent efficiency. At 282 R.P.M. a capacity of 3300 gal. per min. against 40 ft. head would result in the same efficiency.

In order to pump water efficiently to a given height it is necessary that the **rim speed of the impeller** be kept within certain limits, depending upon the head against which it is desired to deliver the water. For engine driven pumps of the slow speed type a circumferential speed at the tip of the impeller vanes will vary from 1.0 to 1.2 $\times \sqrt{2gh}$. For steam turbine driven high speed circulating pumps from 1.15 to 1.16 $\times \sqrt{2gh}$.

The **capacity delivered** depends on the area of the inside water passage from suction passage to impeller and thence through to the volute. Velocity (in ft. per sec.) of flow through the suction passage from 0.13 to 0.15 $\times \sqrt{2gh}$; in impellers from 0.13 to 0.17 $\times \sqrt{2gh}$; in the casing from 0.30 to 0.35 $\times \sqrt{2gh}$.

The **total head** as observed on tests is the algebraic sum of the reading of a gage near the suction inlet of the pump plus the reading of a gage in the discharge of the pump plus the vertical distance between the two gages. Generally no provision is made aboard ship to measure capacities, but this could be done in a simple way by installing Venturi meters in the pipe lines.

Construction

The **essential parts** of a centrifugal pump are (1) suction head, (2) impeller or impellers, (3) discharge chamber, (4) shaft and (5) stuffing box. The designs of these parts vary considerably with the pump service and with the available space for installation. Fig. 4 illustrates the combination of these parts in their simplest form.

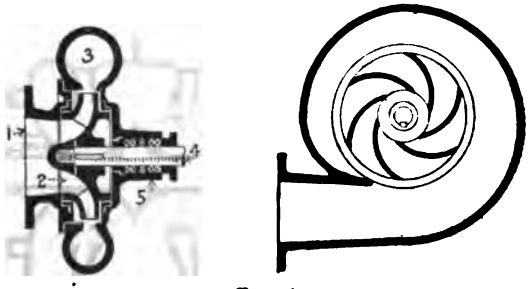


FIG. 4.

Impellers. Figs. 5 and 6 show single suction and double suction types respectively. The rotation of these impellers establishes, by the components of the centrifugal forces thus set up, the flow of water. The **primary object in the design** of impellers is to obtain continuous flow free from shock. At certain definite capacities and speeds the impeller can be designed to create this condition, *i.e.*, given a capacity and a head against which the pump is to deliver, the curvature and angles of the impeller vanes are made to suit that condition. For other than design conditions of capacity and head, the efficiency must necessarily be lower. The impeller consists of two side plates with vanes designed to conditions as stated above. The **double inlet type** is preferred for high speed small diameter impellers. The **single inlet type** is used for low speed reciprocating engine drive. Generally speaking, for a given capacity and head the higher the speed the smaller will

be the impeller diameter. As an extreme form, the screw impeller, Fig. 7, when inclosed by properly designed casing, has all essential characteristics of the centrifugal pump.

FIG. 5.

FIG. 6.

FIG. 7.

The suction head, in its simplest form, is a radial pipe leading the water straight into the eye of the impeller. The best practice demands a design in which the pump can be dismantled without breaking the suction pipe connection. Not only must care be exercised in the design of the suction head itself, but equally important is the approach of the suction pipe to the pump. With a poorly constructed suction line, having high spots, the pump will lose

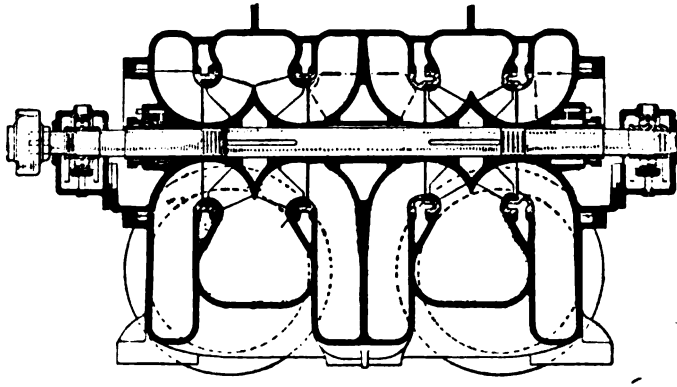


FIG. 8.

the water due to air leaks and air binding. Sharp elbows close to the pump nozzles should be avoided as they cause loss both in capacity and in efficiency due to eddy currents set up which are carried through to the impeller opening. An elbow on the suction nozzle of a double suction pump, placed in a direction parallel to the shaft, will divert water mostly to one side of the impeller and consequently there will be a reduction in capacity and efficiency; such an arrangement should be avoided.

Where a pump handles hot water, as in the case of a boiler feed pump, it should be located below the level of water supply, so that there will be created a net effective head at the suction to prevent vapor binding. The difference in level that should be maintained between the water supply and the center of the pump suction depends on the design. In general eight feet net at the pump nozzle is sufficient for water of 210 deg. F. If there are bends in the pipe line to the pump suction or if it is long, the head should be increased to overcome the losses in the suction pipe. Where pumps operate at exceptionally high speeds or deliver against very high heads per stage, the design of pump might require that this suction head be as high as fifteen feet net. The direct result of too little effective head at the suction is loss in capacity and a possibility of vapor binding. Excessively high rotative speed should not be used and ample suction head should be provided for safe and reliable operation of pumps when used for boiler feed service.

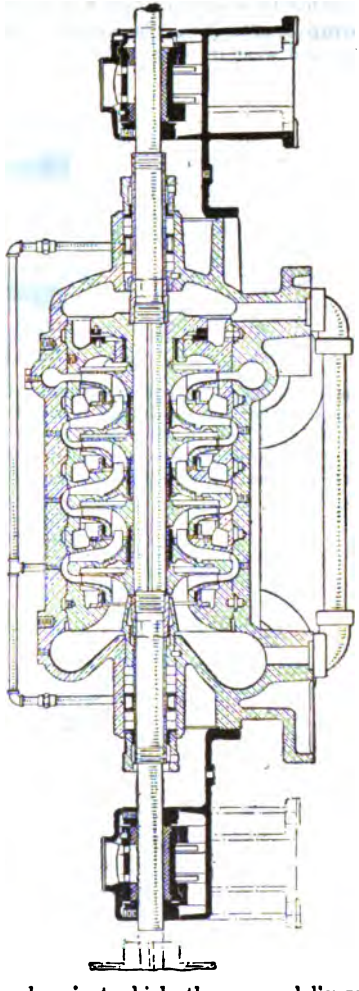


FIG. 9.

Casings. The usual form of casing is a **volute chamber** of cross section made suitable to conduct the water from the circular orifice of the impeller and to the discharge opening. (See Fig. 4.) Where large capacities are to be handled, an arrangement of two volutes, side by side, can be combined in a single casing, Fig. 8. Such a pump is called a **bi-rotor** and as a rule this combination will give minimum weight for capacity. In this arrangement the water enters both impellers at the same pressure and is discharged at the same pressure, *i.e.*, the impellers are operating in parallel. Where the head against which the pump delivers is very high, it becomes necessary to arrange the impellers in series, Fig. 9, instead of in parallel, in which case the design cannot be well accomplished with the volute type of casing. In this case water enters, through the suction head, the first impeller only and on leaving the impeller is dis-

charged through a diffusion ring for reducing the velocity and redirecting it to the succeeding stage. In this way the pressure is brought up by equal increments to a velocity sufficient to overcome the discharge head.

Pump shafts should be covered with brass sleeves in order to avoid corrosion and wear in the stuffing boxes. All centrifugal pumps (except those

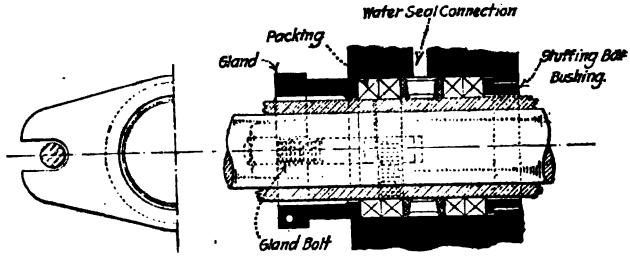


FIG. 10.

of the double-suction type) will develop an axial thrust which must be counterbalanced either by a suitable thrust bearing at the end of the shaft of the multicollar or ball bearing type or some form of hydraulic counterbalance. Figs. 8 and 9 show these arrangements and are more fully discussed in the section on centrifugal boiler feed pumps, p. 1127.

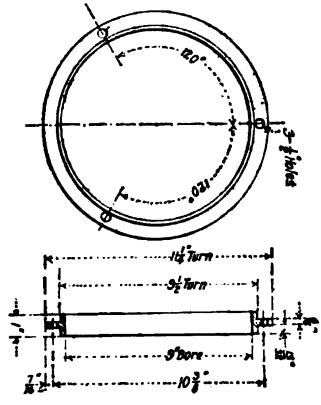
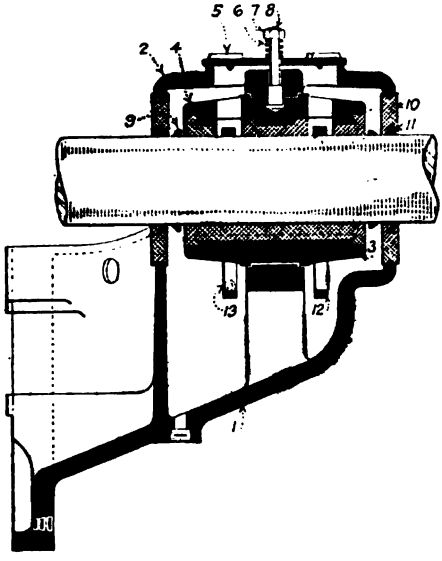


FIG. 11.

Stuffing boxes, Fig. 10, prevent air leakage into the pump and water leakage from the pump. Four or six rings of graphite packing should be used with a seal cage between them as indicated. Sealing of the stuffing boxes is essential to exclude the air, providing the pumps are operating above the level of water supply. If located so that there is a positive head on the suction side, no water sealing is required. Stuffing boxes must be kept sufficiently loose to allow a slight seepage of water through the packing. On ship-

board this seepage must be reduced to a minimum, but should never be stopped entirely as rapid wear of the sleeves will result.

Bushing rings, Fig. 11, are placed between the impeller inlet and the casing to form a leakage joint. These rings are subject to much wear;



1. Bearing Body
2. Bearing Cap
3. Lower Bushing
4. Upper Bushing
5. Oil Hole Cover
6. Oil Hole Cover Spring
7. Nut
8. Dowel Pin
9. Oil Thrower Ring
10. Oil Retainer
11. Shaft Wiper
12. Oil Ring
13. Oil Gauge Connection

FIG. 12.

they are machined with running clearance between the casing and the impeller varying from 0.010 in. to 0.015 in. These rings must be replaced when, through wear, the clearance becomes excessive.

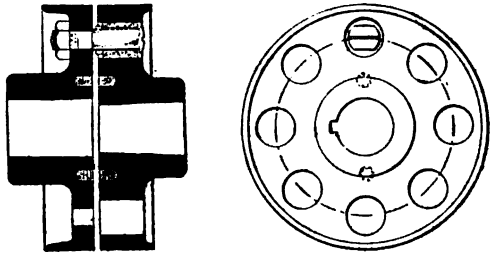


FIG. 13.

Bearings are of various forms. **Ring oil babbitted** bearings, Fig. 12, are employed extensively; they consist of split bearing bushings and bearing body of sufficient depth to hold an ample supply of oil. **Ball and roller bearings**

the impeller. The impeller is of the overhung type. A grease lubricated bearing is provided in the discharge head.

Electric motor driven circulating pumps for small capacities are of the single suction type shown in Fig. 15. A flexible coupling is used in this combination to allow some axial play to the motor armature. Hydraulic balance in this type of pump is maintained by mounting a disk on the shaft of sufficient diameter to equalize the excess pressure on the discharge side of the impeller. The space in the back of the balancing disk is connected with the suction chamber.

Motor or steam turbine driven circulating pumps for medium capacities are designed generally with the casings split horizontally and impellers of the double-inlet type giving the greatest accessibility and freedom from troubles due to thrust. The horizontally split casing had both the suction and discharge nozzles in the lower half of the casing, Fig. 16. Removing the upper half of the casing and bearing cover, the whole rotor of the pump is open for inspection and can be removed, if necessary, for repairs after the coupling bolts have been taken out. The advantages of this design are obvious.

There have been developed in this type of unit, arrangements in which the pump and turbine rotors are mounted on the same shaft, omitting the usual coupling between the prime mover and the pump, Figs. 17 and 18. This arrangement eliminates the possibility of trouble due to misalignment. Such arrangements allow the use of emergency governors only and their largest application is for pumps used for circulating in water condensers, coolers, distillers, etc.

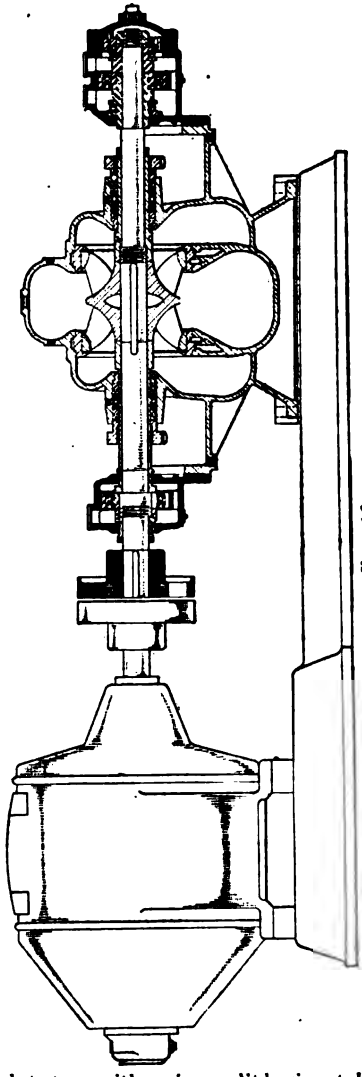


FIG. 16.

Electric motor driven or steam turbine driven pumps for large capacities are of the double suction volute type with casings split horizontally and flexible coupling between prime mover and pump, Fig. 19. Radial ball

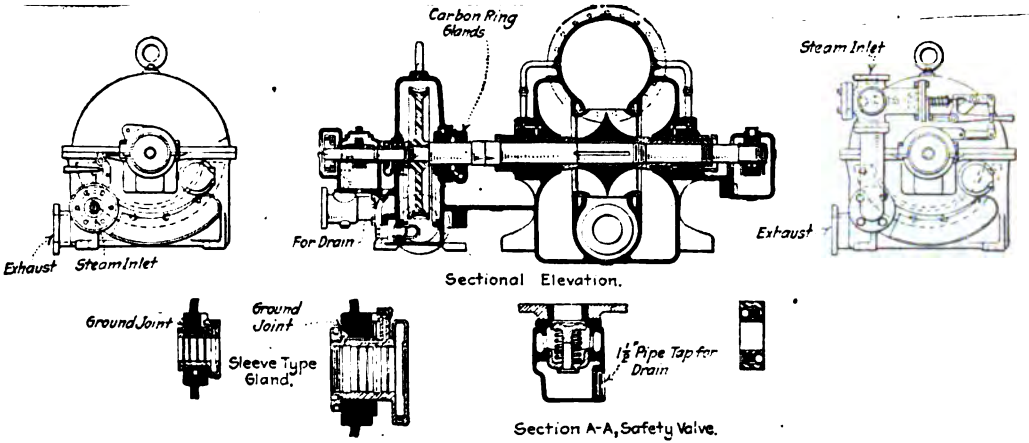

Carbon Ring Glands
Steam Inlet
Exhaust
Steam Inlet
For Drain
Exhaust
Sectional Elevation.
Ground Joint
Ground Joint
Sleeve Type Gland.
1½" Pipe Tap for Drain
Section A-A, Safety Valve.

FIG. 17.

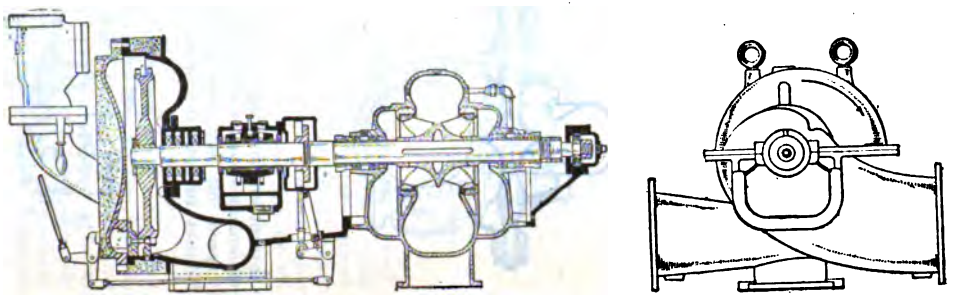


FIG. 18.

the impeller. The impeller is of the overhung type. A grease lubricated bearing is provided in the discharge head.

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Motor or steam turbine driven circulating pumps for medium capacities are designed generally with the casings split horizontally and impellers of the double-inlet type giving the greatest accessibility and freedom from troubles due to thrust. The horizontally split casing had both the suction and discharge nozzles in the lower half of the casing, Fig. 16. Removing the upper half of the casing and bearing cover, the whole rotor of the pump is open for inspection and can be removed, if necessary, for repairs after the coupling bolts have been taken out. The advantages of this design are obvious.

There have been developed in this type of unit, arrangements in which the pump and turbine rotors are mounted on the same shaft, omitting the usual coupling between the prime mover and the pump, Figs. 17 and 18. This arrangement eliminates the possibility of trouble due to misalignment. Such arrangements allow the use of emergency governors only and their largest application is for pumps used for circulating in water condensers, coolers, distillers, etc.

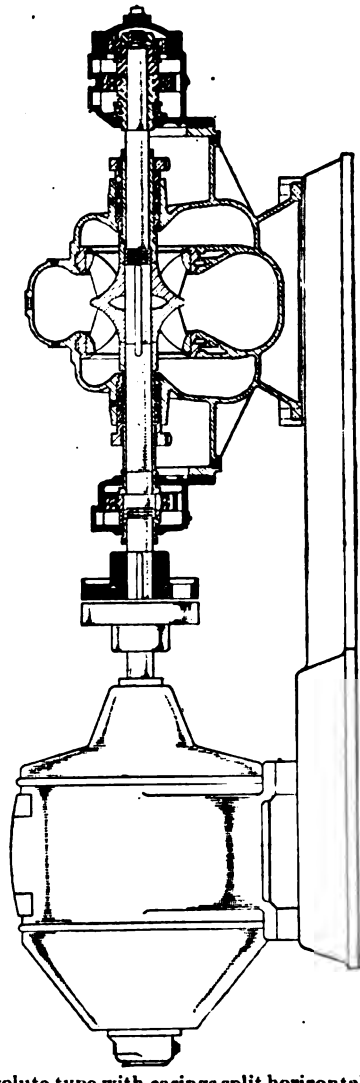


FIG. 16.

Electric motor driven or steam turbine driven pumps for large capacities are of the double suction volute type with casings split horizontally and flexible coupling between prime mover and pump, Fig. 19. Radial ball

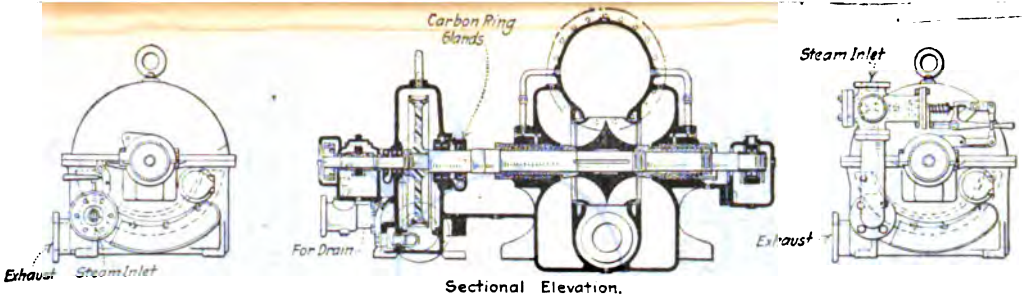

Carbon Ring Glands
Steam Inlet
Exhaust
Steam Inlet
For Drain
Exhaust
Sectional Elevation.
Ground Joint
Ground Joint
Sleeve Type Gland.
1½" Pipe Tap for Drain
Section A-A, Safety Valve.

FIG. 17.

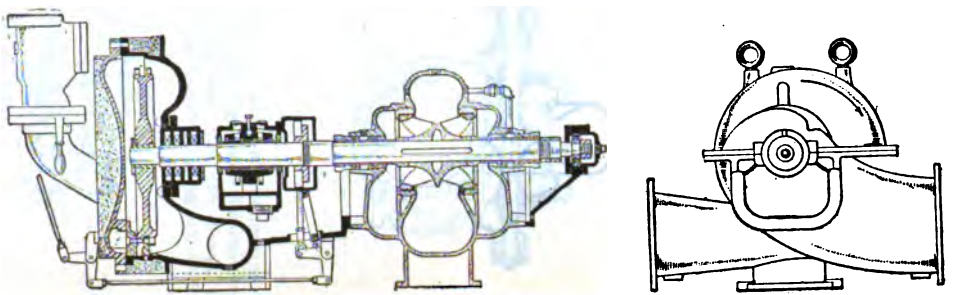


FIG. 18.

bearings are used to keep the overall dimensions of the pump a minimum. A ball thrust bearing at the outboard end locks the shaft in position. Where steam turbine drive is used, the pumps are driven directly from the shaft of the prime mover or through reduction gears. Where the volume of water to be handled is large, pumps of the **bi-rotor** type are employed with casings split in the horizontal plane. Two impellers are used, operating in parallel, *i.e.* the water is taken at the same pressure at the inlets of the impellers and the discharge is to a common chamber by the impellers at the same pressure. With this type of pump relatively high speeds can be used for capacities up to 35,000 gal. per min., making it possible to operate direct connected to steam turbines at speeds of from 1000 to 1300 R.P.M. For greater steam economy at the turbine, the reduction gear

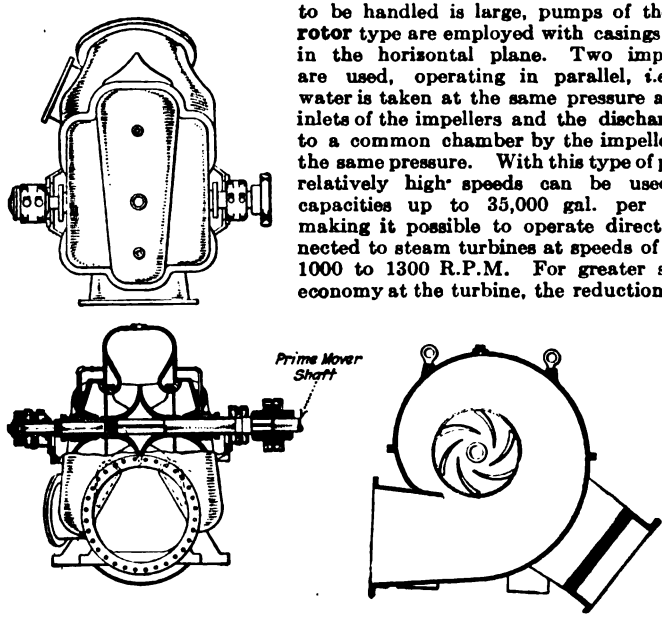


FIG. 19.

is interposed, making it possible to run the steam turbine at the most economical speed varying from 3600 to 7200 R.P.M. and the pump at a speed of, say, 800 R.P.M. thus approaching conditions of best efficiency in both the prime mover and pumping elements. There have

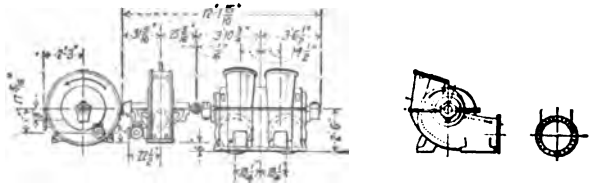


FIG. 20.

been special types of pumps developed for large capacities for special installations, having screw impellers with axial flow of water through them, but their efficiency is somewhat lower. Fig. 20 shows a special arrangement

in which two pumps, right and left hand, are mounted together in such manner as to eliminate end thrust.

Centrifugal Boiler Feed Pumps

For feeding boilers with centrifugal pumps the **multi-stage unit** is employed to develop the required pressure. Two general types are in general use; (1) one with the turbine and pump mounted on a common bedplate with flexible coupling between, the turbine and pump each provided with two bearings, or (2) as a combined set in which no common bedplate or coupling is employed and the turbine and pump casings are joined by suitable flanges. In this case three bearings are used, one on each end of the unit and one between the pump and turbine casings. Fig. 9 shows a horizontally split case centrifugal boiler feed pump having single suction impellers.

Combined unit construction is shown in Fig. 21. The shaft is continuous through the pump and turbine casing. Removal of the rotor involves dismantling both the turbine and the pump top half casings, which are split

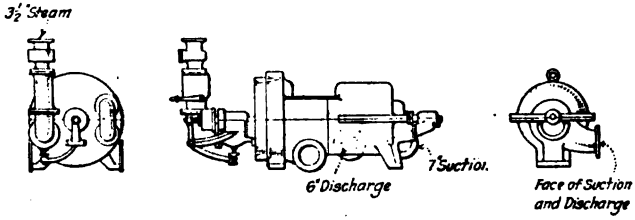


FIG. 21.

horizontally with suction and discharge connections in the bottom half of the pump. Suitable feet are provided for bolting the unit to the hull foundation and **care must be exercised in pulling down these bolts** so that the unit is not sprung out of line. To ensure this remove top casings and make all pipe connections before pulling down the bolts With the bearing caps removed it can then be determined whether misalignment is taking place while adjusting the foundation bolts. Misalignment is indicated if the shaft does not seat properly and uniformly in all of the bearings as determined by a feeler gage placed between the shaft and the bearing shells. By removing the middle bearing bushing entirely and assembling the end bearings, alignment can be further checked by measuring with suitable calipers the distance from the outer shaft diameter to the bore in the bearing bracket that holds the bearing shell. This distance must be the same all around the shaft; in a similar manner caliper between the shaft and the stuffing box bores in the turbine and pump casings The clearance between the turbine wheel buckets, the nozzle plates and the reversing buckets should be checked also. It is necessary to have more clearance on the steam inlet side of the buckets to compensate for elongation of the shaft and consequent reduction of this clearance when the unit is warmed up to working temperature.

Separate Unit Set on Common Bedplate. The general appearance of this pump and its driving turbine is shown in Fig. 22. Water should come to the pump under a head of from 10 to 12 ft. for temperatures of water of 190 to 210 deg. F. For lower temperatures the head should be about 2½ ft. less.

Operation, Repairs and Inspection. The general characteristic of this type of centrifugal pump is such that an increase in capacity is accompanied by a decrease in the head and vice versa, providing the speed is held constant. This is true until the capacity becomes only a fraction of the normal output of the pump, in which case a change of from 10 to 15 per cent either way in capacity does not affect the head. It is important, therefore, in operating these pumps at such a condition, to have sufficient excess head over the boiler pressure plus pipe friction to be able to deliver the volume required with certainty.

The pump should be opened and examined for wear at regular periods about once every three months of continuous operation. Such parts as show undue wear should be replaced. If the wear on the balancing disk rings shows more than $\frac{1}{16}$ in., the condition of the pump should be investigated. Boiler compounds, if introduced on suction side of pump, will cause deposits in the pump and hasten the wear of these parts. Graphite compounds, if used, should be introduced on the discharge side of the pump—never in the suction. These pumps should never be operated with discharge valves entirely closed, *i. e.* with the pump delivering no water. The energy necessary for driving

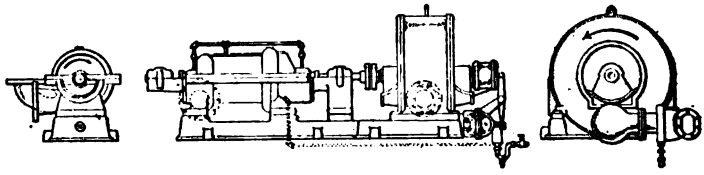


FIG. 22.

the pump under these conditions is largely dissipated in heat produced by churning the water by the impellers. This heat naturally raises the temperature of the water so that in a short time it becomes hot enough to vaporize and allow the pump to become steam or vapor bound, and to eventually seize. A bypass should be arranged somewhere in the pipe lines so that when it becomes necessary to operate at low discharge capacity, the bypass can be opened and prevent this difficulty.

Regulating devices are employed with turbine driven centrifugal feed pumps for safe and economical operation. With the steam turbine it is usual to have a **speed governor** on the end of the turbine shaft which will maintain a predetermined speed irrespective of the load. This usually is of the centrifugal type, having two counterweights on either side of the shaft and held in position by suitable knife edges and a central spring. (See Turbines, Sec. 5.)

An **emergency speed device** is provided which will close an emergency valve and thus shut the unit down entirely in case of the failure of the speed governor to operate. This overspeed or emergency governor is also discussed under Turbines, Sec. 5.

A machine arranged with the two above governing devices is intended for constant speed operation. For boiler feed work, however, it is desirable to maintain constant pressure for varying capacities demanded of the pump. This pressure cannot be maintained at constant speed, so that it has been found advisable to have a third regulator which operates on the governor valv or through a separate valve incorporated in the pressure regulator to adjust the flow of steam to the turbine and therefore control the speed of the turbine

and pump to give a constant pumping pressure irrespective of the capacity delivered. This **pressure governor** consists of a differential piston, on the two sides of which the steam and water pressure act, and is designed of such proportions that when the pump is delivering against the required pressure with a given steam pressure, the steam valve is held in a balanced position. A spring is attached so that a certain tension or compression will require a given amount of excess water pressure before it can act to close the steam valve. This excess pressure is about 10 to 15 per cent of the normal pump pressure. When the capacity drawn from the pump is decreased, thus raising the pressure, it acts on the regulator to throttle the steam and thus decrease the pump speed until stable conditions obtain. The reverse action takes place

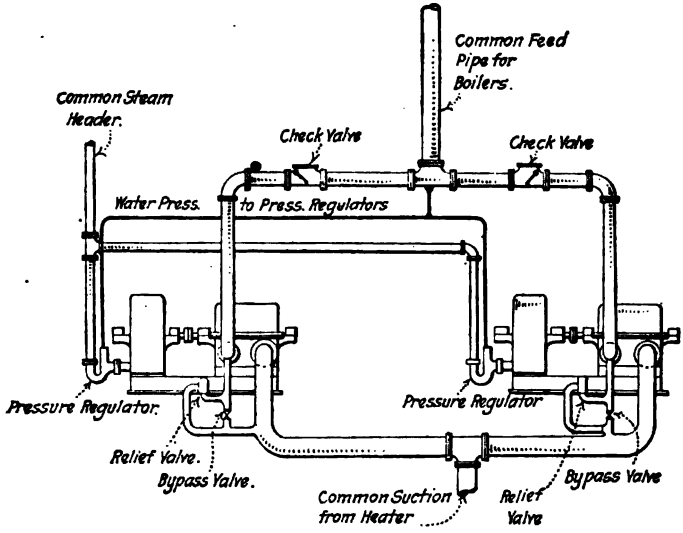


FIG. 23.

when a greater amount of water is drawn from the pump. This method of regulation insures the best steam economy due to using just enough to maintain the capacity and pressure desired. For pressure governors, see also Sec. 11.

A relief valve in the discharge line set at 10 per cent in excess of working pressure will function the same as a pressure regulator, but is wasteful of steam. However, as a safety device for the pressure regulator, the relief valve is a desirable feature in case the pressure regulator does not function properly and as such its use is to be recommended.

Installation and Operation of Centrifugal Feed Pumps in Parallel. Fig. 23 shows an arrangement of operating centrifugal boiler feed pumps in parallel. Where two pumps are discharging into the same feed line, correct and equal adjustment is required on all regulating devices on pumps and turbines. The relief valves should be set for 115 per cent of the water pressure at full capacity. All relief valves should be set as nearly alike as possible.

Pressure regulator should be set to operate when the pumps are delivering full capacity. Hand regulated valves on the boiler should be wide open under this condition. These valves should be manipulated so that they will create an artificial head on the pump when operating at less than full capacity to maintain constant working pressure at the pumps under all conditions.

When in port and small capacity is required, the bypass valve on the pump should be opened to prevent vapor binding and to allow sufficient discharge to permit correct operation of the regulators. Speed governors should be set so that they will operate above the predetermined limit of the other regulators, so that in case one or more of the pumps operating in parallel delivers less water than it should, the unloading of this pump will allow the turbine to speed up to counteract the cause of the falling off in capacity. This condition requires all turbines to have an equal number of jets. Speed governors set to work within the range of the other regulating devices will cause an unbalancing in the even distribution of the work among the different pumps. These governors are useful only as overspeed devices.

RECIPROCATING PUMPS

BY

MARTIN L. KATZENSTEIN

Marine pumping machinery differs in no essential from that used in similar service for shore power stations, but in many cases the lack of space and the necessity for limiting weights have caused the development of designs which are known as Marine Pumps.

The same principles of design as apply to pumping machinery for similar purposes on land are followed for naval and merchant marine vessels. With the many changes in the designs and types of ship propelling engines during the last twenty years the pump industry has kept pace, bringing out designs and developing types of apparatus to meet the demands for ever-increasing efficiency and reliability. Many ships' engines and fire-rooms today are in arrangement quite different from those of ten years ago, and pumps have played no inconsiderable part in bringing about the changes that have made it possible to operate vessels with the low fuel consumption which is now the rule.

The manufacture of pumps is well standardised and except where designs are made for special purposes, the details of construction should be left to reliable manufacturers. With information supplied as to the service to be performed, reliable pump makers will supply the designs and will provide amply to take care of the stresses incident to operation. Some makes of pumps are superior to others; they will accomplish the required results with considerably less weight, will occupy less space, and will be found more dependable in operation.

An equipment of pumping machinery poorly selected can easily mar the performance of an otherwise well arranged engine room. It has been found necessary many times to replace with proper equipment badly designed pumps selected originally on account of lower first cost, and the change has been expensive for not only must new equipment be purchased, but in most cases, elaborate changes in piping must be made. As in all purchases the best is the cheapest. Only manufacturers of established reputation should be consulted when specifications are in course of preparation, and no deviation from specified designs and materials should be permitted during construction without good reason.

TYPES OF MARINE PUMPS

Marine pumps are **classified** as follows: (1) Reciprocating pumps, (2) centrifugal pumps and (3) rotary pumps.

Reciprocating pumps are those in which the water pistons or plungers and the steam pistons travel alternately forward and backward in cylinders. When the pressure exerted behind the steam piston is transmitted directly through a piston rod to the water piston, the combination is known as a **direct-acting pump.** This is the type mostly employed aboard ship.

Where the pump is complete with steam cylinder and driving mechanism, i.e., a self-contained unit independent of the propelling engines, it is known also as an **independent pump.** Reciprocating pumps are frequently driven through links and beams from the crossheads of propelling engines, and in such cases are known as **connected pumps.**

Where reciprocating pumps discharge only at one end of the stroke, they are known as **single-acting;** where they discharge at both ends, they are known as **double-acting.**

Reciprocating pumps with one steam piston, one piston rod and one water plunger or piston are known as **single or "simplex"** pumps. Single pumps may be either single-acting or double-acting and either independent or connected. They are made both in vertical and horizontal designs.

Where two pumps of the same size are arranged side by side and the steam valve of one side takes its motion from the piston rod movement of the other side, the pump is known as duplex. **Duplex pumps** can be either single-acting or double-acting, independent direct-acting or driven from the propelling engines.

Reciprocating pumps are used for practically every pumping purpose aboard ship except when unusually large volumes of water are to be handled such as are required for circulating water through condensers. For some of the functions where reciprocating pumps have been employed, in recent years, there are certain cases where other types have been found to apply with equal, and in some cases with better adaptability; these include pumps of centrifugal type and vacuum apparatus of centrifugal, rotary or ejector types.

Centrifugal pumps, driven either by reciprocating steam engines, steam turbines or electric motors, are those in which the water enters a casing containing one or more revolving disks known as impellers. The impellers are arranged with vanes so designed that water enters at the hub and is guided along the vane surfaces with minimum losses to give sufficient velocity in the discharge piping to overcome the head against which the pump is operated.

There are two general types of centrifugal pumps in use, (*a*) **volute pumps** and (*b*) **turbine pumps.** The volute pump derives its name from the form of the casing which in section is similar in shape to the involute curve. **Volute pumps are used for** such service as supplying large volumes at low head as to condensers, sanitary systems, coolers, etc. These are made of the **single suction** and **double suction** types depending on the conditions of operation for which the pump is installed. **Turbine pumps are used where** water is to be delivered against high heads as for feeding boilers. In such pumps there are one or more impellers depending on the work to be performed. These impellers correspond to the water pistons or plungers of reciprocating pumps. They revolve with the shaft and are secured by keys. They are formed by two parallel disks with vanes cast between. Immediately surrounding the impellers are the diffusion vanes through which the water is delivered from the impellers to the pump casing. Where more than one impeller is used, the pump is known as multistage; the water in such an arrangement is delivered from the first stage into a channel ring which forms the suction chamber for the next impeller. This operation is repeated according to the number of stages in the pump, each impeller giving an equal increment of pressure; for construction see p. 1119.

Rotary pumps are principally used aboard ship for pumping fuel oil to burners, see p. 304.

Functions. The following are the functions of the pumping equipment of ships. Some vessels include pumps performing all the duties named be-

low; others contain few of them depending upon the type of ship and engine room arrangement.

1. Main condenser air pumps.
2. Main condenser circulating pumps.
3. Main boiler feed pumps.
4. Auxiliary boiler feed pumps.
5. Fire pumps.
6. Bilge pumps.
7. Salt water sanitary pumps.
8. Fresh water pumps.
9. Lubricating oil pumps.
10. Ballast pumps.
11. Fuel oil transfer pumps.
12. Fuel oil service pumps.
13. Fuel oil booster pumps.
14. Cargo oil pumps.
15. Auxiliary condenser air pumps.
16. Auxiliary condenser circulating pumps.
17. General service pumps.

Construction

Independent pumps consist of three elements: (1) steam end with its valve mechanism; (2) liquid or air end; and (3) a structure connecting the steam and pump ends to make the unit compact and self-contained.

Sizes of pumps are described by naming first the bore of the steam cylinders in inches, next the bore of the liquid or air cylinders and last the stroke. Thus a 14 in. × 9 in. × 24 in. "single" pump has a steam cylinder 14 in. in diam. a water cylinder 9 in. in diam. and both have a common stroke of 24 in.; or, if the pump is a "duplex" it has two steam cylinders each 14 in. in diam., two water or air cylinders 9 in. in diam., all with a common stroke of 24 in. There are also pumps having four dimensions, used as combined air and circulating pumps and these are made, as a rule, of the horizontal, single, piston, double-acting design. Such a pump designated as 10 in. × 14 in. × 16 in. × 12 in. would have a steam cylinder 10 in. in diam., an air cylinder 14 in. in diam., a circulating water cylinder 16 in. in diam., all having a common stroke of 12 in.

Materials used in the construction of pumps are outlined briefly as follows and, depending on the type and equipment of a vessel, certain extras are specified, all of which are noted below:

Commercial Specification

Steam End. Close-grained cast iron of ample thickness to allow for reboring of cylinder

Pump End. Close-grained cast iron cylinder fitted with brass lining, which in some pumps extends between the ports; in others it extends the entire length of the cylinder with the ports cut through the lining. The lining used and the manner of securing it vary, depending upon the design of the pump. Pump ends are made entirely of gunmetal at times, and when so made, they should be lined so that when reboring takes place a sufficient number of times, a new lining can be introduced.

Structures between steam and pump end consist either of heavy steel rods or cast iron columns.

Piston rods, where the stroke exceeds 12 in. are usually in two pieces, *i.e.*, piston rod of mild steel for the steam end and bronze for the liquid end; in smaller pumps the rods are made in one piece, either of steel or of bronze.

Crossheads are made of steel forgings or steel castings and at times are made of bronze.

Steam Valve Gear. For description of single piston types, see p. 1156. For descriptions of duplex type see p. 1160.

Pump valve seats and valves are usually of bronze and are of various types and design depending on the pump service; for various arrangements of these parts, see p. 1153.

Steam pistons are made of cast iron, fitted with carefully finished self-adjusting cast iron packing rings.

Water pistons are made of brass or iron with a removable follower for convenient access to the packing.

Fittings. Commercial pumps are supplied with all necessary special wrenches.

Extras. There are a number of materials and fittings in use which are regarded by manufacturers in the nature of extras. These are supplied with pumps when specially called for. Some of these extras are as follows: **Lagging steam cylinder barrels** with non-conducting material and **covering with planished steel** or Russia iron (at times the specifications call for the lagging of steam chest, but this is not done as a rule); **metallic packing** for steam piston rod and valve rod stuffing boxes (metallic packing is supplied where the rods are over 1 in. in diam. It is not considered advisable to use metallic packing for smaller rods and the use of **soft packing** is recommended); **lubricators** for steam cylinders; **mechanical lubricators** for moving parts of gear outside of the steam chest; **brass bushing** of steam valve gears; **relief valves** for water cylinders; **pump governors** or regulators; steam and exhaust **throttle valves; special materials** such as Monel Metal, acid-resisting bronzes, etc.

It is not always possible to apply all of these extras to pumps and care should be observed that they can be applied before specifying them.

Discharge air chambers are furnished as a regular part of the equipment with large pumps. Frequently **air chambers** are installed **on the suction side** of the pump where the pipe lines are long and have many branches and bends. Such chambers, however, must be fitted into the piping, and, as a rule, are not regarded as a part of the pump equipment but are supplied by the shipbuilder.

Service of Pumps. When pumps are made suitable for working pressures on liquid ends in excess of from 75 to 100 lbs. they are known as "Heavy Pressure" pumps, and when designed for lower pressures are designated as "Light Pressure." **Heavy pressure pumps are suited to** boiler feeding, fire service, for pumping fuel oil to burners, pumping cargo oil and for general service. **Light pressure pumps are suitable for** sanitary and fresh water systems, pumping ballast, lubricating oil, evaporator feed, condenser circulating, etc.

Commercial Sizes. Tables 1 to 15 indicate the sizes of pumps obtainable commercially. The size nearest that required to do the specified work should be selected from these tables so that pumps as made in quantities by manufacturers can be provided within reasonable limits of cost. Manufacturers will not, as a rule, undertake the manufacture of a few pumps of an odd size, but always offer the nearest manufactured size for reasons that are obvious. The steam cylinders used in both light pressure and heavy pressure pumps are alike as a rule, in so far as their ability to withstand pressure is concerned, and manufacturers do not maintain a separate line of patterns for light steam pressures, but combine their steam cylinders with pump cylinders of various sizes to perform the work to best advantage.

The maximum steam and liquid pressures for which various pumps are suited, are given in these tables. Figs. 1 to 22, which accompany the tables show the general appearance of pumps of the various sizes as well as the general features of design of the pumps listed.

Table 1. Vertical, Piston, Double-acting, Single, Heavy Pressure

(See Fig. 1. See page 1136)

Maximum Working Pressure: Steam End, 300 Lbs.; Water End, as per Table Below.

Size, in.			Maximum water pressure, lb. sq. in.	Gallons per stroke	Maximum capacity per minute			H. P. boiler at 35 lbs. Pump will feed at slow speed	Diam. pipes, in.				Space occupied, (without air chamber) in.		
Diam. steam cylinder	Diam. water cylinder	Stroke			No. single strokes	Ft. piston speed	Gallons		Steam	Exhaust	Suction	Discharge	Width	Depth	Height
3½	1¾	4	300	0.04	165	55	7	49	¾	½	¾	¾	9	12	32
3½	2¼	4	300	0.07	165	55	11	80	¾	½	1	¾	9	12	32
4	2½	4	300	0.09	165	55	14	100	¾	½	1	¾	9	12	32
4½	2¼	6	300	0.15	140	70	21	150	½	¾	1¼	1	10	15	41
6½	3½	8	300	0.33	135	90	45	320	¾	1	2	1½	13	19	51
7½	4	10	300	0.54	114	95	62	450	1	1¼	2½	2	15	21	63
8	5	12	300	1.02	100	100	102	725	1	1¼	3	2½	16	21	72
10	6	12	300	1.47	100	100	147	1050	1¼	1½	3½	3	16	27	72
10	7	12	300	2.00	100	100	200	1425	1¼	1½	4	3½	17	25	78
10	6	24	300	2.94	60	120	176	1510	1¼	2	3½	3	22	29	115
10	7	24	300	4.00	60	120	240	2060	1¼	2	½	4	23	29	115
12	8	12	300	2.61	100	100	261	1865	1½	2	4½	4	20	34	85
12	8	24	300	5.22	60	120	313	2685	1½	2	4½	4	28	34	128
14	9	24	300	6.61	60	120	397	3400	2	2½	5	4½	30	35	130
14	10	24	300	8.16	60	120	490	4200	2	2½	6	5	33	35	131
18¼	13½	24	300	14.85	60	120	890	7600	2½	3	8	7	44	41	149

For continuous service these pumps should be operated at from 25 to 50 per cent less than maximum listed speed.

Table 2. Horizontal, Piston, Double-acting, Single, Heavy Pressure

(See Fig. 2. See page 1136)

Maximum Working Pressure: Steam End, 250 Lbs.; Water End, 250 Lbs.

Size, in.			Gallons per stroke	Maximum capacity per minute			Boiler H.P. pump will feed at slow speed	Diam. pipes, in.				Floor Space, in.
Diam. steam cylinder	Diam. water cylinder	Stroke		No. single strokes	Feet piston speed	Gallons		Steam	Exhaust	Suction	Discharge	
3½	2¼	4	0.069	165	55	11	80	¾	½	1	¾	29× 8
4½	2¾	6	0.15	140	70	21	150	½	¾	1¼	1	39×11
5½	3¼	7	0.25	137	80	34	250	½	¾	1½	1¼	44×11
6½	4½	8	0.46	135	90	62	450	¾	1	2½	2	50×13
7½	4½	10	0.69	114	95	78	575	1	1¼	3	2½	59×15
8	5	12	1.02	100	100	102	725	1	1¼	3½	3	67×15
10	6	12	1.47	100	100	147	1050	1¼	1½	4	3½	70×15
12	7	12	2.00	100	100	200	1425	1½	2	5	4	70×15
14	8	12	2.61	100	100	261	1900	2	2½	5	4	75×18
16	10	18	6.12	74	110	453	3500	2½	3	8	6	96×22

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, we recommend from 25 to 50 per cent less than maximum listed speed.

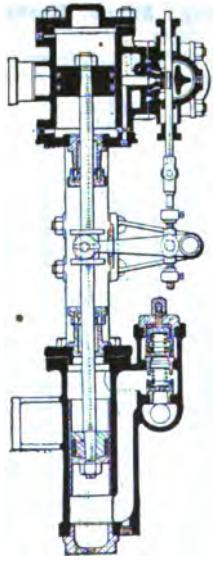


FIG. 1.—See Table 1, Page 1135.

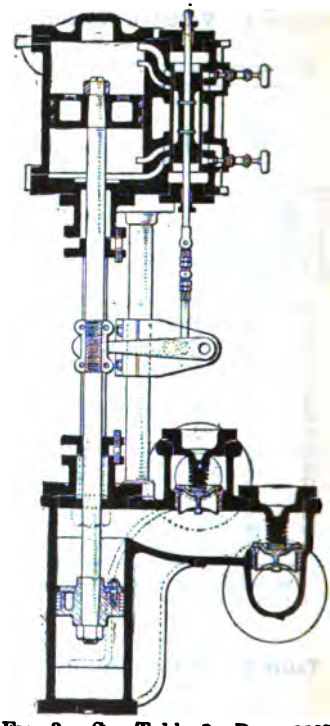


FIG. 3.—See Table 3, Page 1137

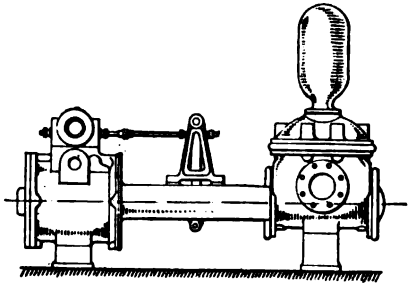


FIG. 2.—See Table 2, Page 1135.

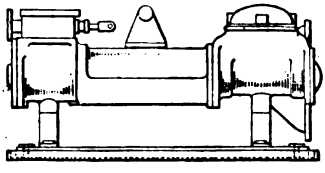


Fig. 4.—See Table 4, Page 1137.

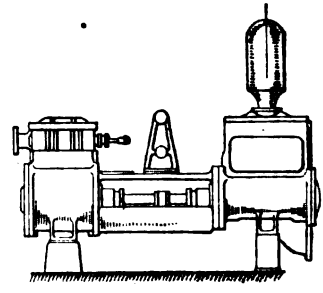


FIG. 5.—See Table 4, Page 1137

Table 3. Vertical, Piston, Double-acting, Duplex, Heavy Pressure

(See Fig. 3. See page 1136)

Maximum Working Pressure: Steam End, 300 Lbs.; Water End, 300 Lbs.

Size, in.			Gallons per stroke each piston	Maximum capacity per minute			H. P. boiler at 35 lbs. pump will feed at slow speed	Diam. pipes, in.				Space occupied, in.		
Diam. steam cylinder	Diam. water cylinder	Stroke		Single str. each piat.	Ft. piston speed	Gallons both cyls.		Steam	Exhaust	Suction	Discharge	Width	Depth	Height
4½	3	4	0.12	150	50	37	150	¾	1¼	2	2	23	19	38
5¼	3½	5	0.21	140	58	58	250	1	1½	2½	2½	29	22	46
6	4	6	0.33	130	65	85	350	1¼	2	4	3	32	23	50
7½	5	6	0.51	130	65	133	550	1½	2	5	4	33	28	56
9	6	6	0.73	130	65	191	800	2	2½	5	4	36	28	57
7½	4½	10	0.68	108	90	149	650	1½	2	4	3	32	26	66
9	6	10	1.22	108	90	264	1200	2	2½	5	4	36	27	66
10	7	10	1.66	108	90	360	1600	2	2½	6	5	40	34	77
12	8½	12	2.95	96	80	566	2800	2½	3	6	5	43	41	85

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, we recommend from 25 to 50 per cent less than maximum listed speed.

Table 4. Horizontal, Piston, Double-acting, Duplex, Heavy Pressure

Maximum Working Pressure: Steam End, 150 Lbs.; Water End, 150 Lbs. Pumps sizes to 12 × 6 × 10 inch with Cap and Valve Plate (see Fig. 4), and all larger sizes with Turret Top Water End (see Fig. 5. See page 1136)

Size, in.			Gallons per stroke each piston	Maximum capacity per minute			H. P. boiler at 35 lbs. pump will feed at slow speed	Diam. pipes, in.				Floor Space, inches
Diam. steam cylinder	Diam. water cylinder	Stroke		Single str. each piston	Ft. piston speed	Gallons both cyls.		Steam	Exhaust	Suction	Discharge	
2	1¼	2¾	0.014	160	37	4.5	25	¾	½	1	¾	21× 7
3	2	3	0.039	160	40	13	70	¾	½	1¼	1	24× 9
3½	2¼	4	0.065	150	50	19	100	¾	¾	1½	1¼	28× 9
4½	2¾	4	0.098	150	50	29	150	¾	¾	2	1½	33×13
5¼	3½	5	0.201	140	58	56	250	¾	1¼	2½	1½	38×16
6	4	6	0.310	130	65	80	350	1	1¼	3	2	44×17
7½	5	6	0.485	130	65	126	550	1½	2	4	3	45×22
7½	4½	10	0.654	96	80	132	650	1½	2	4	3	59×22
9	5¼	10	0.900	96	80	173	900	2	2½	4	3	61×23
10	6	10	1.18	96	80	225	1800	2	2½	5	4	62×26
10	6	12	1.47	90	90	260	1400	1½	2	5	4	68×22
12	6	10	1.18	96	80	225	1200	2	2½	5	4	71×31
10	7	10	1.62	96	80	310	1600	2	2½	6	5	60×26
12	7	12	1.94	90	90	350	1900	2	2½	6	6	76×31
14	7	12	1.94	90	90	350	1900	2½	3	6	6	78×33
12	8½	12	2.87	90	90	515	2800	2	2½	6	6	76×31
14	8½	12	2.87	90	90	515	2800	2½	3	6	6	78×33
16	8½	12	2.87	90	90	515	2800	2½	3	6	6	79×40
14	10¼	12	4.09	90	90	735	4000	2½	3	8	7	82×37
16	10¼	12	4.09	90	90	735	4000	2½	3	8	7	82×40

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, we recommend from 25 to 50 per cent less than maximum listed speed.

Table 5. Vertical, Twin Beam, Single-acting Air Pumps With One Steam Cylinder for Use with Surface or Jet Condensers

(See Fig. 6. See page 1139)

Pump size, inches				No. of revolutions per minute*	Ft. piston speed per minute	Normal capacity Lbs. steam per hr. 70 deg. inj., 30 in. bar.				Steam pressure necessary to drive	Diameter pipes, inches								Space occupied without condenser, inches		
						26″ vacuum		28″ vacuum			Pump					Condenser					
Diam. steam cylinder	Diam. air cylinder	Stroke	Jet condenser cone type			Jet condenser	Surface condenser	Jet condenser	Surface condenser		Steam	Exhaust	Suction	Disch. for jet cond.	Disch. surf. condenser	Injection inlet	Stand exh. inlet.	Max. Exh. inlet†	Height	Width	Depth
5	9½	8	12	38	51	1,750	6,480	620	3,680	75	¾	1	4	4	3½	3	5	8	58	31	16
6½	12	8	15	38	51	2,800	10,300	990	5,880	120	¾	1	5	5	4	4	6	10	58	36	19
7½	14	10	18	34	57	4,250	15,700	1,500	8,940	105	1	1¼	6	6	5	5	8	12	69	42	21
8	16	12	18	30	60	5,850	21,600	2,070	12,300	120	1	1¼	7	7	6	5	8	12	78	47	23
10	20	12	21	34	57	8,680	32,100	3,070	18,250	120	1¼	1½	8	8	7	6	12	16	82	51	28
10	20	12	24	30	60	9,140	33,800	3,230	19,200	120	1¼	1½	8	8	7	7	12	18	86	51	28
12	20	12	24	30	60	9,140	33,800	3,230	19,200	85	1½	2	8	8	7	7	12	18	86	51	28
12	22	18	24	28	70	12,900	47,700	4,560	27,100	105	1½	2	9	10	8	7	14	18	103	61	30
14	25	18	26	25	75	17,850	65,950	6,300	37,500	100	2	2½	10	12	9	8	18	20	116	65	36

*These speeds are recommended for continuous running, but may be increased from 15 to 25 per cent in case of emergency.

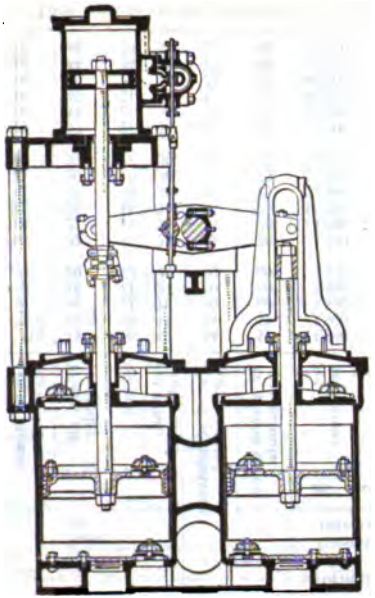


FIG. 6.—See Table 5, Page 1138.

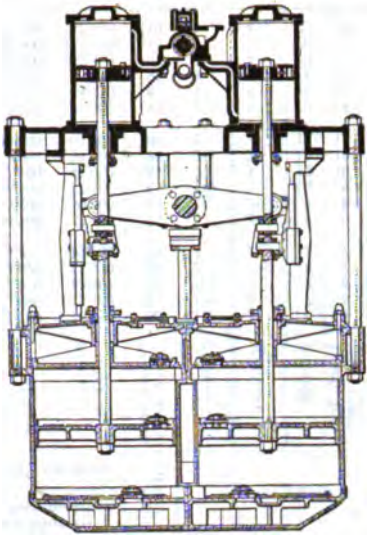


FIG. 7.—See Table 6, Page 1140.

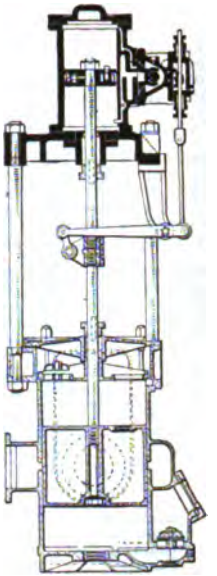


FIG. 8.—See Table 7, Page 1141.

Table 6. Vertical, Twin Beam, Single-acting Air Pumps for Use with Surface or Jet Condensers with Two Steam Cylinders

(See Fig. 7. See page 1139)

Size, in.				Normal capacity						Steam press. necessary to drive	Diameter pipes, in.								Space occupied without condenser, in.		
						Pounds steam per hour 70° injection, 30 in. barometer					For pump					For cond.					
						26" vacuum		28" vacuum													
Diam. steam cylinder	Diam. air cylinder	Stroke	Jet cond. cone type	No. of revolutions per minute*	Ft. piston speed per minute	Jet condenser	Surface condenser	Jet condenser	Surface condenser		Steam	Exhaust	Suction	Disch. for sur. cond.	Disch. for jet condenser	Injection inlet†	Standard Exh. inlet†	Maximum Exh. inlet†	Height	Width	Depth
7½	15	10	18	34	57	4,880	18,050	1,700	10,250	60	1½	2	6	5	6	5	8	12	72	42	23
7½	16½	10	18	34	57	5,900	21,850	2,080	12,400	75	1½	2	7	6	7	5	10	12	72	44	25
8	16	12	18	30	60	5,850	21,600	2,070	12,300	60	1½	2	7	6	7	5	10	12	80	49	28
9	18	12	21	30	60	7,400	27,350	2,620	15,550	60	2	2½	8	7	8	6	10	16	83	56	30
9	20	12	24	30	60	9,140	33,800	3,230	19,200	75	2	2½	8	7	8	7	12	18	85	56	30
10	22	15	24	28	70	12,900	47,700	4,560	27,100	75	2	2½	9	8	10	7	14	18	102	65	32
10	23½	15	24	28	70	14,700	54,400	5,200	30,900	85	2	2½	9	8	10	7	16	18	104	65	32
12	25	18	26	25	75	17,850	65,950	6,300	37,500	65	2½	3	10	9	12	8	18	20	118	68	37
12	28	18	30	25	75	22,400	82,700	7,900	47,050	85	2½	3	12	10	12	10	20	24	121	68	40
12	30	18	30	25	75	25,700	95,000	9,080	54,000	95	2½	3	12	10	14	10	22	24	122	70	40
14	32	21	30	22	77	30,050	110,900	10,600	63,100	80	2½	3	14	12	14	10	24	24	139	85	48
14	35	21	36	22	77	35,900	132,700	12,700	75,500	95	2½	3	16	14	16	12	24	30	143	85	48
16	38	21	36	22	77	42,300	156,400	15,000	88,950	85	3	4	18	16	18	12	24	30	145	89	58
16	40	21	36	22	77	46,950	173,500	16,600	98,750	95	3	4	18	16	18	12	28	30	150	93	60
16	40	24	36	20	80	48,800	180,200	17,250	102,450	95	3	4	18	16	18	12	28	30	160	93	60
16	44	24	40	20	80	58,900	217,250	20,800	123,750	115	3	4	20	18	20	14	30	36	167	108	60
18	48	24	58	20	80	70,150	259,100	24,800	147,400	110	3	4	22	20	22	16	36	42	188	129	64

*These speeds are recommended for continuous running, but may be increased from 15 to 25 per cent in case of emergency.

Table 7. Vertical, Single, Double-acting, Suction Valveless Air Pumps for Use with Surface Condensers

(See Fig. 8. See page 1139)

Size, in.			Displacement, cubic feet per stroke	*Normal capacity per minute			Diam. pipes, in.				Space occupied, in.		
Diam. steam cylinder	Diam. air cylinder	Stroke		No. single strokes	Ft. piston speed	Lbs. steam per hour 28" V., 70° inj. 30" B.	Steam	Exhaust	Suction	Discharge	Height	Width	Depth
3½	5	4	0.037	195	65	1,840	⅜	½	2	2	33	9	14
4½	8	6	0.153	140	70	5,060	½	¾	3½	3	40	13	16
6½	10	8	0.318	120	80	9,060	¾	1	4½	4	57	17	24
6½	12	8	0.458	120	80	13,060	¾	1	5	4½	61	23	25
7½	14	10	0.802	102	85	18,900	1	1¼	6	5	72	25	27
8	16	12	1.220	90	90	26,100	1	1¼	7	6	86	27	31
10	18	12	1.546	90	90	33,100	1¼	1½	8	7	86	29	36
10	20	16	2.640	75	100	45,400	1¼	1½	9	8	102	35	40
14	24	18	4.320	67	100	65,300	2	2½	12	10	115	42	42

*These speeds are recommended for continuous running, but may be increased from 15 to 25 per cent in case of emergency.

Table 8. Horizontal, Single, Double-acting Air Pumps for Use with Surface Condensers

(See Fig. 9. See page 1142)

Size, in.			Maximum capacity surface *condenser		Diam. pipes, in.				Floor space, in.
Diam. steam cylinder	Diam. air cylinder	Stroke	Single strokes per min.	Lbs. steam per hour pump will handle	Steam	Exhaust	Suction	Discharge	
3½	5	4	...	.....	⅜	½	2	1½	37×9
4½	6½	6	...	.....	½	¾	3	2½	42×12
5½	8	7	137	5,800	½	¾	3½	3	46×18
6	6	12	...	.....	¾	1	3	2½	65×14
6	8	10	114	6,530	¾	1	4	3½	65×19
6	8	12	100	7,250	¾	1	4	3½	65×19
6	10	12	100	11,300	¾	1	5	4	67×24
8	10	12	100	11,300	1	1¼	5	4	70×24
8	12	12	100	16,300	1	1¼	5	4	70×25
8	14	12	100	22,200	1	1¼	6	5	75×28
8	14	18	74	24,700	1	1¼	6	5	86×28
10	12	12	...	.....	1¼	1½	5	4	70×25
10	14	12	100	22,200	1¼	1½	6	5	78×28
10	14	18	74	24,700	1¼	1½	6	5	86×29
10	16	18	74	32,100	1¼	1½	8	6	88×30
12	18	18	74	40,700	1½	2	10	8	94×30
12	18	24	60	44,100	1½	2	10	8	118×30
12	28	24	60	54,400	1½	2	12	10	118×32
14	20	24	60	54,400	2	2½	12	10	118×32
14	22	24	60	65,800	2	2½	12	10	118×35
14	24	24	60	78,400	2	2½	14	12	118×39
16	24	24	60	78,400	2½	3	14	12	122×39
16	26	24	60	91,900	2½	3	14	12	124×45
16	28	24	60	106,500	2½	3	14	12	124×50
16	30	30	52	132,500	2½	3	14	14	136×52

*For continuous operation we recommend a speed from 20 to 25 per cent less than maximum given above.

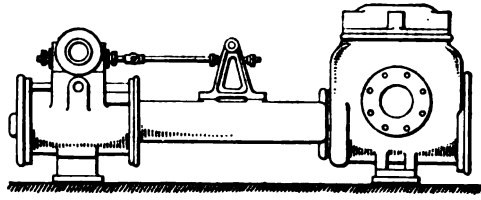


FIG. 9.

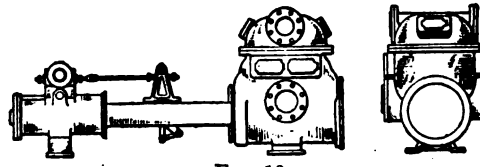


FIG. 10.

Table 9. Horizontal, Single, Double-acting Air Pumps for Use With Jet Condensers

(The jet condensers with which these pumps are fitted are of the spray pipe or adjustable cone type. Generally pumps of sizes 8 × 12 × 12 and smaller are cap and valve plate type, see Fig. 9. Sizes 8 × 14 × 12 and larger are close-clearance type, see Fig. 10.

Size, in.				†Maximum capacity		Diameter pipes, in.						Floor space without cond.
						For pump			For condenser			
Diam. steam cylinder	Diam. air cylinder	Stroke	Jet cond. cone type size, in.	Single strokes per min.	Lbs. st. hr. 26" vac., 70° inj., 30" bar.	Steam	Exhaust	Discharge	Injection inlet	Standard exh. inlet†	Maximum exh. inlet†	
5½	8	7	10	137	1,950	½	¾	3	2	5	6	46×18
6	8	10	12	114	2,190	¾	1	3½	3	6	8	65×19
6	8	12	12	100	2,435	¾	1	3½	3	6	8	65×19
6	10	12	15	100	3,800	¾	1	4	4	8	10	67×24
8	10	12	15	100	3,800	1	1¼	4	4	8	10	70×24
8	12	12	15	100	5,480	1	1¼	5	4	8	10	70×25
8	14	12	18	100	7,460	1	1¼	6	5	10	12	75×28
8	14	18	18	74	8,280	1	1¼	6	5	12	12	86×28
10	12	12	15	100	5,480	1¼	1½	5	4	8	10	74×25
10	14	12	18	100	7,460	1¼	1½	6	5	10	12	78×28
10	14	18	18	74	8,280	1¼	1½	6	5	12	12	86×29
10	16	18	21	74	10,800	1¼	1½	8	6	12	16	88×30
12	18	18	24	74	13,700	1½	2	8	6	14	18	94×30
12	18	24	24	60	14,800	1½	2	8	7	16	18	118×30
12	20	24	24	60	18,250	1½	2	10	7	16	18	118×32
14	20	24	24	60	18,250	2	2½	10	7	16	18	118×32
14	22	24	26	60	22,100	2	2½	10	8	18	20	118×35
14	24	24	26	60	26,300	2	2½	12	8	20	20	118×39
16	24	24	26	60	26,300	2½	3	12	8	20	20	122×39
16	26	24	30	60	30,850	2½	3	12	10	22	24	124×45
16	28	24	30	60	35,800	2½	3	12	10	22	24	124×50
16	30	30	30	52	44,500	2½	3	14	10	24	24	136×52

† For continuous operation a speed of from 20 to 25 per cent less than the maximum given above is recommended. In case of emergency speeds may be increased.

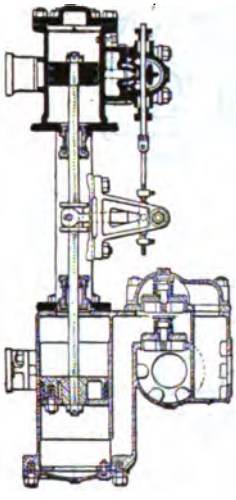


FIG. 11.

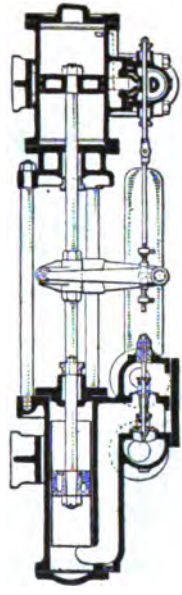


FIG. 12.

Table 10. Vertical, Piston, Double-acting, Single, Light Pressure

(See Figs. 11 and 12)

Size, inches			Maximum water pressure, lbs. sq. in.	Gallons per stroke	Maximum capacity per minute			Diameter pipes, inches				Space occupied (without air chamber) inches		
Diam. steam cylinder	Diam. water cylinder	Stroke			No. single strokes	Ft. piston speed	Gallons	Steam	Exhaust	Suction	Discharge	Width	Depth	Height
3½	4	4	75	0.22	165	55	36	¾	½	2	1½	23	16	38
4½	5	6	75	0.51	140	70	71	½	¾	2½	2	23	17	41
4½	6	6	75	0.73	140	70	103	½	¾	3	2½	23	21	42
6½	7	8	75	1.33	135	90	180	¾	1	3½	3	26	23	61
6	7	12	50	2.00	100	100	200	¾	1	4	3½	24	26	72
7	8	12	50	2.61	100	100	261	1	1¼	4½	4	30	26	75
8	10	12	35	4.08	100	100	408	1	1¼	6	5	36	32	75
10	10	12	35	4.08	100	100	408	1¼	1½	6	5	36	32	77
10	12	12	35	5.87	100	100	587	1¼	1½	7	6	37	33	78
12	14	12	35	8.00	100	100	800	1½	2	8	7	41	38	81

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, we recommend from 25 to 50 per cent less than maximum listed speed.

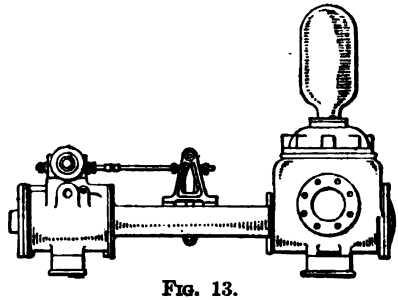


FIG. 13.

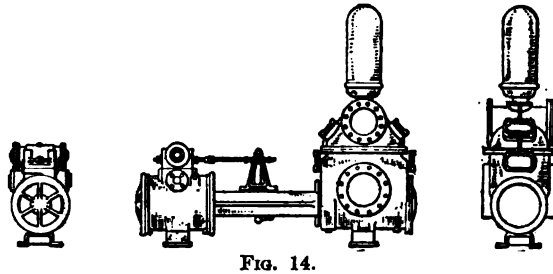


FIG. 14.

Table 11. Horizontal, Piston, Double-acting, Single, Light Pressure

Maximum Working Pressure: Steam End, 250 Lbs.; Water cylinders 12 in. in diam and smaller are of the "cap and valve plate" pattern, like Fig. 13; larger sizes are of the "close clearance" type (see Fig. 14.) Water cylinders 8 in. in diam. and smaller are good for water pressure corresponding to 230 feet (100 lbs.) head, or less; larger sizes, for pressures corresponding to 175 feet (75 lbs.) head, or less.

Size, inches			Gallons per stroke	Maximum capacity per minute			Diameter pipes, inches				Floor space, in.
Stm. cyl.	Water cyl.	Stroke		No. single strokes	Ft. piston speed	Gallons	Steam	Exhaust	Suction	Discharge	
3½	3½	4	0.17	165	55	27	⅜	½	1¼	1	28×10
4½	4⅛	6	0.35	140	70	49	½	¾	2½	2	46×14
5½	5½	7	0.72	137	80	98	½	¾	3	2½	46×14
6	6	12	1.47	100	100	147	¾	1	3½	3	66×15
7½	7	10	1.67	114	95	190	1	1¼	4	3	54×16
8	8	12	2.61	100	100	261	1	1¼	5	4	66×16
10	10	12	4.08	100	100	408	1¼	1½	6	6	74×22
10	10	18	6.12	74	110	453	1¼	1½	6	6	89×19
10	12	12	5.87	100	100	587	1¼	1½	8	6	74×25
10	12	18	8.81	74	110	652	1¼	1½	8	6	90×25
12	12	12	5.87	100	100	587	1½	2	8	6	76×25
12	12	18	8.81	74	110	652	1½	2	8	6	93×25
14	14	18	12.00	74	110	888	2	2½	10	8	99×29
14	16	18	15.66	74	110	1159	2	2½	12	10	101×31
16	16	18	15.66	74	110	1159	2½	3	12	10	101×31

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, we recommend from 25 to 50 per cent less than maximum listed speed.

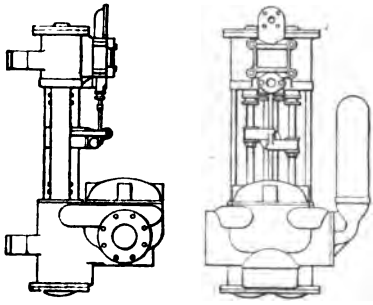


FIG. 15.

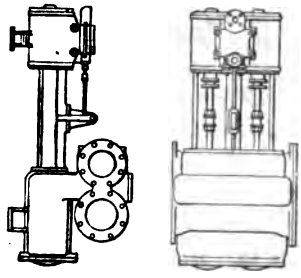


FIG. 16.

Table 12. Vertical, Piston, Double-acting, Duplex, Light Pressure

(See Figs. 15 and 16)

Maximum Working Pressure: Steam End, 125 Lbs.; Water End, Size 7½ × 6 × 6 and Smaller Cap and Valve Plate Type, (Fig. 15), 75 lbs.; Larger Sizes, (Fig. 16), 50 Lbs.

Size, inches				Maximum capacity per minute			Diameter pipes, inches				Space occupied, inches		
Diam. steam cylinder	Diam. water cylinder	Stroke	Gallons per stroke each piston	No. single strokes each piston	Ft. piston speed	Gallons both cyls.	Steam	Exhaust	Suction	Discharge	Width	Depth	Height
3	3	3	0.09	160	40	29	½	¾	1½	1¼	9	12	28
4½	3¾	4	0.19	150	50	57	½	¾	2½	1½	16	23	34
5¼	5	5	0.42	140	58	119	¾	1¼	4	3	17	24	41
6	6	6	0.73	130	65	191	1	1½	5	4	21	27	47
7½	6	6	0.73	130	65	191	1¼	1½	5	4	21	28	51
6	7½	6	1.15	130	65	298	1	1½	5	4	36	24	55
7½	7½	6	1.15	130	65	298	1¼	1½	5	4	36	24	58
6	9	6	1.65	130	65	429	1	1½	6	5	41	26	58
7½	9	6	1.65	130	65	429	1¼	1½	6	5	41	26	58
9	9	6	1.65	130	65	429	2	2½	6	5	41	26	60
7½	9	10	2.75	108	90	595	1¼	1½	6	5	41	30	70
7½	10¼	10	3.57	108	90	771	1¼	1½	8	7	36	33	70
9	12	10	4.90	108	90	1058	2	2½	10	8	52	38	74
10	12	10	4.90	108	90	1058	2	2½	10	8	52	38	74

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, from 25 to 50 per cent less than maximum listed speed is ecommended.

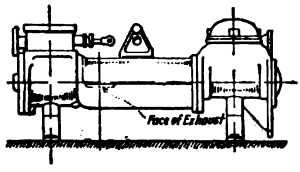


FIG. 17.

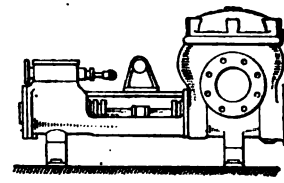


FIG. 18.

Table 13. Horizontal, Piston, Double-acting, Duplex, Light Pressure

Sizes as listed, up to and including 7½ × 6 × 10, generally have "cap and valve plate" water ends, Fig. 17. All larger sizes have "turret top" water ends, Fig. 18.

Maximum Working Pressure: Steam End, 150 Lbs.; Water End, 75 Lbs.

Size, inches			Gallons per stroke each piston	Maximum capacity per minute			Diameter pipes, inches				Floor space, inches
Diam. steam cylinder	Diam. water cylinder	Stroke		No. single strokes each piston	Ft. piston speed	Gallons both cylinders	Steam	Exhaust	Suction	Discharge	
3	2¾	3	0.08	160	40	24	¾	½	1¼	1	24× 9
4½	3¾	4	0.19	150	50	56	½	¾	2½	1½	36×13
5¼	4¾	5	0.38	140	58	105	¾	1¼	3	2	39×16
6	5¾	6	0.67	130	65	170	1	1¼	4	3	47×17
6	7½	6	1.14	130	65	295	1	1¼	6	5	48×21
7½	7½	6	1.14	130	65	295	1½	2	6	5	49×22
6	8½	6	1.46	130	65	380	1	1¼	6	5	49×22
7½	8½	6	1.46	130	65	380	1½	2	6	5	49×22
7½	6	10	1.19	96	80	228	1½	2	5	4	57×30
7½	7	10	1.63	96	80	312	1½	2	6	5	58×24
7½	8½	10	2.42	96	80	465	1½	2	6	5	59×24
9	8½	10	2.42	96	80	465	2	2½	6	5	61×24
10	10¼	10	3.54	96	80	680	2	2½	8	7	66×30
9	12	10	4.89	96	80	940	2	2½	10	8	75×36
10	12	10	4.89	96	80	940	2	2½	10	8	75×36
12	12	10	4.89	96	80	940	2½	3	10	8	80×36
12	14	10	6.62	96	80	1270	2½	3	12	10	84×42
14	14	10	6.62	96	80	1270	2½	3	12	10	85×42
8	8½	12	2.91	90	90	525	1½	2	6	5	65×24
10	10¼	12	4.24	90	90	765	2	2½	8	7	72×36
10	12	12	5.84	90	90	1050	2	2½	10	8	78×36
12	12	12	5.84	90	90	1050	2½	3	10	8	84×36
12	14	12	7.94	90	90	1430	2½	3	12	10	88×42
14	14	12	7.94	90	90	1430	2½	3	12	10	88×42

These pumps can be run at a greater speed than above listed, in case of emergency, but for continuous work, from 25 to 50 per cent less than maximum listed speed is recommended.

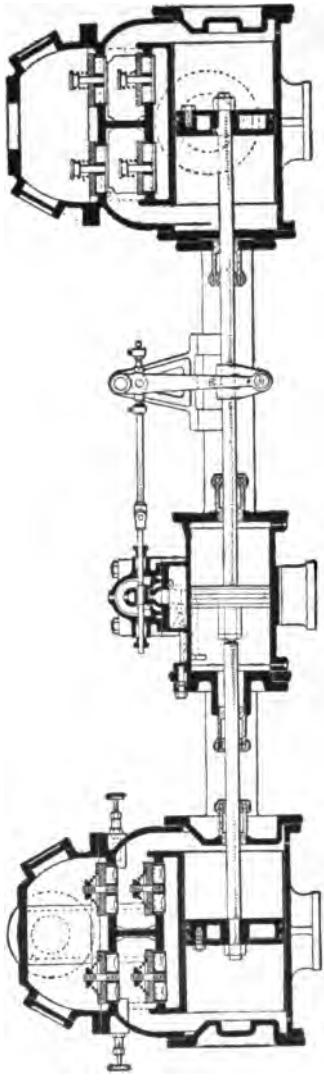


FIG. 19.

Table 14. Combined Air and Circulating Pumps

(See Fig. 19)

Size, inches				*Maximum capacity pounds steam per hour, 26″ vacuum, 30″ barometer				Diameter pipes, inches						Boiler H. P. necessary at 35 lbs. evap. †	Max. cap. per min.			Floor space, inches
				Temperature injection water						Air end		Cir. end.						
Diam. steam cylinder	Diam. air cylinder	Diam. water cylinder	Stroke	50°	60°	70°	80°	Steam	Exhaust	Suction	Discharge	Suction	Discharge		Number of single strokes	Ft. pist. speed	Gallons circ. water	
3½	5	5	4	1,470	1,227	988	743	⅜	½	2	1½	2½	2	3	150	50	51	54×13
4½	5½	5½	6	2,260	1,875	1,518	1,141	½	¾	2½	2	3	2½	5	130	65	80	64×13
5½	6	6	7	3,071	2,562	2,052	1,555	½	¾	3	2½	3½	3	6	120	70	102	72×15
5½	8	8	7	5,460	4,560	3,673	2,762	½	¾	3½	3	4	4	11	120	70	182	86×18
7½	8	8	10	6,450	5,420	4,360	3,280	1	1¼	4	3½	5	4	13	102	85	221	94×18
7½	9	9	10	8,230	6,870	5,530	4,160	1	1¼	4	3½	5	4	17	102	85	280	97×19
8	10	10	12	10,680	8,920	7,175	5,400	1	1¼	5	4	6	5	22	90	90	367	110×25
10	12	12	12	15,350	12,820	10,325	7,780	1¼	1½	5	4	8	6	31	90	90	528	113×25
12	14	14	12	20,900	17,480	14,150	10,580	1½	2	6	5	10	8	43	90	90	720	121×31
12	14	14	18	23,000	19,210	15,460	11,630	1½	2	6	5	10	8	48	67	100	800	140×30
14	16	16	18	30,000	25,050	20,280	15,200	2	2½	8	6	12	10	62	67	100	1044	150×30
14	18	18	18	38,030	31,700	25,590	19,250	2	2½	10	8	12	10	79	67	100	1322	150×33
16	18	18	18	38,030	31,700	25,590	19,250	2½	3	10	8	12	10	79	67	100	1322	154×33
16	18	18	24	41,500	34,650	27,900	21,000	2½	3	10	8	12	10	87	55	110	1454	181×33
16	20	20	18	47,000	39,200	31,500	23,750	2½	3	10	8	12	10	97	67	100	1632	154×33

* For continuous operation a speed of 20 to 25 per cent less than maximum given working pressures: Steam end, 250 lbs.; circulating end, 35 feet head; vacuum end, 26 in. with 30-inch barometer is recommended.

† Conservative estimate for pump working at above capacity.

Air inlet and circulating discharge are regularly on top for mounting surface condenser, but if pump is to be located independently of condenser, openings can be placed on sides of cylinders if necessary.

Cargo Oil Pumps

For handling oil in bulk special types of pumps have been designed. When Pennsylvania oils of paraffin base were the only ones handled, ordinary duplex pumps were employed and in some cases types such as shown in Fig. 20 were used. These were made up with the liquid end in three parts, (1) the cylinder

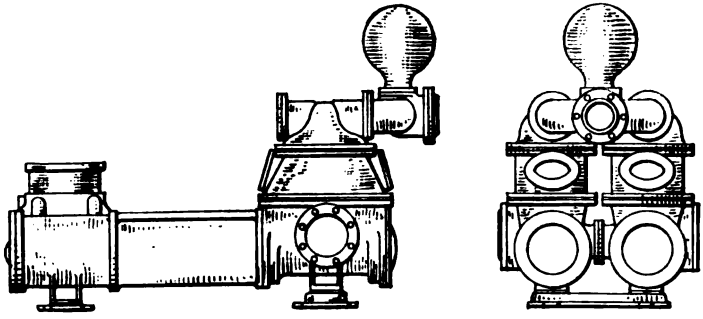


FIG. 20.

barrels, (2) the intermediate chamber, (3) the pump cap or discharge chamber The suction and discharge valves of these pumps are ordinary poppet valves carried on plates, the suction valves below the intermediate piece and the discharge valves above These plates are removable and can be replaced without disturbing parts of the liquid cylinder. With the development of oil

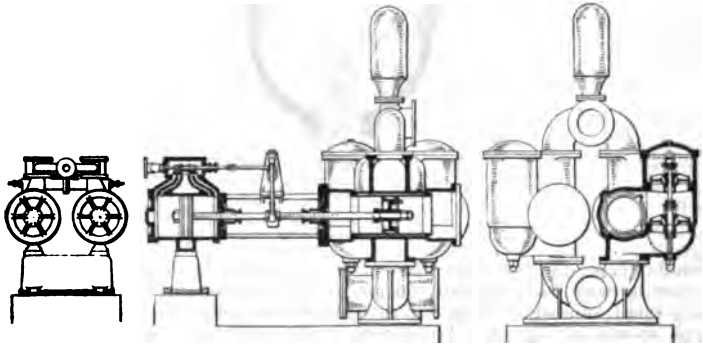


FIG. 21.

fields in California and Mexico, the handling of asphalt base oils resulted in the development of pumps of a different type, Fig. 21. This type, known as the **valve pot pattern**, is of simple construction and has but eight valves in the entire pump, *i.e.*, a suction and a discharge valve for each end of each cylinder. Fig. 22 shows the arrangement of the liquid valve chamber.

In all oil pumps the **fluid passages** should be liberal and direct and so designed that the direction of the flow of the oil shall be reversed as few times as possible in its passage through the pump and so as to afford unobstructed flow without the formation of air pockets. On account of grit and other foreign matter in the oil, the linings in the fluid cylinder should be removable and made of material suitable for the service. They should be carefully finished, fitted in place and secured by proper bolting. The free area through the valve seats should be ample. Pumps of the type shown in Fig. 21 are made with valves of the double beat type, composed of an annular ring joined to a hub by arms or ribs. This construction affords a fluid passage around both the inner and outer periphery of the valve. The valve seats are made of the

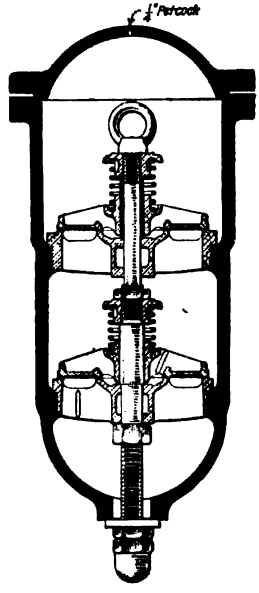


FIG. 22.

annular type, the two rings which form the seat so made that the inner ring is cast integral with the hub and joined to the outer ring by ribs. The use of a small number of large valves reduces obstruction to flow over that of type shown in Fig. 20. These pumps are made both simple duplex and compound duplex. Table 15 gives dimensions, valve areas, etc., of pumps of the type shown in Fig. 21, which are in common use in modern oil tankers. The material of which these are made depends upon the nature of the oil handled and whether the pumps alternately handle oil and salt water. Where the pumps are delivering oil in which there is an appreciable amount of sulphur, all brass fittings should be omitted and the liquid end made entirely of iron. Where, however, the oil is free from sulphur and the pump is used at times for pumping salt water ballast, brass fittings referred to for pumps generally can be used.

Table 15. Duplex Cargo Oil Pumps
200 Lbs. Maximum Working Pressure (Fig. 21)
(Barrel = 42 U. S. Gallons)

Size, in.	Normal rating			Pipes				Valve service				Floor space		
	Gallons per minute	Barrels per hour	Piston speed in feet	Steam	Exhaust	Suction	Discharge	No. per corner	Size	Area	Percentage	Length	Width	Height
12×10 ×18	710	1012	90	2½	3	10	8	1	11″	60	77	9′ 0″	5′ 8″	8′ 7″
12×10½×18	782	1115	90	2½	3	10	8	1	11″	60	70	9′ 0″	5′ 8″	8′ 7″
12×11 ×18	858	1222	90	2½	3	10	8	1	11″	60	62	9′ 0″	5′ 8″	8′ 7″
14×10 ×18	710	1012	90	2½	3	10	8	1	11″	60	77	9′ 0″	5′ 8″	8′ 7″
14×10½×18	782	1115	90	2½	3	10	8	1	11″	60	70	9′ 0″	5′ 8″	8′ 7″
14×11 ×18	858	1222	90	2½	3	10	8	1	11″	60	62	9′ 0″	5′ 8″	8′ 7″
16×12 ×18	1010	1444	90	2½	3½	12	10	1	15¼	130	112	10′ 5″	6′ 11″	10′ 4″
16×13 ×18	1200	1715	90	2½	3½	12	10	1	15¼	130	98	10′ 5″	6′ 11″	10′ 4″
16×14 ×18	1390	1986	90	2½	3½	12	10	1	15¼	130	84	10′ 5″	6′ 11″	10′ 4″
16×12 ×24	1360	1944	120	2½	3½	12	10	1	15¼	130	112	11′ 6″	6′ 11″	10′ 4″
16×13 ×24	1600	2286	120	2½	3½	12	10	1	15¼	130	98	11′ 6″	6′ 11″	10′ 4″
16×14 ×24	1850	2641	120	2½	3½	12	10	1	15¼	130	84	11′ 6″	6′ 11″	10′ 4″
18×13 ×24	1600	1944	120	3	4	12	10	1	15¼	130	98	11′ 7″	6′ 11″	10′ 4″
18×14 ×24	1850	2641	120	3	4	12	10	1	15¼	130	84	11′ 7″	6′ 11″	10′ 4″

PERFORMANCE FACTORS

Volumetric efficiency = $E_v = V_1 \div V$, where V_1 = volume discharged by pump and V = displacement of plunger, both expressed in same units. The loss (1 — Volumetric efficiency) is called the **slip.** This slip **represents losses due to** defective piston packing, leaky stuffing boxes or valves, delayed closing of valves and air entrained in the liquid being pumped. The slip in a new pump is very small, 3 to 5 per cent may be expected on test but this includes a possible error of observation of 2 per cent. When the slip is excessive the pump should be overhauled. To test marine pumps for tightness, close the discharge valve and admit sufficient steam to produce the required pressure. For marine pumps the number of strokes, called **lost actions,** should not exceed 1 per cent of the normal speed if the pump is in perfect condition.

Hydraulic efficiency = $E_h = h_1 \div h$, where h_1 = total head pumped against and $h = h_1 +$ hydraulic losses. These **hydraulic losses** consist of two parts, suction and discharge losses, the former comprise (1) velocity head, (2) entrance head, (3) friction in suction pipe, (4) losses in bends and (5) losses through suction valves. The discharge losses comprise (1) losses passing through discharge valves and (2) friction in discharge passages; these discharge losses are small amounting to possibly 1 per cent of the discharge

pressure. The sum of the suction losses will rarely exceed 3 ft. of water column in a well designed marine installation. The main discharge pipe friction losses are not charged against hydraulic efficiency.

Indicated efficiency = E_i = volumetric efficiency times hydraulic efficiency and all losses in the pump end are represented.

Mechanical efficiency = E_m = H.P.$_w$ + H.P.$_s$, where $H.P._w$ equals power developed in the water end of pump and $H.P._s$ equals power developed in steam end; this efficiency represents friction losses in the mechanism transmitting power to the pump. The following table gives approximate values for mechanical efficiencies of various types direct-acting pumps:

Mechanical Efficiencies of Direct-acting Pumps

Stroke, in.	3	4	5	6	8	10	12	15	18	24
E_m, Piston and Plunger Pumps, 150 lbs.	50	55	60	65	70	75	77.5	80	82.5	85
E_m, Outside Packed Pumps, 300 lbs.	47	52	57	61	66	71	74	76	78	81
E_m, Pressure Pumps, 1000 lbs.	45	50	54	58	63	67	70	72	74	77

Total Efficiency = $E_r \times E_h \times E_m$. This comprises all losses in the pump and driving mechanism and it indicates the economy of the whole installation.

Capacity of a single, double-acting pump in gal. per min. equals

$$D = \frac{a \times P \times 12}{231} = 0.0408 d^2 P \text{ where}$$

a = area of plunger in sq. in.,
d = diameter of plunger in sq. in., and
P = piston speed in ft. per min.

From these formulæ the displacement of any other type pump may be obtained as follows:

Single, single-acting pump, displacement = $D \div 2$.

Duplex, double-acting pump, displacement = $D \times 2$.

The displacement of the plunger rod must be allowed for in the above formulæ.

The actual delivery of a pump will be less than that given by the above formulæ, due to the slip; deduct about 3 per cent for slip.

Speeds of all sizes and types of marine pumps are given in Tables 1 to 15.

DESIGN FACTORS

Cylinders. For all curved or cylindrical surfaces

$$d \times p = 2t \times f$$

where
d = inner diameter of curvature in in.
p = internal pressure in lbs. per sq. in.
t = thickness of metal in in.
f = allowable stress of material, lbs. per sq. in.

Spherical parts, being twice as strong as cylinders of the same diameter, can be computed by the formula

$$d \times p = 4t \times f.$$

Flat surfaces may be calculated by the formula

$$t = 0.1d\sqrt{\frac{p}{200}}$$

Head Bolting. Allow 6000 lbs. per sq. in. of the stud area measured at root of thread. **Min. diam. of stud circle** = $d + 2(S + \frac{1}{4}$ in.), where d = diam. of cylinder in in. and S = diam. of stud in in. For through bolts this minimum diameter is determined by the outside diameter of the cylinder. **Min. thickness of flange** = $S + \frac{1}{4}$ in. Min. diam. of flange = $B + 2(S + \frac{1}{8}$ in.), where B = diam. of stud circle in in.

Pitch of Bolts. To prevent the flange from springing between bolts the pitch should not exceed the following:

Pressure, lbs. per sq. in.	Pitch, in.
0– 75	$10 \times b$
75–150	$7.5 \times b$
150–300	$5 \times b$

where b = diam. of bolt in in.

Service Condition vs. Various Types

Boiler feed pumps should be supplied in sufficient capacity so that the speed can be reduced to ½ max. speed under service conditions. This will allow ample reserve capacity in case of low water or other emergency. **For capacities of reciprocating boiler feed pumps,** see pp. 1135 to 1137. **For centrifugal boiler feed pumps,** see p. 1127. As a rule steam used in the pump is utilized to heat feed water and, since most of the heat is thus reclaimed, steam economy is not of primary importance.

For Condenser Circulating Pumps, see p. 1123.
For Surface Condenser Air Pumps, see pp. 1138 to 1141.
For Jet Condenser Air Pumps, see pp. 1140 and 1147.

VALVE SERVICE

There is no more important part of a pump than the pump valve service. The pump will be an unsatisfactory and inefficient machine if valves are not of suitable design and of proper materials. Figs. 23 to 30 depict excellent valve

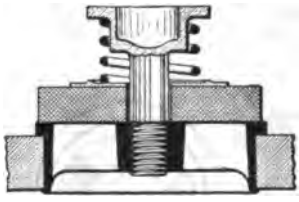


FIG. 23.—Standard Bronze Valve for Water Pump.

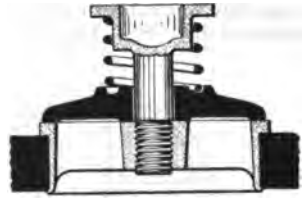


FIG. 24.—Standard Rubber Valve for Water Pump.

practice. For unusual conditions, where work is such that other than regular service is desired, the pump manufacturers should be consulted for a proper determination of the requirements. Details of construction will vary with the valve size and with the structural requirements and service of the pump.

Standard valves for water pumps are shown in Figs. 23 and 24, the **bronze valve** being regularly used for general service and where hot water is to be handled, in pumps shown in pp. 1135 to 1137. The **rubber valve** is an alternative for moderately warm or cold water service or for light pressure.

Valve seats may be forced into **valve decks**, the fit being made on a slight taper and the projecting edge peaned over to prevent the seat from working out. On some types of pumps seats are screwed into the deck, the threads being on a slight taper to insure a tight fit. The **stem** on which the valve rises is made with a guard which limits the valve movement, and the depth of the guard is such that when valve strikes

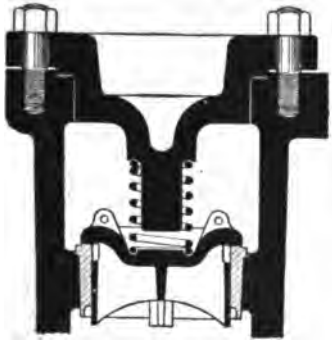


FIG. 25.—Wing Guided Valve.

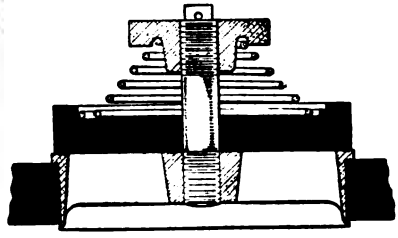
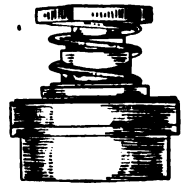


FIG. 26.—Air Pump Rubber Valve.

against it there is room for the **spring** to compress so that coils will not hit each other and possibly break.

Flat-faced bronze poppet valves, guided by wings which work in the valve seat opening, this being clear of ribs or other obstructions, Fig. 25, are used on pot valve pumps and vertical duplex pumps of the type on pp. 1136 and 1137.



Horizontal double-acting air pump valve service, for pumps shown on p. 1142, is of the type shown in Fig. 26, the valve being of a moderately soft rubber composition so as to insure maximum tightness. The raised edge around the

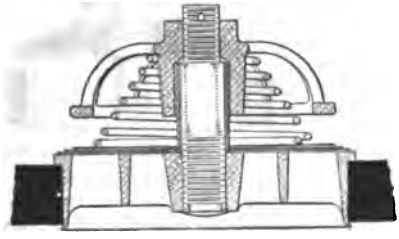


FIG. 27.—Kinghorn Valve.

FIG. 28.—Turbine or Rotating Valve.

outside rim is for the twofold purpose of stiffening the valve and for guiding the valve spring which is conical, with its lower coils extending out to the rim. This form of spring is very flexible, and at the same time assists materially in keeping the valve in proper shape.

The Kinghorn Valve, Fig. 27, which is without superior for high class air pump service is generally used for Vertical Simplex Beam, Twin Beam, Twinplex Beam, Vertical Single-acting and Vertical Double-acting Air Pumps, pp. 1138 to 1140. It consists of three bronze disks about $\frac{1}{32}$ in. thick placed loosely together, making a strong, flexible and long-lived unit which will seat absolutely tight and adapt itself to conditions as they are found. A conical spring holds the valve on its seat, and a broad guide protects it on the rise. This type of valve can be used also on any of the many types of water pumps, and although this will increase first cost, it is highly recommended for efficiency and durability.

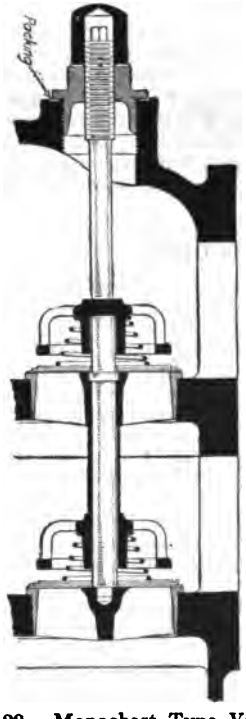


FIG. 29.—Monochest Type Valve Assembly.

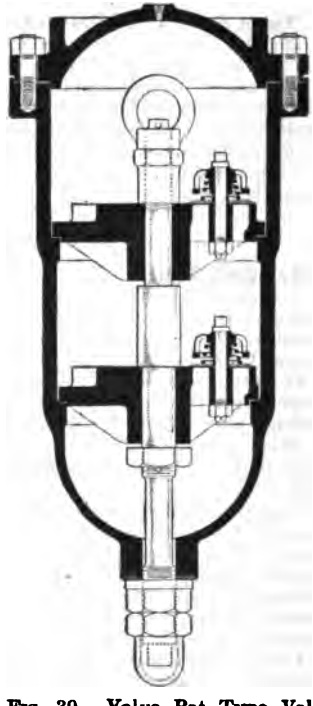


FIG. 30.—Valve Pot Type Valve Assembly.

Turbine or Rotating Valve, Fig. 28, is a special type that has met with some favor for use with rubber valves. The distinctive feature of this service is the partial rotation of the valve as it lifts from its seat; this results in even distribution of wear on valve face. Rotation is accomplished by inclining the valve seat ribs so that water passing through will give the valve a small rotary movement each pump impulse. The valve spring guard is of such design as to form with the valve stem a chamber which acts to eliminate any friction tending to interfere with rotation.

Valve of Kinghorn type may be used with this service. The disadvantage of the rotating valve is the increased number of parts to keep in adjustment and the fact that this valve is not standard with manufacturers and therefore repair parts must be made to order with attendant delay.

"Monochest" Type Valve Assembly is shown in Fig. 29. In this equipment suction and discharge valve seats, springs and bolts are all in line and held in position by binding screws and caps accessible from outside the cylinder. This arrangement prevents either valve, seat or spring from getting adrift which is possible with ordinary valve service where very hot water is pumped, due to unequal expansion between cast iron valve deck and bronze seat.

"Valve Pot" Type Valve Assembly is shown in Fig. 30. Valves are arranged in nests on suction and discharge valve plates, the complete equipment for each pot being carried on one central bolt which passes through the lower end of the pot and is secured by a gasketed, flanged and capped nut. The whole service for each pot is removed as a unit through upper cover of chamber; every part is quickly and easily accessible. Locking nut may be on cover of pot instead of below as shown. This type assembly gives the same assurance against valves, bolts, etc., getting adrift as does the Monochest Type.

Steam End and Valve Gear

Davidson Type, Fig. 31. The admission of steam to cylinder is controlled by a main steam valve V operated by valve pistons P which are balanced by steam admitted to ends of chests through small holes S.

Steam to operate pump is admitted to cylinder alternately through ports M and M and exhausted through exhaust chamber in the usual manner.

From the crosshead by means of connecting rod C, side lever L and auxiliary valve driver D, the movement of the pump oscillates auxiliary valve A which through ports registering with small ports v and v', exhausts from either end of chest, causing the main valve V to throw, in which operation it covers small port v and cushions on steam in end of chest, thus preventing striking of head.

Keep the auxiliary valve A a good "finger tight" fit in its housing. Taper pin is provided to set out the valve and it should be ground to fit if necessary. Do not allow this valve to become loose enough to permit leakage of steam between the ports.

The valve driver is packed with a lead gasket and this should be kept in good condition. Do not oil either the main or auxiliary valves as the valve motion is designed to operate without lubrication.

The quadrant, side lever and valve driver are marked showing their proper positions. If pump strokes unevenly, loosen bolt securing quadrant to side lever and adjust by shifting quadrant to make the stroke central, as indicated by the movement of crosshead arm between notches on the tie bar.

Worthington Vertical Simplex Type, Fig. 32. Main slide valve (1) controls admission of steam to the main steam cylinder (5). A steam-thrown plunger (3) operates the slide valve (1). A small auxiliary slide valve (2) controls the motion of the steam-thrown plunger (3). These three parts, together with the valve rod connections (6 and 7) to the lever, which move the auxiliary valve (2), make up the working mechanism. Referring to Fig. 30, main piston (29) is shown at mid-stroke, while making its downward stroke. Auxiliary valve (2), through valve connections (6 and 7), has just been moved

to the upper end of its stroke, admitting steam through port (21) and passage (17) to right end of auxiliary cylinder (4), at the same time connecting left end of (4) to exhaust by port (22) and passage (18). This has forced auxiliary plunger (3) with attached main valve (1) to the left-hand end of its movement.

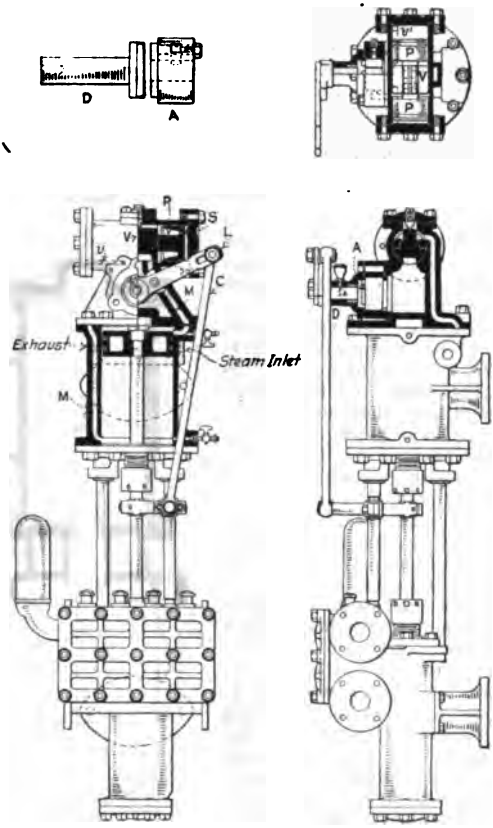


FIG. 31.—Davidson Steam Valve Gear.

The main valve has opened steam passage to port (14) leading to upper end of main cylinder and exhaust passage to port (12) leading from lower end. Steam is entering cylinder (5) through steam pipe opening (25), filling steam spaces of auxiliary cylinder (4) and upper end of cylinder through passage (14), while at the same time exhaust steam from the lower end is passing out at main exhaust opening (28) through connection (12), exhaust cavities (30) and (13).

As the main piston nears the end of its stroke, lever (8) actuated by crosshead (31) causes tappet (9), which it carries, to strike collar (10) and push down link (7) and rod (6), to which it is attached, until auxiliary valve (2) fastened on valve rod, opens port (22) for steam admission and connects port (21) with exhaust. Steam is now admitted to the left of auxiliary plunger (3) through ports (22 and 18) while at the same time steam is exhausted from the right of plunger through ports (17 and 21), cavity (33) and port (30). Auxiliary plunger (3), with main valve (1) attached, now moves to the right until port (21) is closed to exhaust cavity (33) by the movement of main valve across it. Port (21) being now closed, the plunger cushions on the steam confined between its end and auxiliary cylinder head coming to rest with main valve in proper position for the return stroke of the main piston.

For pumps having steam cylinders 10 in. diam. and less, ports (12 and 14) open into the cylinder at a short distance from the ends, and tapering V passages extend from these openings to the face of the cylinder heads. As piston (29) covers port opening at the beginning of its stroke, steam can only enter or leave cylinder through the small tapering V extension passage. This promotes shockless starting and reversal of piston motion and allows pump valves to seat quietly.

For pumps having steam cylinders over 10 in. diameter, the main steam cylinder (5) is arranged with two ports leading, side by side, to each end of the cylinder—one of small area extending to the extreme end, called the "starting port," and the other of larger sectional area extending to a point a little distant from the end, called the "main port." These two ports are cross-connected by a small relief opening. The "starting port" functions in a similar manner to the V passage in the smaller types of pumps.

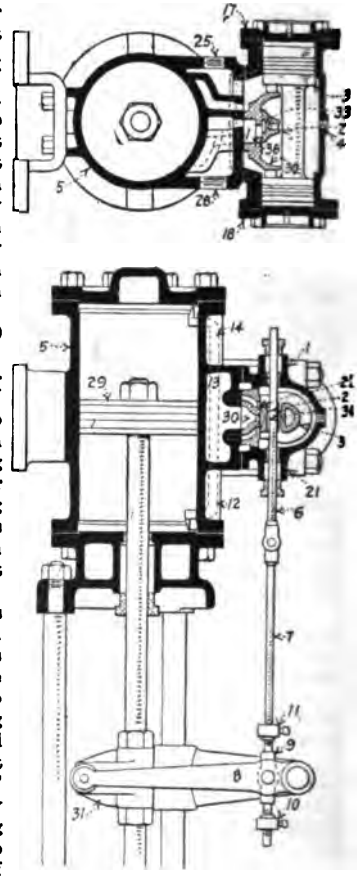


FIG. 32.—Worthington Simplex Steam Valve Gear.

The main piston length of stroke is controlled by movable screw collars (10 and 11) on valve rod link (7) against which tappet (9) carried by lever (8) strikes, and these may be adjusted while pump is in motion. To lengthen stroke, it is only necessary to screw collars farther apart, while to shorten stroke, collars are, of course, moved nearer together. Stroke should be made as long as possible, taking care that piston does not strike cylinder heads, as

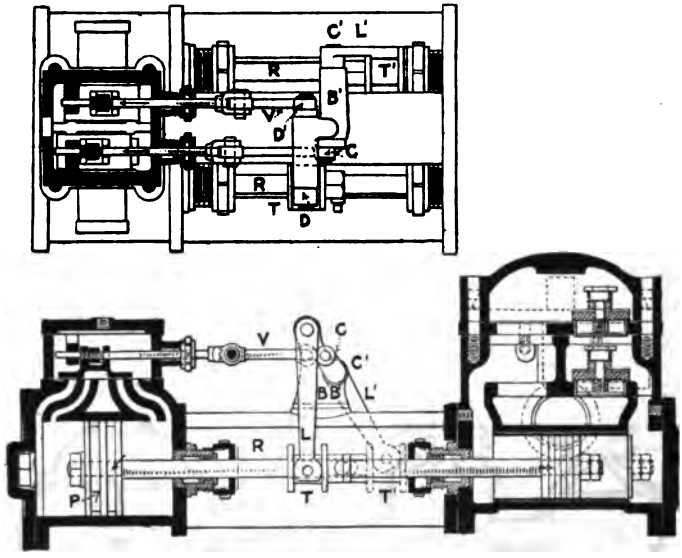


FIG. 33.—Duplex Steam Valve Gear, Inside Adjustment.

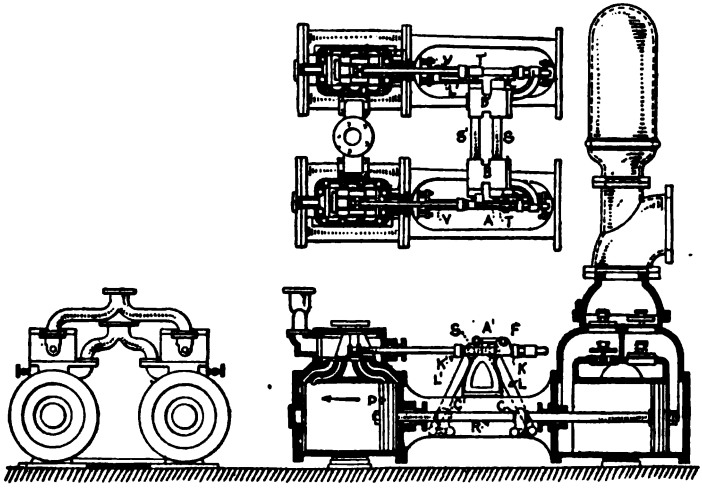


FIG. 34.—Duplex Steam Valve Gear, Outside Adjustment.

this reduces clearance spaces in cylinder and lessens amount of steam wasted at each stroke.

The interior operating parts of the steam end are so designed that they can in most cases be run with perfect satisfaction without any lubrication. A tapped boss, however, is provided on the auxiliary cylinder, to which a cylinder lubricator may be attached if interior lubrication should be preferred; this lubricator supplies oil to both auxiliary and main cylinders, and is the only supply needed for all interior lubrication. A good quality mineral oil

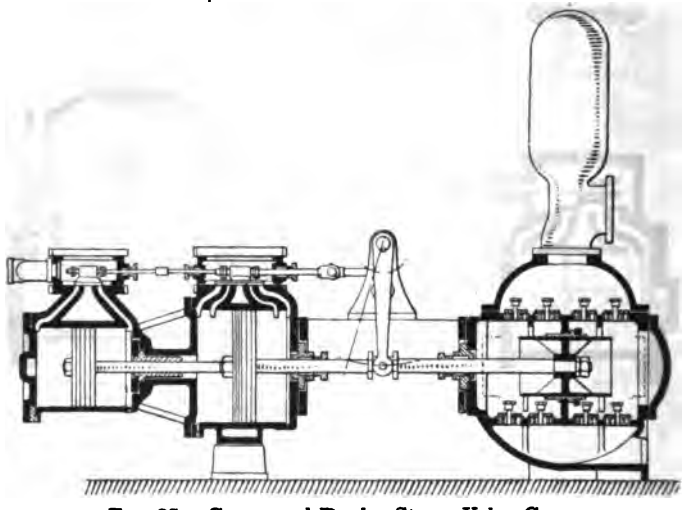


FIG. 35.—Compound Duplex Steam Valve Gear.

should be used for this purpose, and it should be fed constantly, but sparingly; if supplied too freely, it may collect in the ports and passages and tend to gum up the parts generally.

Duplex Steam Valve Gear is of three general types: Fig. 33, gear with **inside adjustment**; Fig. 34, gear with **outside adjustment**; and Fig. 35, **compound duplex.** All pumps have valves set in proper running adjustment when shipped by reliable manufacturers. In duplex valve gear the valve of one steam piston is driven by the pump rod on the opposite side. Referring to Fig. 33, pump rod R carries crosshead T which actuates bell crank consisting of long lever L and upper crank and shaft DD' which in turn actuates the valve stem V. In a similar manner the opposite valve stem is actuated off pump rod R' through crosshead T', long lever L' and lower crank and shaft CC'.

Installation and Care of Pumps

Alignment. Pumps should be set positively in line on a stiff foundation. Particular care should be taken with long stroke vertical pumps which have tie bar connections. When set, to test alignment, remove glands from both piston rod stuffing boxes, move piston rod to one end of stroke, try glands in both boxes. Remove glands, move piston rod to opposite end of stroke and try the glands again. If any sign of binding exists make proper adjustment in setting of pump.

Location. The pump should be located in the ship as close to its work as conditions will conveniently allow, with as short and direct suction and discharge pipe connections as circumstances will permit. **An air pump should be located** lower than the condenser so that condensate will drain directly into pump cylinders. Provide a proper space around pump so that all parts requiring inspection or adjustment are conveniently accessible.

Pump Connections. Always connect pump so as to secure a full and uniform supply of liquid to be handled. The extreme theoretical height to which water can be lifted by atmospheric pressure alone is 33.9 feet. In practice this seldom exceeds 25 feet; the smaller the suction lift can be made, the better. Suction pipe should be as short as possible and never smaller than opening on pump. As there is a considerable loss of head due to friction, the diameter of long lines of pipe should be increased to allow for this, and for the same reason bends of large radius and as few in number as possible should be used.

The same remarks apply to discharge pipe, although not with the same force, since in the discharge pipe the full power of the pump is always available to force the liquid through the piping, while in the case of the suction pipe, only the atmospheric pressure is available to force the liquid into the pump. Avoid all air leaks in air pump connections; otherwise vacuum will be impaired.

All valves in the suction and discharge piping should be gate valves, and at least as great in opening as the area of the pipe. Globe valves offer too much resistance to the flow of the liquid.

Steam Connections. Steam and exhaust pipe connections should always be made with due allowance for expansion of steam pipe, and of ample size—never less than the opening on the pump.

A throttle valve should be placed in the steam pipe as close to the pump as possible, and a drip cock or bleeder valve should be provided for draining the main steam pipe before starting the pump.

Blow out the steam pipe thoroughly before connecting the pump. Any dirt or chips carried into the steam cylinder will cut and injure it seriously.

Air Chambers. A suction air chamber placed on the suction pipe close by the pump is distinctly recommended for fire pumps with high suction lift, short stroke pumps and pumps running at high speed. Care should be taken to locate this suction chamber in a continuation of the line of flow in the suction pipe so as to receive the impact of the water column and thus cushion the pulsations in the most efficient manner. (See p. 304, Fig.20.)

A discharge air chamber should also be provided on the discharge of a heavy pressure pump in a continuation of the line of discharge pipe; in exceptional cases where the pump is subject to very heavy duty, it is an advantage to provide an air pet cock on the suction pipe close to the pump. By opening this pet cock slightly, a small amount of air may be admitted to the pump with the water, and the discharge air chamber kept supplied with the proper quantity to cushion the outflowing water.

Lubrication. All bearings, joints of moving parts and piston rods should be lubricated before starting pumps, and also at short intervals when in operation. All grease cups should be packed with Albany grease. Where bearings are divided, keep same adjusted with shims. For lubrication of steam end see p. 1160.

Packing. Keep stuffing boxes well filled with a good quality of packing, but do not screw up glands too tight, and do not allow same packing to remain in stuffing boxes long enough to become so hard that it will score the piston rods.

Both main and auxiliary **steam pistons are packed** by means of carefully fitted cast iron spring packing rings, which are self-adjusting and need no attention whatever; replacement on account of wear is necessary only after many years of service.

When **water pistons are packed** with fibrous packing, trouble sometimes arises from swelling of the packing, causing the pump to operate stiffly and make uneven strokes, etc. This is especially so when pumping hot liquids, and sometimes it is necessary to take out the packing and thin it down. This is done by stripping a layer from one side of the strand or ring, or hammering it if packing is of the braided type, as the swelling is usually lateral. In fitting a piston with new packing, it is well to soak the packing in warm water over night before fitting it. New packing should not be crowded into the piston, but should fit the packing space loosely. When it becomes wet the swelling will cause it to fit in the space.

Air pump buckets are packed with rings of a good quality, thoroughly tallowed, flax packing. Renewal or adjustment of packing can be accomplished without removing bucket from air cylinder, it only being necessary to unbolt cylinder head and head valve plate, slide them up on bucket rod and block them so as to allow working space between heads and cylinder top. Bucket follower can then be unbolted and raised. This exposes packing groove, and adjustment or renewal can be effected.

Packing rings should be carefully cut to length, each length being a little short of the exact measure to allow for the extension which is bound to occur after packing is wet. If packing is not thoroughly filled with tallow or grease, it may be well to soak it in warm water over night before fitting to bucket. Packing must be an easy fit in groove, as it will surely swell and completely fill the space. If pump works stiffly after being newly packed, it is probably due to over-swelling of packing, and it may be necessary to take out packing and hammer it a little so it will fit more loosely in space. If, in time, packing becomes so worn that considerable leakage past bucket occurs, strips of sheet brass or copper of proper thickness may be laid in the groove behind packing so that it will not be necessary to replace packing until it is very badly worn, or becomes too hard and dry to be effective.

Pump Air Bound. Hot water cannot be raised by suction, as it vaporises when atmospheric pressure is removed, and the vapor is alternately expanded and compressed in pump cylinders without being expelled. The pump must be located so that the water will flow to suction opening of the pump under a sufficient head to insure the water cylinders always being completely filled with water, and not with a mixture of water and vapor. Where the temperature of water approaches the boiling point, the head on the suction opening at the pump should be from ten to fifteen feet.

Suction pipes and connections must be perfectly air-tight, as a very small leak will prevent the proper working of the pump and impair its efficiency. For test pressure for suction pipes see Section 15, p, 1430.

A pump may fail to lift water while the full pressure against which it is expected to work is resting on the discharge valves, because the air in the pump cylinder is not discharged but is only compressed by the motion of the pistons. In such a case the pump should be run without pressure until the air is expelled from the pump and suction pipe. This is done by providing a check valve in the discharge pipe near the pump and opening the air vent which is provided on the discharge cap of the pump.

Laying Up. When pump is to be kept out of commission for any considerable time, slush interior of steam chest with cylinder oil to prevent corrosion. If steam end is fitted with lubricator, fill this with oil and open cock so oil can flow to steam chest. A few quick pump strokes will distribute oil over steam end of pump.

CENTRIFUGAL FANS

BY

LOUIS LANYI AND RAINES KESSLER

Nomenclature:

A = area in sq. ft.
B = barometer in in. of mercury.
C = tip speed in ft. per sec.
E = efficiency.
E_m = manometric efficiency.
E_v = volumetric efficiency.
g = acceleration due to gravity, 32.2 ft. per sec. per sec.
H = head of air in ft.
h = inches water gage.
h_s = inches water gage static.
h_t = inches water gage total or impact.
h_v = inches water gage velocity.
O_e = equivalent orifice.
P = pressure in lbs. per sq. ft.
q = quantity of air in cu. ft. per sec.
Q = quantity of air in cu. ft. per min.
t = temperature in deg. F.
v = velocity in feet per sec.
V = velocity in feet per min.
W = weight of air moved per min
$a.h.p.$ = air horsepower.
$b.h.p.$ = brake horsepower.
$c.f.m.$ = cubic feet per minute.

A centrifugal fan is one which produces a flow of air or gas by means of a rapidly rotating impeller usually placed inside of a suitably designed casing or housing which may be conveniently connected to a duct system. **Fans are usually classified** as blowers and exhausters accordingly as they do their useful work on the discharge or suction side. The following classification shows the customary application of fans in marine work.

		Service	Type	
Fans	Blowers	Ventilating—	housed	Fig. 1
		Forced draft—	housed and open	Figs. 2, 3 and 4
	Exhausters	Ventilating—	housed	Fig. 1
		Induced draft—	housed	Fig. 2

Ventilating fans sometimes act as both exhausters and blowers drawing air over steam coils and blowing the heated air into a duct system; see p. 1189.

The performance of fans is expressed in cubic feet of air per minute discharged and static pressure or suction produced expressed ir inches water gage. Some manufacturers still give pressures in ounces per sq. in. One ounce per sq. in. is equivalent to 1.73 in. water gage.

Electric motor driven **heating and ventilating fans** such as shown in Fig. 1 are of the multivane type and common to all types of ships. They produce maximum pressures of $2\frac{1}{2}$ in. static.

Fans such as shown in Fig. 2 are used in the **merchant marine for forced and induced draft.** They are either of the steel plate or multivane type and operate at low pressures. For forced draft work they operate between

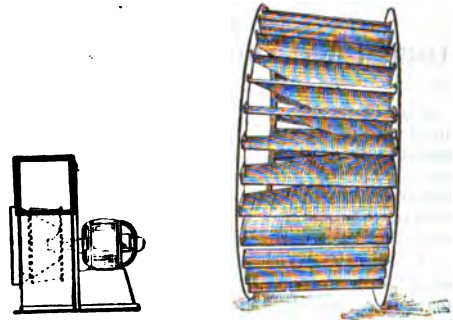


FIG. 1.—Motor Driven, Multivane, Heating and Ventilating Fan.

1 in. and 3 in. static water gage—for induced draft work they produce suctions ranging from $\frac{1}{2}$ in. to $1\frac{1}{2}$ in. static water gage with 600 deg. to 700 deg. F. gas temperatures.

Forced **draft blowers** as shown in Figs. 3 and 4 are used **on Naval vessels.** They produce pressures as high as 8 in. static usually discharging into a closed stokehold. Such blowers are usually turbine driven.

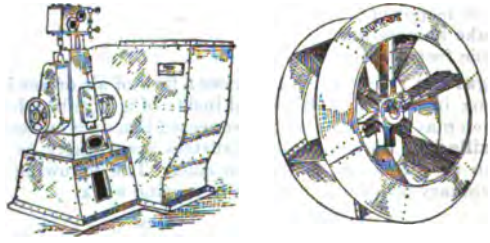


FIG. 2.—Engine Driven, Steel Plate, Forced and Induced Draft Fan. (For Blades, see Fig. 10.)

The open type fans are always provided with **diffusor plates** which materially improve the efficiency. These plates are either parallel or inclined toward each other and about 8 in. in radial depth; see Fig. 5. In installing open type fans ample free space must be allowed, about the diffusor plates, as the wheels have a large air handling capacity and unless the discharge from the diffusor is free the efficiency will be very materially affected and hunting may be set up where two or more fans discharge into the same room.

All cased high pressure blowers in naval vessels are provided with **balanced dampers** so that in case one blower shuts down the damper closes automatically preventing a sudden loss of pressure in the boiler room and back firing of the oil burners which would occur if the pressure were suddenly reduced.

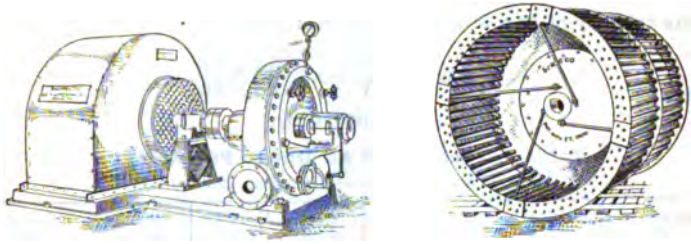


FIG. 3.—Turbine Driven, Direct Connected, Forced Draft Fan (For Blades, see Fig. 11.)

Static pressure is pressure energy—a form of potential energy.
Velocity pressure represents kinetic energy.
Total or impact pressure is the sum of the static and velocity pressures.

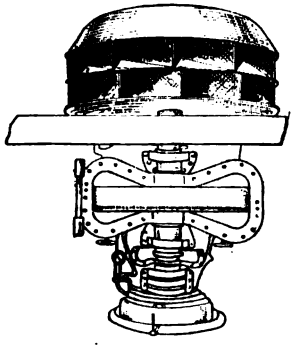


FIG. 4.—Vertical Turbine Driven, Forced Draft Fan.

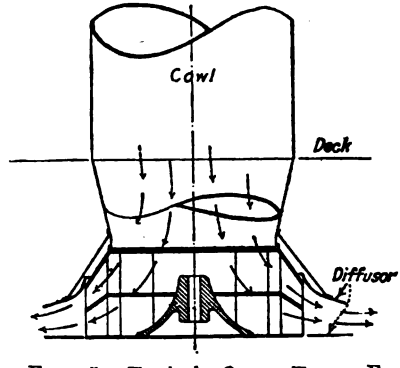


FIG. 5.—Typical Open Type Fan, Showing Diffusor.

Pressure and velocity energy are mutally convertible under ideal conditions. This can be shown in the same way for air as in the case of water or steam. With a perfect nozzle all of the pressure energy could be converted into velocity energy and conversely velocity energy could be completely converted into pressure energy. In every duct system (fire room, burner, boiler, breechings and stack) we have several large and sudden changes of area and consequently also of air velocity from the time the air leaves the blower until it reaches the free air. Each change of velocity is imperfect and is accompanied by a severe loss due to shock. These losses all occur at the expense of static pressure.

In selecting fan equipment it is desirable therefore to secure the maximum per cent of the energy in static pressure at the fan discharge and to keep duct velocities as uniform as possible. This statement is strictly true for closed stokehold installations. In cases where the blower is connected to ducts a certain proportion of velocity energy is useful depending upon the size of the ducts.

The velocity of flow is derived from the formula

$$v = \sqrt{2gH} = \sqrt{2g \times \frac{62.3}{0.075} \times \frac{h_v}{12}}$$

$$V = 4000\sqrt{h_v}$$

Table 1. Velocity of Air Due to Pressure

(Air at 65 deg. F.; barometer reading, 29.92 in.)

Pressure, in. of water	Velocity, ft. per min.	Pressure, in. of water	Velocity, ft. per min.	Pressure, in. of water	Velocity, ft. per min.	Pressure, in. of water	Velocity, ft. per min.
0.1	1,265	0.9	3,790	1.7	5,210	3.25	7,210
0.2	1,790	1.0	4,000	1.8	5,360	3.50	7,490
0.3	2,190	1.1	4,190	1.9	5,510	3.75	7,750
0.4	2,530	1.2	4,380	2.00	5,650	4.00	8,000
0.5	2,830	1.3	4,560	2.25	6,000	4.25	8,250
0.6	3,100	1.4	4,730	2.50	6,320	4.50	8,490
0.7	3,350	1.5	4,900	2.75	6,630	4.75	8,720
0.8	3,580	1.6	5,050	3.00	6,930	5.00	8,950

For commercial purposes **the weight of standard air** (70 deg. F., 30 in. barometer and 70 per cent relative humidity) is taken as 0.075 pound per cubic foot. The actual weight per cubic foot under standard conditions as given above is 0.07465 pound per cubic foot.

For any given condition of temperature and barometer (disregarding the small correction for humidity) $V = 956\sqrt{\frac{(460 + t)h_v}{B}}$.

Pressure or Draft is measured in inches of water column above or below atmosphere. Glass "U" tubes are employed for this purpose. They are partly filled with water and have a properly graduated scale arranged to read directly in inches. If the gauge is in the closed stoke hold then one branch is piped to the free air in order to secure a differential pressure. If the air pressure within a pipe or duct is desired one branch of the gauge is connected to a small pipe introduced into the duct in such a way that the measuring orifice lies along a stream line of a parallel flow. The pressure then obtained is the static pressure. Oil is sometimes used in gauges because it has a good meniscus but when used graduations are arranged to read directly in inches; see Peabody Draft Gage, p. 345.

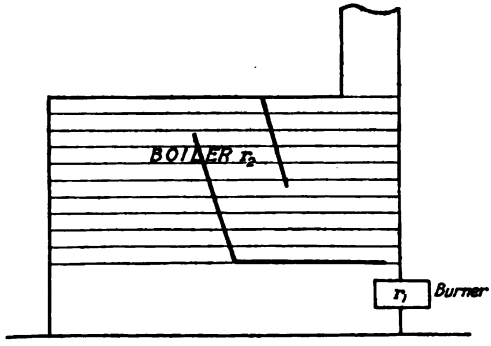
Air horsepower is defined by the following equation:

$$a.h.p. = \frac{W \times H}{33{,}000} \text{ and since } W = Q \times 0.075 \text{ and } H = \frac{P}{0.075}$$

$$\text{a.h.p.} = \frac{Q \times P}{33{,}000} = 0.000188 \times h_t \times \text{c.f.m.}$$

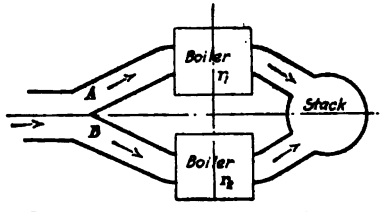
The inches water to be used in the above formula will be total or static depending upon whether total or static efficiency is used in computing the brake horsepower input.

Efficiency of a blower is the ratio of the air horsepower output and brake horsepower input. If the total pressure has been used in computing the *a.h.p.* the **total efficiency** is obtained. If the static pressure has been used then the **static efficiency** is obtained.



Let R = Total resistance of system
r_1 = resistance of burner
r_2 = resistance of boiler
then $R = r_1 + r_2$

FIG. 6.—Resistance in Series.



Let Q = Total cu. ft. per min. passing through system
Q_1 = " " " " " " " " duct A
Q_2 = " " " " " " " " duct B
H = total head necessary to pass Q through system
$r_1 + r_2$ = resistance of boilers
R = resistance of complete system
$H = r_1 Q_1^2$
$H = r_2 Q_2^2$
$H = R Q^2$

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{r_1}} + \frac{1}{\sqrt{r_2}}.$$

FIG. 7.—Resistance in Parallel.

Equivalent orifice, Oe $= \dfrac{q}{K\sqrt{2gH}}$, is the area of a circular hole in a thin plate whose restricting effect upon the flow of air is the same as that of the duct system regardless of the nature of the individual resistances. The value K has been found to be nearly 0.65 in close accord with the value for K

in the case of water. Nothing in this expression bears upon the size of the fan so that equivalent orifice has little value to the commercial engineer. The conception of equivalent orifice helps to visualize fluid flow when the resistances of a system are changed and will be used in explaining the operation of fans in parallel.

Resistances in Series. If the same quantity of air flows through a number of resistances in series the resistance of the path equals the sum of the individual resistances. Adding resistances in series is in effect a reduction of the equivalent orifice. See Fig. 6.

If a quantity of air flows from a region of higher pressure to one of lower pressure through a number of different channels simultaneously (each having equal or different resistance) the drop in head over each path will be the same

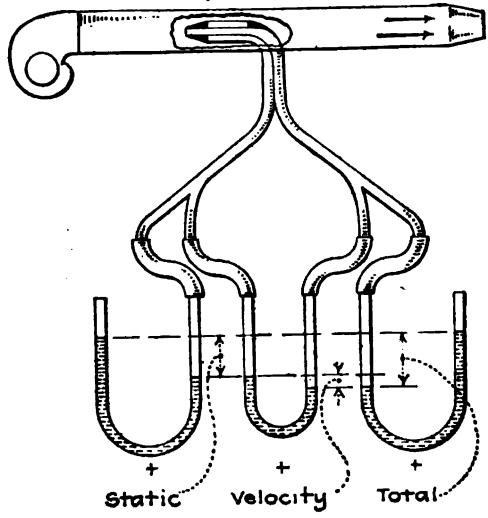


FIG. 8.—Gages Placed on Discharge Side of Fan.

and the equivalent orifice of the system of channels will be equal to the sum of the equivalent orifices of the individual paths Adding restrictions in parallel is in effect an increase of equivalent orifice. See Fig. 7.

Manometric Efficiency (pressure efficiency) is the ratio of the pressure developed by a fan and the pressure against a plane surface due to a velocity equal to the peripheral speed of the wheel; $E_m = \frac{gH}{C^2}$. This term has little real meaning for the commercial engineer and is not used in other than design problems.

Volumetric Efficiency, $E_v = \frac{Q}{\text{R.P.M.} \times (\text{content of wheel in cu. ft.})}$ is the ratio of volume of air delivered per revolution divided by the cubical content of the wheel. This number is not an efficiency and can be in excess

of 100 per cent. It was largely used in Europe as a means of comparing fan designs.

The effect of speed upon blower performance if the equivalent orifice (duct condition) does not change but the speed of the fan is changed is as follows: Q is directly proportional to the speed; H is proportional to the square of the speed; $a.h.p.$ is proportional to the cube of the speed; efficiency remains constant.

On the discharge side of a blower the impact or total pressure reading is always greater than the static and equal to the sum of the static and velocity pressures.

On the suction side the static suction is always a greater negative reading than the impact. This is due to the fact that the flow is induced by the static suction.

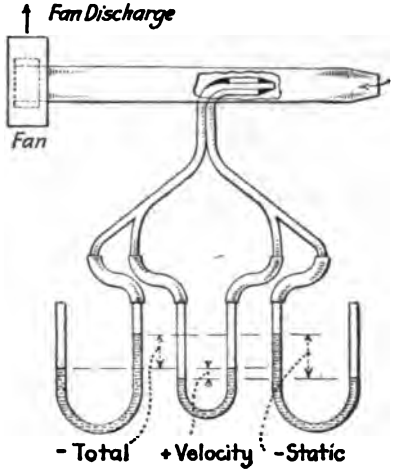


FIG. 9.—Gages Placed on Section Side of Fan.

Effect of Density Upon Fan Performance. If the fan operates at constant speed, then (1) the *volume* delivered remains constant irrespective of change of density due to change of barometer or temperature; (2) the weight flow is affected by density and is directly proportional to it; (3) the head produced in feet of air remains constant irrespective of change of density but the pressure in inches of water changes for a change in density; (4) the brake horsepower changes with the density.

The effect of temperature upon fan performance is to change the density of the air. The inches of water, pressure or suction, is inversely proportional to the absolute temperature with a constant barometer. Knowing the inches of water a fan produces on 70 deg. F. air with a given barometer then the inches of water at temperature t is given by the following formula

$$h_t = h_{70} \frac{460 + 70}{460 + t}$$

In the case of induced draft problems it is most convenient to work on a cold air basis since usually operation at several temperatures is required. For any temperature

$$a.h.p. = 0.000158 \times inches\ water \times c.f.m.$$

Example. Find brake horsepower required to drive a fan for induced draft capable of handling 50,000 lbs. air per hr. at 600 deg. F. and producing 2 in. suction with a 30 in. barometer and 60 per cent. static efficiency.

$$\frac{50{,}000}{60 \times 0.0375} = 22{,}250\ c.f.m.$$

$$\frac{2 \times 22{,}250 \times 0.000158}{0.60} = 11.7\ b.h.p.$$

If this blower were starting with cold air the volume at the same speed as above would be the same, the weight of air handled would double and the inches of water and horsepower would double. When electric motor driven blowers are used for induced draft this condition must be provided for either by the use of sufficiently large motors or suitably arranged dampers which can be opened gradually as the gas temperature rises.

NOTE: For the low pressures of forced and induced draft work the air or gas is assumed as being incompressible.

Types of Fan Impellers

Two distinct types of fan impellers are usually recognized—the steel plate and the multivane. Each of these are frequently built with modified blade forms producing distinct changes in operating characteristic. Fig. 10 shows a typical **steel plate impeller,** Figs. 11, 12 and 13 show the most frequently encountered forms of **multivane impellers.**

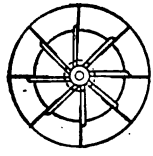


FIG. 10.

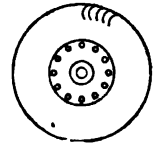


FIG. 11.

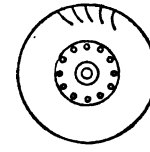


FIG. 12.

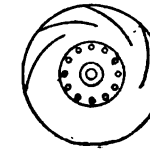


FIG. 13.

FIG. 10.—Steel Plate Impeller. Has From 8 to 10 Blades.

FIG. 11.—Forwardly Curved Multivane Impeller. Has From 30 to 60 Blades.

FIG. 12.—Radial Tip Multivane Impeller. Has from 30 to 60 Blades.

FIG. 13.—Backward Curved Multivane Impeller. Usually 12 to 20 Blades.

The change in characteristic produced by the various types of blading as shown in Figs. 10 to 13 is reflected in the pressure-volume curve at constant speed and also in the efficiency and horsepower curves. This is discussed in detail later. For usual arrangement of blower for forced or induced draft see pp. 1164 and 1165.

In **merchant marine installations** one fan usually serves one or more boilers connected to the same duct. Any properly designed fan under such conditions will work well. In **naval installations** several fans usually discharge into the fire room. Special care must be exercised in selecting equipment which will run in parallel without hunting or blowing back.

In selecting a fan it must be borne in mind that fans, like all centrifugal machinery, have a wide field of utility; in other words with a fan operating at a constant speed arranged with a pipe at the discharge and a damper in the

pipe it is quite possible to obtain various quantities of air discharged with varying pressures at the fan discharge accompanied by varying power input to the fan. This result can be secured by merely changing the damper in the duct. However, the efficiencies at the various points obtained would be different so that there would be only one point at which the fan would operate at maximum efficiency. The efficiency curve has a maximum point and for a considerable volume range at constant speed it is fairly flat. If then a minimum efficiency is assumed as commercially permissible the utility of the fan extends over a considerable portion of the constant speed curve.

Limits on Maximum Output. Assuming in Fig. 14 that a and a' are the minimum commercial efficiencies then the field of usefulness lies between these points. This is the limit placed upon capacity of the fan by **efficiency.**

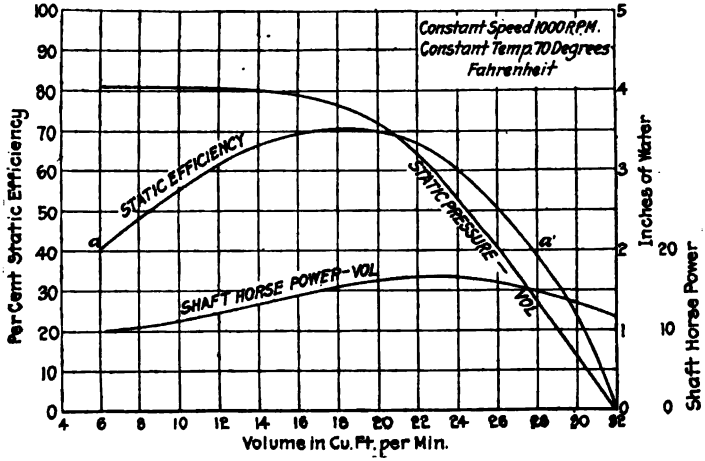


FIG. 14.

Another limiting feature of the volume which a fan can produce is the **entrance and discharge velocity.** If those velocities become very great a considerable amount of energy will be lost. If fan ducts are poorly designed, as they usually are, these very high velocities will be dissipated in the form of heat and fail to produce a useful result. Modern fans operate with discharge velocities ranging from 1500 ft. to 6000 ft. per min. 2500 ft. per min. or less would be a fair discharge velocity for the low pressure fans. Velocities from 2400 to 3600 ft. per min. are usually encountered in Navy work. A third limiting feature of fan capacity is the **strength of the wheel.** The above gives the three principle limitations in securing the maximum output of a given size fan. Therefore, the utility of a fan lies at any optional point within these limitations. It is now easy to see that a fan suitably selected can operate properly under varying conditions on the discharge side; brought about, for instance, by different thicknesses of fire bed, different positions of dampers, accumulations of dust on boiler tubes or in boiler flues.

The terms "full-load" and half-load" as applied to a fan are very frequently encountered. These terms are meaningless. "**Load**" must be

defined as power supplied to or delivered by a machine. If in the case of a fan the duct conditions are not altered the power supplied at half speed is only ⅛ that consumed at full speed as the power required by a fan is proportional to the cube of the speed.

The well known **fan laws** are: (1) the volume delivered is proportional to the speed; (2) the pressure developed is proportional to the square of the speed; (3) the power consumed is proportional to the cube of the speed.

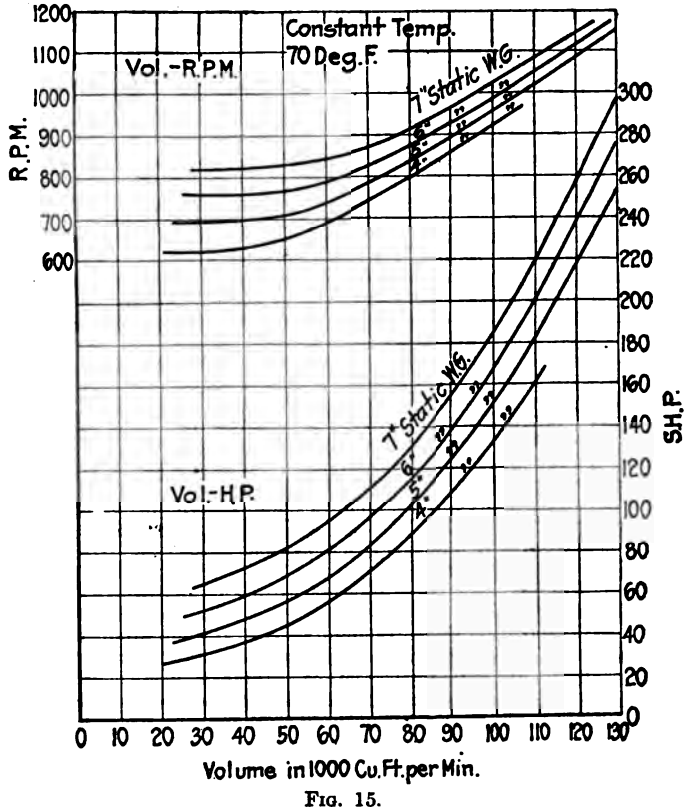


FIG. 15.

However, all of the above mentioned laws presuppose that the orifice condition remains unaltered. If the orifice condition varies then the application of these laws will not give a correct result. For instance, if the speed were varied and a damper setting changed at the same time, the fan laws as stated above could not be used.

Fans at constant speed can deliver their rated capacity only as long as the equivalent orifice remains constant. If the equivalent orifice changes the

fan speed must be changed accordingly in order to deliver the same volume of air. It is evident then that if a fan is started with a fire under a boiler with clean tubes and clean flues and operated until the boiler becomes full of dust and fires are no longer clean, it will be necessary to either clean fires or increase the fan speed in order to pass the same mass of air through the fire and retain the same rate of combustion. This will result in an increase in pressure at the fan. In other words, it is evident that a fan can produce the same volume at different pressures and in the same way it can be reasoned that a fan can produce different volumes at the same pressure. This leads to the conclusion that the fan characteristic and orifice characteristic must cross if it were possible to plot them.

Fig. 15 shows variations for any constant speed, any constant pressure, any constant volume or any constant horsepower.

This leads to the only correct conclusion: The volume and pressure produced by a fan connected to a duct is governed by the resistance of the duct.

It is interesting to note at this point and naturally follows from the fan laws enumerated above that the characteristic of the orifice is such that the velocity through it is proportional to the volumes, that the pressure is proportional to the square of the volume passing through, therefore, knowing

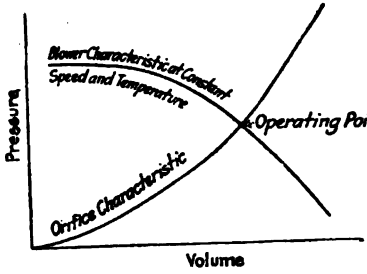


FIG. 16.

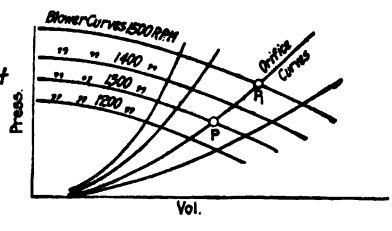


FIG. 17.

the pressure necessary to pass a given volume through the orifice you can readily determine the pressure necessary to force a different volume through. With a variable orifice, such as described above, and operating the fan so that its speed can be varied, a set of characteristics for the various orifice conditions can be plotted. To the same scale a set of blower characteristics taken at various constant speeds can also be plotted and superimposed. With these two sets of curves a method can be evolved by which the necessary fan pressure to pass a given quantity of air through the orifice can be accurately determined and also the speed at which the blower would have to operate and then by means of the efficiency curve the horsepower which the blower would require can be predicted.

If the pressure and volume conditions of one point on each orifice curve can be determined the entire curve can very easily be produced.

$$v^2 = 2gh, \frac{v^2}{h} = \frac{v_1^2}{h_1}, v_1 = v\sqrt{\frac{h_1}{h}}$$

$$v_1^2 = 2gh_1$$

If v and h are known and h_1 is the assumed new pressure solve for v_1, the new volume. As regards the blower curve this must be secured from an actual

test and is usually supplied by the manufacturer. The constant speed characteristics are similar curves and the corresponding efficiency curves are also similar so that from a known manufacturer's curve the remaining curves for the scope of the blower can easily be drawn.

The **orifice characteristic** must usually be assumed for a new installation based upon previous experience with similar installations. The maximum and minimum probable pressures assumed necessary to pass the maximum and minimum quantities of air needed will allow us to construct limits of operation and we can then select a blower which will give us the best range of blower operation.

Multivane fans can be designed to give differently shaped characteristics. The forwardly curved blade is best suited for engine speeds, the radial tip blade gives moderate speeds for motor driven apparatus and the backwardly curved blade gives the highest speeds for the same pressure in each case.

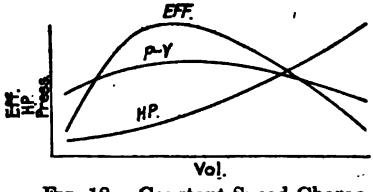


FIG. 18.—Constant Speed Characteristic for Forwardly Curved Blade Fan.

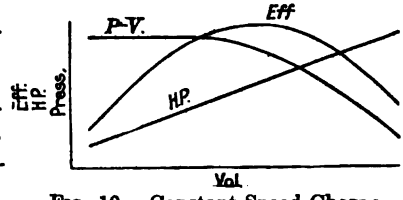


FIG. 19.—Constant Speed Characteristic for Radial Tip Blade, Multivane Fan.

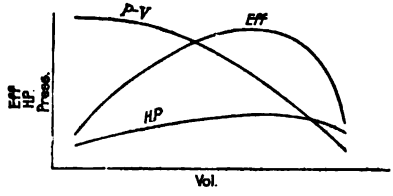


FIG. 20.—Constant Speed Characteristic of Backwardly Curved Blade, Multivane Fan.

The **forwardly curved blade characteristic** is an unstable one having a maximum point and showing a falling off in pressure from this maximum point as volumes are either increased or decreased while the speed remains constant; see Fig. 18. The horsepower curve increases very rapidly being concave up, the efficiency curve has its maximum point comparatively speaking close to the pressure axis. At the first glance it is evident if such a blower is driven with a motor that the motor must have a sufficient excess capacity so that in case the equivalent orifice had been assumed small under actual operating conditions it would permit more volume at a less pressure to pass through. This would overload the motor.

The **radial tip blade characteristic** is shown in Fig. 19. This curve is flat for a considerable distance and one very well adapted for service in con-

nection with oil fired boilers where varying volumes are required under constant pressure conditions. The horsepower curve in this case is almost a straight line and at a large volume the horsepower does not increase as rapidly as for a forwardly curved blade fan of similar capacity. The efficiency curve has its maximum point at about 50 per cent maximum volume.

The **backwardly curved** type of blade shows decidedly different **characteristics**, Fig. 20. It will be noted that the pressure volume curve is steep, that the horsepower curve reaches a maximum point which means that at a maximum permissible speed of fan a motor can be installed which will actually meet the fan requirement or if the maximum speed is infrequent it is possible to overload the motor by a predetermined amount. The efficiency curve has its maximum point toward the larger volumes. The steepness of the pressure-volume curve and the maintainence of the high efficiency at the wide open points accounts for the flatness of the horsepower curve. A blower of this type is admirably adapted for parallel operation.

Parallel Operation

The parallel operation of blowers is a very important subject when designing such installations as are found on torpedo boat destroyers with two to six blowers discharging into the same room. With varying speeds the blowers must have a steep characteristic otherwise hunting would take place and some blowers would be discharging air through the inlet while others

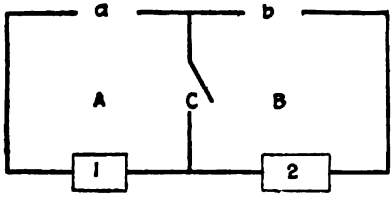


FIG. 21.

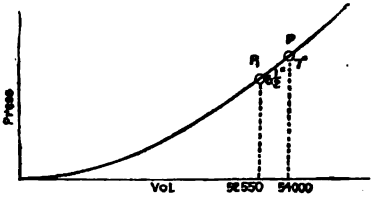


FIG. 22.

were trying to run away with consequent tripping of the emergency governor. It is, of course, very difficult to predict how much work each blower is doing under such circumstances. However, the following device will prove very useful in determining the work done by each blower under such disadvantageous conditions and will also enable the approximation of the extent of speed difference possible before blowing back will occur.

Assume two similar chambers "*A*" and "*B*," Fig. 21, and two similar blowers 1 and 2 each discharging into one of the chambers. The blower speeds are the same and their capacities exactly alike. Assume that a door "*c*," which is free to swing, is placed in the wall separating chambers "*A*" and "*B*." The orifices "*a*" and "*b*" represent the load and must be considered equal and constant for the purpose of this discussion. Let the parts be proportioned so that when a pressure of 7 in. obtains in the chambers "*A*" and "*B*" a total discharge from both orifices will be 54,000 cu. ft. per min. and the blower speed in each case will be 1400 R.P.M., the orifice characteristic will then be as in Fig. 22. Assume now that something has occurred such that No. 1 blower operates at 1400 R.P.M. and No. 2 blower at 1300 R.P.M. The problem is to find what pressure will be produced by this combination and how much air will be delivered. It is evident that as soon as No. 2 blower slows down the door "*c*" will swing open and allow the pressure in

chambers "*A*" and "*B*" to equalize; this will permit equal quantities of air to flow through openings "*a*" and "*b*."

Reference to the characteristic curve shows that each blower can deliver air against 7 in. pressure, although the blowers are operating at different speeds. No. 1 blower at 1400 R.P.M. can deliver 27,000 cu. ft. against 7 in. and No. 2 blower can deliver 22,500 cu. ft. against 7 in. This would indicate that both blowers could deliver 49,500 cu. ft. against 7 in. However, with the given system this volume would be passed with less than 7 in. and consequently the following device must be used. Select a point at 7 in. half way between 27,000 and 22,500 c.f.m., namely, at 24,750 c.f.m. and draw through this a characteristic curve between the 1300 and 1400 R.P.M. curves. Upon this characteristic curve sheet draw an orifice characteristic. These two curves will cross at the pressure under which the system will operate. When both blowers produce this pressure read the corresponding volumes on the 1300 and 1400 R.P.M. characteristics. The total air delivered is 52,000 cu. ft. per min. against a pressure of 6½ in. It will be found that this point lies on the orifice characteristic and therefore is the one at which the system must operate. What is really done in the above operation is to assume that the two blowers have been replaced with two others operating at the same speed each doing half the work. This gives an intersection which determines the operating pressure and, referring this pressure point to the actual characteristics at the speeds under consideration, the proportion in which the discharge is divided between the two units is obtained.

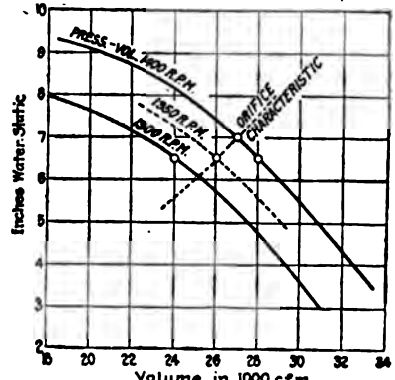


FIG. 23.

Blowers with different characteristics may under certain circumstances be made to operate in parallel but it is not advisable to do this since difficulty will undoubtedly be experienced in starting up; however, when the blowers are once brought into step they may be successfully operated. The method used above would be the method followed in designing. However, if the apparatus were once installed and respective blower speeds obtained and the pressure in the fire room were determined, it would be a simple matter to find out how much volume is passing through each blower. Simply take the characteristic curve and locate the pressure n the respective curves and immediately obtain the respective volumes.

Since the duct determines the pressure required to pass a given amount of air, ducts must be designed with a minimum resistance since then the horsepower requirement is a minimum. This resistance is kept low by using low velocities, straight ducts and long radius bends where change in direction is necessary. The manner in which connections between the ducts are made is also worthy of consideration. A stream of air in passing through an opening suffers contraction, and the coefficients are nearly similar to those which are encountered with water. Therefore in designing air ducts, precautions such as are necessary in designing water piping systems should be taken in order to avoid excessive friction losses.

HEATING AND VENTILATION, ASH HANDLING MACHINERY, MISCELLANEOUS

BY

FRANK W. STERLING

HEATING

The agents available for marine heating are steam, electricity and air; the most common in marine service is steam; electricity is sometimes used for isolated compartments to avoid long steam leads; air is used in connection with steam in the indirect system of heating. **Steam has the advantage** that it may be taken from the auxiliary steam line or, for large installations, an independent line may be run; in either case it is taken through a reducing valve set at 15 to 25 lbs. in merchant practice and up to 50 lbs. for Naval work. **As a safeguard** a safety valve is installed after the reducing valve. It is not good practice to have the heating systems of cabins and living quarters dependent on each other for their steam supply; generally several lines are run to various parts of the ship when living quarters are separated, so that either may be cut in or out independent of the other.

Radiator pipes may be installed in two ways; one pipe alone may lead to a radiator, the condensed steam flowing back the same pipe or being drawn off by cocks; more commonly, two pipes are installed, one pipe leading steam to the radiator while the second pipe conducts the condensed steam to steam traps, from whence it is discharged to the main feed tanks in the engine room. Commercial types of cast iron radiators are coming into marine practice. Provision is made so that all pipes may be drained in case of repairs or when the system is laid up. Where compartments are far from the engine room, long leads of steam piping are required. In this case **electric heaters** may be installed. This is expensive and is only justified at isolated points or under special conditions.

To determine the square feet of heating surface necessary for a certain room, account is taken of the temperature to which the various sides are subjected and of the material of which the sides are composed. Allowance should also be made for direct leakage, or escape of air at the temperature of the room. This loss is probably considerably less in ships than in buildings due to the materials and methods of construction used. From tests it has been determined that, with a standard make radiator when operated with steam at 212 deg. F. and with the surrounding air at 70 deg. F., the heat transmission is at the rate of 1.57 B.t.u. per hr. for 1 deg. temp. difference. The steam condensed per sq. ft. per hr. weighs 0.23 lb. Knowing the heat loss through the various sides of the room, the **square feet of radiator surface** is given by the formula

$$\frac{\text{B.t.u. radiated and conducted away per hour}}{1.57 \times (\text{Diff. in temp. of steam in radiator and of outside air})}$$

According to good practice of leading shipbuilding companies, allowance of radiator surface is proportioned to the cubic contents of the room. The following may be taken as a typical example, but these figures are subject to considerable modification as not only the class of room but the location, the type of ship, and the service must be considered.

Captain's room, 1 sq. ft. H.S. per 45 cu. ft. of space.
Captain's bath, 1 sq. ft. H.S. per 30 cu. ft. of space.
Pantry, 1 sq. ft. per H.S. per 85 cu. ft. of space.
Crew's mess, 1 sq. ft. H.S. per 35 cu. ft. of space.
Chief engineer, 1 sq. ft. H.S. per 43 cu. ft. of space.
Spare room, 1 sq. ft. H.S. per 80 cu. ft. of space.
Steward and mess boy, 1 sq. ft. H.S. per 66 cu. ft. of space.
Officer's wash-room, 1 sq. ft. H.S. per 27 cu. ft. of space.
First and second mate, 1 sq. ft. H.S. per 33 cu. ft. of space.
Chart house, 1 sq. ft. H.S. per 44 cu. ft. of space.
Pilot house, 1 sq. ft. H.S. per 35 cu. ft. of space.
Officers' mess, 1 sq. ft. H.S. per 51 cu. ft. of space.

Newport News Shipbuilding and Dry Dock Co. Practice.

Mr. W. T. Dimm, Assistant Chief Engineer of the Newport News Shipbuilding and Dry Dock Co. describes practice at that yard as follows (see Figs. 1, 2 and 3):

"In merchant vessels for both freight and passenger vessels, steam heat by direct radiation has been our usual practice. The vessel is divided into a series of steam and drain circuits, each circuit covering a certain deck or section of the ship. The steam circuits are usually taken from a manifold at one or two central points, a reducing valve and a relief valve being installed between the steam supply pipe and the manifold. The drain circuits return to a manifold and thence through a trap to the auxiliary condenser or feed tank. The steam pressure of the heating system is usually 25 lbs. The amount of radiating surface is usually determined in a somewhat arbitrary manner based on the ratio of the cubic feet of volume of the space to the square feet of heating surface. For officers' quarters and staterooms this ratio varies from 45 to 80 depending upon the location and exposure of the space. In public spaces such as the social hall it may run as high as 120. Radiators which were formerly made of iron or brass pipe coils are now frequently of the cast iron, commercial, wall radiator type though the coil radiators are retained to some extent.

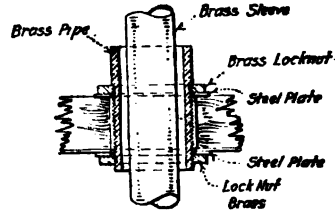


FIG. 1.

"It is customary to design heating systems on shipboard for an extreme condition of zero deg. F. outside and 70 deg. F. inside."

Standard plans for a passenger steamer built by this Company set forth the following requirements:

All pipes except expansion bends and piping in Captain's stateroom and Pilot House are of standard black wrought iron; expansion bends are of copper, screwed fittings to be brazed on. All joints are made with unions, cast iron fittings for wrought iron pipes, brass fittings for brass pipes. All pipes exposed to weather, in cargo spaces, branches to radiators and valves in living spaces are to be lagged.

Pipes passing through wooden decks are arranged as shown in Fig. 1. Locate **drain plugs** in circuits so that all pipes may be drained free from water when ship is out of commission. Pipes should be so led that they will

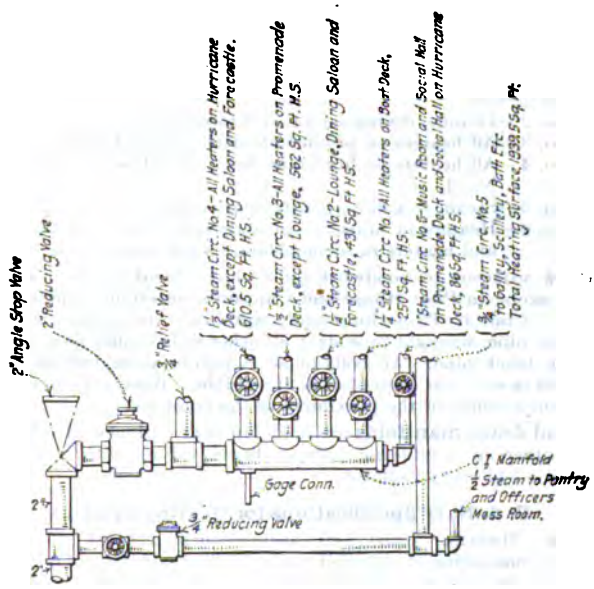

2" Angle Stop Valve
2" Reducing Valve
3/4" Relief Valve
1/2" Steam Circ. No 4 - All Heaters on Hurricane Deck, except Dining Saloon and Forecastle. 610.5 Sq. Ft. H.S.
1/2" Steam Circ. No 3 - All Heaters on Promenade Deck, except Lounge. 382 Sq. Ft. H.S.
1/2" Steam Circ. No 2 - Lounge, Dining Saloon and Forecastle, 431 Sq. Ft. H.S.
1/2" Steam Circ. No. 1 - All Heaters on Shelter Deck, 250 Sq. Ft. H.S.
1" Steam Circ. No 6 - Music Room and Social Hall on Promenade Deck and Social Hall on Hurricane Deck, 965 Sq. Ft. H.S.
3/4" Steam Circ. No. 5 to Galley, Scullery, Bath Etc. Total Heating Surface, 1938.5 Sq. Ft.
C.I. Manifold
1/2" Steam to Pantry and Officers Mess Room.
Gage Conn.
3/4" Reducing Valve
2"
2"

FIG. 2.

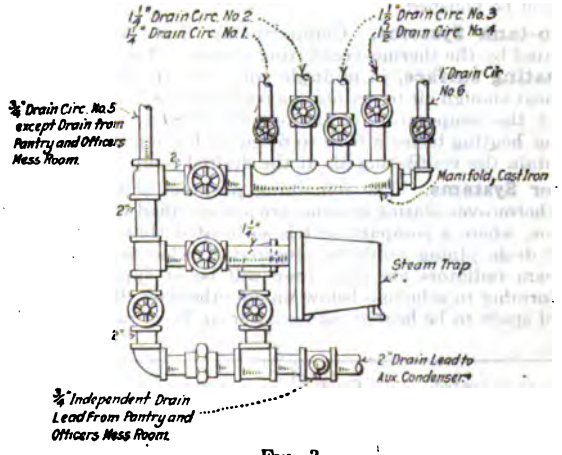

1 1/2" Drain Circ. No 2.
1/4" Drain Circ. No 1.
1 1/2" Drain Circ. No 3
1/2 Drain Circ. No 4
1" Drain Cir. No 6
3/4" Drain Circ. No 5 except Drain from Pantry and Officers Mess Room.
Manifold, Cast Iron
Steam Trap
2" Drain Lead to Aux. Condenser
3/4" Independent Drain Lead From Pantry and Officers Mess Room.

FIG. 3.

not cut tees or angles on deck or at bulkheads; where pipes pass through watertight bulkheads or decks, the same must be made water tight.

Circuits. The heating system is divided into **six circuits,** Fig. 2, as follows:

Circuit No. 1—All heaters on boat deck.
Circuit No. 2—Lounge, dining saloon and forecastle.
Circuit No. 3—All heaters on promenade deck, except lounge.
Circuit No. 4—All heaters on hurricane deck, except as covered by circuit No. 1.
Circuit No. 5—Steam to galley, pantry, baths, etc.
Circuit No. 6—Heaters in music room and social hall on promenade deck and heaters in social saloon on hurricane deck.

Radiators adjacent to woodwork should have shields, to be of galvanized sheet iron except in Pilot House and Captain's stateroom, where they will be of brass. Coils in Pilot House and Captain's stateroom should be made of 1-in. brass pipe, wrought pipe size; all other coils should be made of 1 in. wrought iron black pipe. All coils of all radiators along bulkheads or side of ship should be set 12 in. clear of deck, if possible. Brass pipe is used in the pilot house on account of the proximity to the compass.

Steam and drain manifolds are made up in accordance with Figs. 2 and 3, with commercial regrinding valves; brass valves with wooden hand-wheels are fitted to every radiator.

U. S. Navy Specifications for Heating Systems

Materials. Radiator coils, 1-in. seamless-drawn brass pipe, iron-pipe size; radiators, consisting of pipes led along the decks at bottom of bulkheads, 2-in. brass pipe, iron-pipe size; circuit steam and drain pipes, seamless-drawn brass pipe, iron-pipe size, connected by composition fittings; all exposed pipes will be polished; where polished pipes are used, valves and fittings on radiators will be polished.

Thermo-tank Systems. Compartments listed in ship's specifications will be heated by the thermo-ventilating system. For this purpose sufficient **steam heating surface,** in multiple coils, will be provided in the heating boxes, to heat enough air to maintain a temperature of 70 deg. F. in the above spaces with the temperature of the outside air at zero deg. F. The steam coils in the heating boxes will be so fitted with the necessary shut-off valves as to maintain the ventilating air at the desired temperature.

Radiator Systems. The remaining portions of the vessel and vessels on which thermo-ventilating systems are not specified will be heated by steam radiators or, where a compartment is so located that long separate lines of steam and drain piping would be required, by electric heaters as approved. Where steam radiators are used they will be of approved pattern and of areas conforming to schedule below unless otherwise directed. The number of cu. ft. of space to be heated, allowed per sq. ft. of radiator surface, will be as follows:

Space to be heated	Cu. ft.	Space to be heated	Cu. ft.
Chart house, conning tower, and fire-control tower	50	Bathrooms and water-closets not included in above	100
Quarters above upper deck	60	Storerooms, where directed	125
Sick-bay bathroom	60	Other spaces	As directed.

Steam radiators will consist of coils or of pipes led along the deck at bottom of bulkheads. Coils will be divided into two or more parts when the area called for is more than 10 sq. ft., and the shape of each radiator will be adapted to the space on bulkhead and to the space to be heated. Steam radiators consisting of pipes led along the decks at bottom of bulkheads will have proper provision for expansion and will be covered with an approved metallic casing easily removable. They will be reduced at the drain ends, and the fittings and lead will be such that water will not lodge.

Piping Systems. The steam heating plant will be arranged to work at a pressure of about 50 lbs. Where any steam radiator, coils, or piping are fitted on or near woodwork there will be protecting metal shields between the radiators and the woodwork. The shields will be easily removable, and will be separated from the woodwork by not less than ½ in. of air space. All steam radiators will be fitted with approved automatic air cocks and with valves, those in crew's space having stems of triangular cross-section, valve-stem guards, and socket keys for opening and closing, those in officers' quarters having fixed handwheels. Steam radiators and heaters will be arranged in circuits, as approved, each circuit being so connected that it can be operated independently of the others. Steam connections for the bathtubs, shower baths, and for heating water in pantries will be as short and direct as possible. These circuits will be independent of radiator circuits. There will be a connection, with valve, leading to such distributing water tanks as are exposed, to prevent freezing.

Live-steam pipes of suitable sizes, properly encased or covered, will lead to such water heaters as are installed and piping to drain such heaters will be provided. Where such heaters are used for heating salt water, the drains must not lead to the main feed tanks. The steam radiator circuits will be arranged in groups, each group having a supply manifold and a drain manifold. The supply manifold of each group will take steam from the auxiliary steam pipe, with a stop valve, an adjustable reducing valve, a relief valve, and a steam gage, at the connection with the auxiliary steam pipe. The drain manifold of each group will discharge through a trap with a by-pass into the feed tanks, and will have a stop valve and a check valve for each circuit. Where branches occur in the drain pipes, Y fittings will be used. Arranged as above, each circuit, by means of stop valves at steam and drain ends, can be entirely isolated and water from other circuits prevented from backing up into it.

The circuit steam and drain pipes will be suitably connected by fittings in a manner that will permit of their being easily taken down for repairs. Pipes passing through bulkheads and decks will be made watertight by approved stuffing boxes. Pipes passing through storerooms will have the joints located as near the door as possible. Provision will be made for expansion by horizontal U bends of copper pipe having the same internal diameter as the connecting pipes. All union joints will be coned.

Electric Heaters for the chart house, officers' quarters, and heaters of large capacity should be of the three-heat type and should have sufficient capacity to maintain a temperature of 70 deg. F. with the outside air at 20 deg. F. Heaters of a capacity of less than 1000 watts may be of the single-heat type.

For Combined Heating and Ventilating systems, see p. 1189.

VENTILATION

The purpose of ventilation is to remove air which has become hot, moist, or contaminated with undesirable elements and to replace same with pure air.

The **air should be delivered** to living quarters and similar spaces **at a low velocity** such that it does not create a great draft; for good circulation this velocity should be not less than 8 ft. per second.

The required temperature of the air supply depends upon the compartment which is to be ventilated, viz., cabin, stateroom, storeroom, cargo space or workshop. The **amount of moisture** in the air is another matter of importance, and this in turn depends upon the temperature. Dry air at 80 deg. F. has about four times the capacity for moisture that dry air at 40 deg. F. has. Air must contain sufficient moisture so that it will not have a disagreeable cooling effect. Conditions for obtaining good ventilation are more favorable at sea than on shore. Sea air contains about 3 cu. ft. of carbonic acid gas to 10,000 cu. ft. of air. The per cent content of this gas is taken as the best test for the purity of the air, as its presence also indicates the presence of other undesirable elements.

The volume of air necessary, for the average person per hour varies depending on the desired average purity of the air. If purity is to be low (high CO_2 percentage) the air required will be about 500 cu. ft. per person per hr., whereas, if the purity is to be high, the air must be replaced at the rate of about 2500 cu. ft. per person per hr.

Quantity of Air. For crew space or passenger accommodation the air should be replaced at the rate of 5 to 8 times per hr. In natural ventilation systems accomplished by having ornamental openings at the top and bottom of the partitions whereby air can circulate freely into the passage ways. These passage ways lead directly to the machinery spaces and afford good circulation of air. **Cabins** on the outboard side may be thoroughly ventilated by side lights. On some ships, side lights are installed that will permit light and air to enter at all times; floats and valves are installed which operate when submerged and prevent water from entering the ship. When cabins have open deck space above them, small but very efficient individual mushroom or ejector ventilators are fitted.

For **sick ward** air should be replaced 8 times per hr., and for **lavatories** 15 times per hr.

Storerooms are usually located below decks, and air should be renewed between 2 and 6 times per hr. depending upon the nature of the stores. They are ventilated through a duct which frequently is led to a weather deck and terminates in a goose neck. **Coal bunkers** are often ventilated by leading a large duct from the bunker to the space around the inner and outer stacks. The **shaft tunnel** is ventilated very efficiently by placing a cowl at the extreme end and turning it so that it acts as an uptake. This cowl must be watertight and of heavy construction. Air is supplied to the tunnel from the downcasts in the engine room. **Double bottoms and peak tanks** are ventilated by goose necks attached to the air-vent pipes. The air becomes vitiated only when the tanks remain empty for any length of time. **Machinery spaces** need large volumes of fresh air, the change being about 25 or 30 times per hr.; **workshop,** 15 times per hr. The **boiler room** has usually two or four cowls, the upper ends being close to the stack while the lower end of each cowl leads to within 8 feet of the firing platform. The cowls have a capacity considerable in excess of that needed for combustion; the surplus cools the stokehold and forces the heated air out through the fidley. These cowls are placed on the fidley grating and are provided with gear and pinion so that they can be trimmed from the firing platform. At times when there is little wind on deck, especially when at anchor, little air goes down the cowl (just enough for combustion), and the fireroom may

become very hot. Frequently blowers are installed to assist the cowls and no attempt is made to build up the pressure in the fireroom, the upper portion being left open to the outside atmosphere.

There is great diversity of opinion as to what is really required to adequately ventilate a ship, and generally speaking it resolves itself down to a matter of money, and how far the owners of the ship are prepared to spend money to give good ventilation. Even in the very best of cases it generally ends in a compromise, owing to the lack of adequate space on shipboard to get properly proportioned air trunks fitted, without taking up space that is reckoned more valuable either for cargo or passenger carrying purposes.

The class of the vessel, and the latitude in which it is going to trade are also very important items in arriving at the type and capacity of any ventilating system which is installed upon a ship, as naturally in the Tropics a much more vigorous air change will be required to give comfort when compared with vessels which are running in more Southern or Northern latitudes.

Methods. Ventilation on board ship is accomplished in two ways, by natural draft and by mechanical means. The **natural system** is briefly defined as the circulation of air by either the tendency of hot air to ascend and give place to heavier cold air or by directing through the desired spaces the air currents which are caused by the vessel's motion. **Mechanical ventilation** is obtained by use of fans or blowers. In fine weather, ventilation by natural means may be all that is desired, but in stormy weather, especially, where large openings must be closed, the circulation may become impaired; this may not be serious except in the tropics.

Mechanical ventilation is effected by either of two systems: (1) the **plenum system** whereby a quantity of pure air is forced into a compartment thereby replacing a certain quantity of vitiated air which passes out through exhaust trunks and other openings, and (2) the **exhaust system** whereby fans draw the foul air from the compartments and exhaust it to cowls, the foul air being replaced by air drawn in through supply cowls, air posts, hatches, etc. As a whole the plenum system is preferable as it is more positive. The exhaust system is possibly more healthful, as foul air is taken from the compartment and discharged outside instead of being driven through open spaces of the vessel, as it passes freely from each compartment to the outside air. A proper combination of both systems is preferable to either alone.

The ventilation for staterooms should preferably be by the Plenum system, in which case the fan intakes should be so located as to insure a supply of fresh clean air. The velocity in the discharge ducts from the fans should be between 1500 and 2500 feet per minute depending upon the shape and lead of the duct. In ducts of large cross-sections with direct leads, velocities as high as 4000 are now allowed on naval vessels. The velocity in supply trunks leading to the fan suction is seldom more than 1500 feet per minute. The air inlets to the various cabins in the passenger accommodation should be arranged so that the velocity through same should be reduced to about 800 to 1000 ft. per min., so that direct draft upon the occupants of such cabin should be avoided. The air would then escape through suitable openings in the door from the cabin or crew accommodation into the alley-ways, and the alley-ways, together with kitchen, storeroom, and lavatories should be ventilated upon the exhaust system, so that there would be a leakage inward into these compartments, and thus prevent the egress of objectionable smells, odors, etc.

The fans for the exhaust system applied to these particular rooms should be so arranged that there is no possibility of the discharged air coming near the intake of the fans which are supplying fresh air to the passenger or crew accommodation.

The ventilation of cargo space, if this is to be effected, is really dependent greatly upon the class of cargo carried, and the number of changes varies from 4 to 10 per hour.

The question of ventilation of coal-bunkers is an art by itself, and perhaps has caused more controversy than anything, whether there should be through ventilation or only surface ventilation, and there are advocates of both schools who contend that both systems have a tendency to promote spontaneous combustion.

The Newport News Shipbuilding and Dry Dock Co. briefly describe their practice as follows:

"In merchant vessels mechanical ventilation is used only to a limited extent except for large passenger steamers and those carrying perishable cargo. Our practice in passenger vessels is to fit supply systems for staterooms and living spaces and exhaust systems for toilets, galleys, etc. The rate of change of air is influenced by many conditions. It may be about 10 min. in public spaces, 6 to 8 min. in staterooms and from 8 to 45 min. for different kinds of cargo. Exhaust systems for galleys and lavatories may change the air in from 2 to 5 min.

"While not as essential as in naval vessels the maintaining of the **watertight integrity** of merchant ships must be considered and the ventilation is usually subdivided into a number of systems with vertical leads, the ducts thus being arranged to avoid piercing watertight bulkheads below or near the load water line. In passenger vessels the design is materially affected by the necessity of working in harmony with the decorative effect, the discharge being frequently through ornamental grilles in the ceiling.

"Our practice is to keep **velocities in mains and branches** rather low, seldom exceeding 1500 ft. per min."

Designing the System. The amount of air to be delivered to each compartment is first determined. The fan is then located with regard to economy of piping. The following should be observed in designing on the plenum ventilating system: The mains should be round, if practical. Start with a fan discharge velocity of about 2000 ft. per min. and a press. of about 5 lbs. per sq. ft., and maintain the original area as long as the velocity does not drop below 1200 ft. per min.; then the diameter should be reduced at the rate of 1½ in. per ft. until the velocity is again 2000 ft. per min. The final diameter of the main should not be reduced to less than twice the diameter of the last branch. All branches should be taken off of the main duct at an angle of 30 deg., but near the end of the main the angle should be increased to 90 deg. For bends, the centre line of the pipe should not have a smaller radius than 1½ times the diameter of the pipe. The effect of the air going around a 90-deg. elbow is such that it adds three feet to the effective length of the pipe, for two elbows the length is increased by seven feet and for three by ten feet. The outlets from branch pipes should be enlarged at a rate of 1½ in. per ft. to reduce the velocity at the exit to 1000 ft. per min. This enlargement causes an increase discharge from the line as much as 25 per cent in long branches.

The relation between area of pipe, velocity and volume of air is given by the formula $\textbf{Area} = \frac{\textbf{Volume}}{\textbf{Velocity}}$. The **loss of head** in round or square pipe equals $\frac{0.00032L \times V^2}{d}$ where L and d are length and diameter of pipe in ft. respectively, and V is velocity of air in ft. per sec.

Ventilators

The design of the deck intakes with their cowls and intakes has been so greatly influenced by the requirements of protection from waves and spray in bad weather that insufficient attention has been given to the proper admission

of air without loss or restriction. Frequently the area of the opening is insufficient or not effective. The difficulty of getting air into the shaft leading to the boiler room becomes greater as the speed of the ship increases. Many types of deck intakes, cowls and weather flaps have been tested and it has been found that inlets should have the correct clearance between the vanes, correct angle of entry and that sharp corners should be avoided. It is generally thought that the air delivered to the boiler rooms is uniformly distributed, but the lifts and compartments reaching down into the boiler room add considerable to the difficulty of thorough circulation. For the engine room there are usually two cowls, one so placed as to discharge close to the throttle, both acting as down casts. The heated air escapes out through the machinery hatch and shaft tunnel. Here the cowls should have a capacity to change the air from 15 to 20 times per hr.

Example. It is desired to install 2 engine room ventilators to supply 2000 cu. ft. per min. (1000 cu. ft. each); outlet 30 ft. below throat of cowl; ship speed 12 knots (20.3 ft. per sec.); natural draft. Find diameter of ventilators.

$$\text{Head at mouth} = \frac{V^2}{2g} = \frac{(20.3)^2}{2 \times 32.2} = 6.39 \text{ ft.}$$

By test it is known that head at throat of ventilator is less than head at mouth; generally about 0.6. Therefore, head at throat $= 0.6 \times 6.39 = 3.83$ ft. and velocity at throat $= \sqrt{2gh} = \sqrt{2 \times 32.2 \times 3.83} = 15.7$ ft. per sec. Assuming a diameter of ventilator of 15 in., the loss of head due to friction $= \frac{0.00032 \times 30 \times (15.7)^2}{1.25} = 1.90$ ft. Therefore head at exit to engine room $= 3.83 - 1.90 = 1.93$ and velocity at exit $= \sqrt{2 \times 32.2 \times 1.93} = 11.1$ ft. per sec. Volume of discharge would be area times velocity, which with a 15-in. ventilator and velocity of 11.1 ft. per sec. = 816 cu. ft. per min. This shows that the assumed 15-in. ventilator is too small. With the same computations it is found that a 16-in. ventilator will give a discharge volume of 996 cu. ft. per min., which is the desired capacity.

American Bureau of Shipping Rules for Ventilators

Ventilators. All holds, between decks, and bunkers are required to have two ventilators, placed as nearly as possible at the ends of each compartment; and a ventilator must be lead into the peaks, tunnel, or other compartments where gas is likely to accumulate. Mechanical means of ventilation will be accepted, provided the arrangements are approved by the Committee.

Ventilators on the weather portions of Freeboard Decks and on Forecastle Decks are to have **coamings** 36 in. high; those on the weather portions of Bridge Houses and Poops may be 30 in. high. The **thickness of the coamings** is not to be less than 0.30 in. in vessels not exceeding 100 ft. in length, and 0.40 in. in vessels 200 ft. in length and above; the thickness for vessels of intermediate lengths may be obtained by interpolation. The **coaming angles** are to be 0.10 in. thicker than the plates and secured to deck plating of sufficient thickness and properly stiffened, by rivets spaced not more than 4 diameters centre to centre. If coamings are more than 36 in. high and are not supported by adjacent structures, they are to have additional strength and riveting. **Ventilator cowls** are to fit closely over the coamings and are to have at least 15 in. of housing. Ventilator coamings are to be provided with strong plugs and canvas covers or efficient metal covers which can be rapidly and effectively secured in place.

Lloyd's Rules for Ventilators

Number and Size. It is recommended that ventilators, sufficient in number and size, be efficiently fitted to upper decks of all vessels.

Ventilator Coamings. The coamings of weather deck ventilators should not be less than 3 ft. in height and of the following thickness:

Not exceeding 12 in. in diam.	¼ in.
Above 12 in. and not exceeding 15 in. in diam.	5/16 in.
Above 15 in. and not exceeding 24 in. in diam.	⅜ in.

Where a steel or iron deck is fitted, these coamings are connected to it with angles $\frac{1}{16}$ of an in. thicker than the coaming. Where there is no steel or iron deck fitted, plates are to be fitted under the wooden deck to take the bolt fastenings.

Covers. Means are to be provided for efficiently closing the openings when the cowls are unshipped.

Air Ducts

(N.N.S. & D.D. Co. Practice)

Materials. Light ducts and pipes are made of galvanized sheet steel of thickness in accordance with the following table:

Size of pipe. Diameter if circular, longest side if rectangular	Thickness	
	Inch	U.S.S.G. No.
6 in. and under	0.031	22
Above 6 in. and including 10 in	0.037	20
Above 10 in. and including 12 in	0.050	18
Above 12 in	0.062	16

Watertight ducts are made of lightest obtainable lapwelded casing, not less than 0.12 in. thick. For ducts over 6 in. in diam., for short bends, or where pipes are not round they may be built of $\frac{1}{8}$ in. plate riveted with $\frac{1}{4}$ in. rivets, $1\frac{1}{4}$ in. pitch. Galvanizing must be perfected after completion.

Construction of Rectangular Pipes. Longitudinal seams of 0.031, 0.037 and 0.050 in. (No. 22, 20 and 18 U. S. S. G.) steel are grooved and locked, Fig. 4. Longitudinal seams of 0.062 in. (No. 16 U.S.S.G.) are riveted with 6 lb. rivets, about 3 in. pitch, Fig. 5. Locations of seams are taken to suit stock and work. Lap joints between pipe lengths are made in all cases away from direction of air current, Fig. 6, and all piping is made smooth inside and free from projecting lips and other obstructions.

Corners of bends in rectangular piping 0.031, 0.037 and 0.05 in. (No. 22, 20 and 18 U.S.S.G.) thick are double seamed, Fig. 7; when 0.062 in. (No. 16 U.S.S.G.) thick they are lap riveted, Fig. 8; plates are to be flanged black and then galvanized.

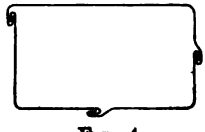
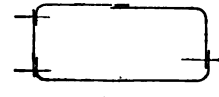


FIG. 4.



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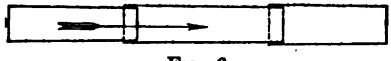

FIG. 6.

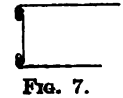

FIG. 7.

FIG 8.

Construction of Elbows. Elbows in round pipe are made wherever possible with radius of throat equal to inside diameter of pipe, Fig. 9. For 0.031, 0.037 and 0.05 in. thick pipe, joint of elbow is grooved and sweat soldered, Fig. 10; for 0.062 in. thick pipe joints are riveted, Fig. 11. For $\frac{1}{8}$ in. watertight, 6 in. diam. and less, the elbow is pressed in two sections and acetylene welded.

All ducts are soldered airtight except ducts in engine rooms and terminal branches in storerooms, workshops, berthing spaces, etc., which are made airtight by riveting or grooving and locking. All piping is fitted with flanges at such intervals as will readily permit removal of piping for repairs or cleaning. Portable plates are provided in the blank ends of vent pipes, where outlets are taken off the side or top, to allow removal of dust. Cones at terminals generally expand 1½ in. per ft.

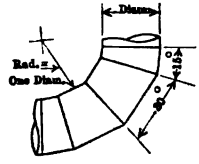

FIG. 9.

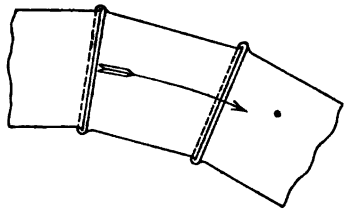

FIG. 10.

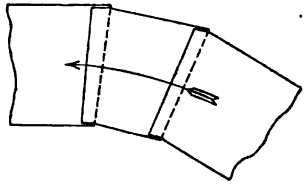

FIG. 11.

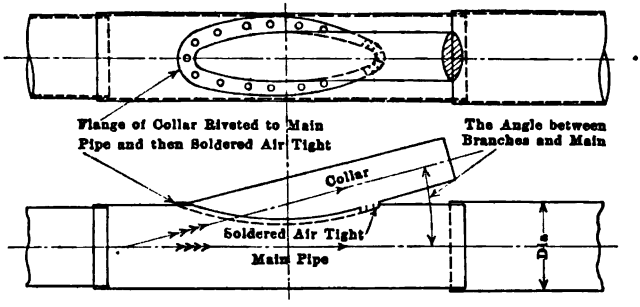
Flange of Collar Riveted to Main Pipe and then Soldered Air Tight
The Angle between Branches and Main
Collar
Soldered Air Tight
Main Pipe
Dia.

FIG. 12.

Joining Branches to Main Ducts, Fig. 12. The angle between branches and main, except branches near to the end of main, is not to exceed 45 deg. as a rule.

Non-watertight flanges in passage ways that extend within 6 ft. 4 in. headroom have the angle cut away on lower side and an outside plate and inside stiffening strip is fitted as in Fig. 13. Joints in all non-watertight ducts are made with 1/16 in. strawboard.

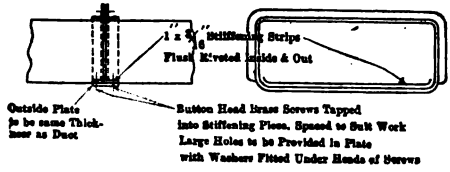
1 x 5/16
Stiffening Strip
Flush Riveted Inside & Out
Outside Plate to be same Thickness as Duct
Button Head Brass Screws Tapped into Stiffening Piece. Spaced to Suit Work
Large Holes to be Provided in Plate with Washers Fitted Under Heads of Screws

FIG. 13.

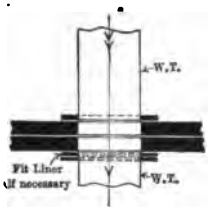

FIG. 14.

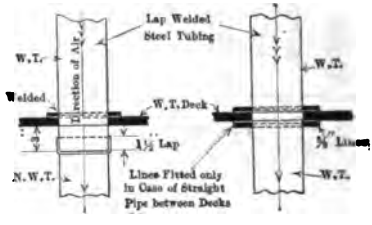

FIG. 15.

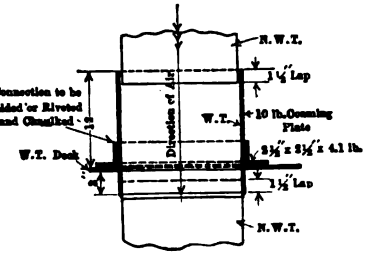

FIG. 16.

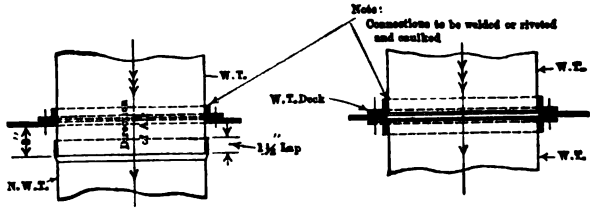

FIG. 17.

Joints of watertight ducts are made with 1/16 in. gum with cloth insertion. Where connected to bulkhead or deck, watertight duct joints are made with 2 thicknesses of No. 10 canvas soaked with red lead.

Connections to Watertight Decks. Where ducts pass through watertight decks the joints are made up as follows: (1) Where duct passes through double course decks for leads to air tested compartments as shown in Fig. 14; the space between double course decks is made tight by welding edges; (2) where steel tubing duct passes through single course watertight decks and bulkheads as shown in Fig. 15; (3) where non-watertight duct passes through watertight decks as shown in Fig. 16; (4) where watertight, built up duct passes through single course watertight decks and bulkheads as shown in Fig. 17.

COMBINED HEATING AND VENTILATION

Where both heating and ventilation systems are required the weight is very great and valuable cargo space may be sacrificed if separate systems are installed.

In Naval and to an increasing extent in merchant vessels direct radiation is being replaced very largely by an indirect system consisting of combined heating and ventilation, the outside air being forced by fans over steam coils and distributed to the various spaces through ducts. The heaters are usually provided with by-passes for the air, and preferably should have the steam coils divided into two or three sections so that the amount of heating surface may be varied to suit the temperature of the outside air.

An objection which has been raised in connection with the indirect method is the inability to regulate the amount of ventilation in different rooms of one system, independently of the amount of heat furnished, it often happening that a person desiring an increased amount of fresh air is unable to obtain it without raising the temperature of the room to an uncomfortable degree. This difficulty may be overcome by a combination of direct and indirect heating, the air being merely tempered by the ventilation heaters to maintain a temperature of perhaps 50 deg., and direct radiation under the control of the occupant of the room being provided for such additional heat as may be required. This involves a somewhat complicated installation and would hardly be justified except in high class passenger service. For such service electric heaters might be employed for the direct radiation.

A combined heating and ventilating system consists generally of a ventilating fan, a heater in intake or discharge duct thereof, and a humidifier. The apparatus may be housed on the open deck, on a sheltered deck or between decks. Air is drawn over or discharged over a heater, brought to the **proper humidity for comfort** (60 to 70 per cent for 60 to 65 deg. F. air) by injection of salt water spray or steam, and discharged to the ventilating main.

The Thermofan System

The Thermofan System, Schutte and Koerting Co., Philadelphia, Pa., is shown in Figs. 18 and 19.

Plenum System Operation. Steam is admitted to the cylindrical coil heater, Fig. 18, through the valve shown in plan; condensed water is discharged through a valve and trap. Air is drawn through the **fresh air intake duct**, is directed to the fan by the **intake damper** and is discharged to the space between the **casing** and the **heater shell.** It is heated on its upward passage by this shell, is humidified, and is further heated on its downward passage through the **heater tubes** and passes out to the ventilating system.

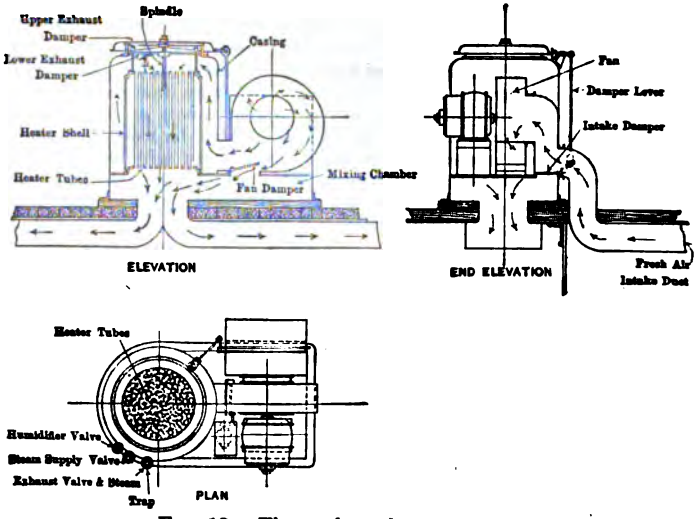

FIG. 18.—Thermofan, plenum system.

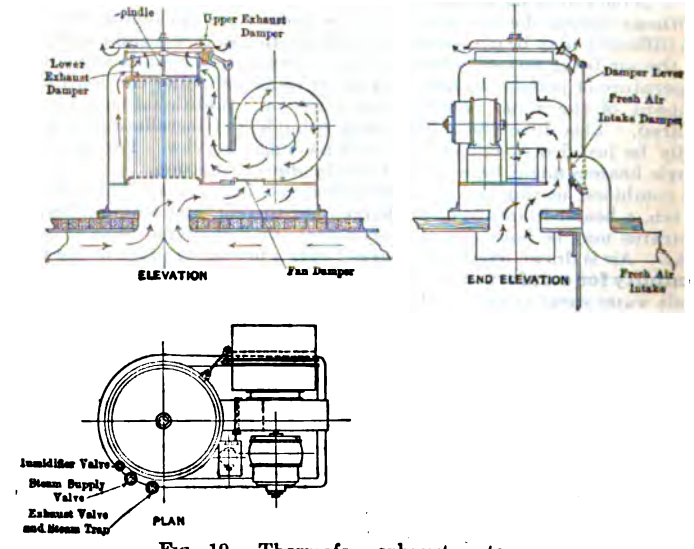

FIG. 19.—Thermofan, exhaust system.

The fan damper acts as a two way discharge damper; in its horizontal position, all air is discharged through the heater. As this damper is opened, more or less air can be admitted without heating to the **mixing chamber**; this damper thus operates as a temperature regulator. The damper is moved by a quadrant geared to a ⅛ to ¼ H.P. motor, the reversing switch of which is operated by a thermostat.

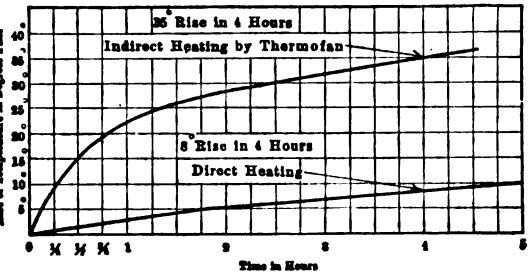
35° Rise in 4 Hours
Indirect Heating by Thermofan
8° Rise in 4 Hours
Direct Heating
Rise of Temperature in Degrees Fahr
Time in Hours

FIG. 20.

Exhaust System Operation. The Thermofan System may operate as an exhaust system as follows: The **upper and lower exhaust dampers** operate simultaneously through the **spindle**. As the upper damper is raised the lower damper is lowered until they assume the positions of Fig. 19. At the same time the fresh air intake damper is placed in the vertical position by the **damper lever**, which is operated by the upper exhaust damper. The fan damper is retained in the horizontal position. The heater is now bypassed, air is drawn from the ventilating system and discharged to the atmosphere through the annular space as shown by the arrows, Fig. 19.

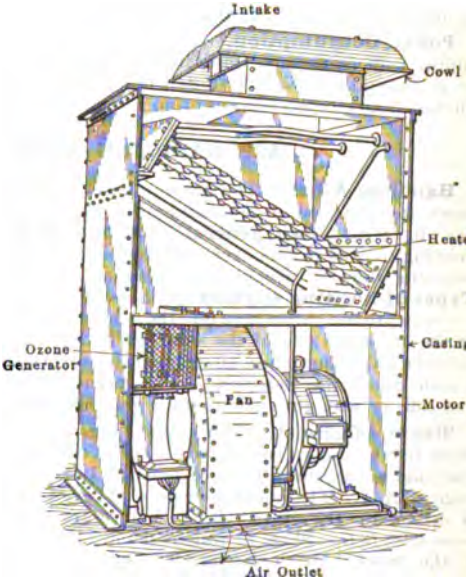

FIG. 21.

Performance Curves, Fig. 20, show the advantage of indirect heating and ventilating by this system compared with direct steam pipe or radiator heating with ventilation.

The Siroccozone System

The Siroccozone System, Davidson and Co., Ltd., Belfast, Ireland, is shown in Fig. 21. It is designed so that it can be used for either weather deck or 'tween deck use as may be required. It consists of a sheet steel **casing** fitted with heavy base angles for bolting to the deck, with the necessary internal framework, watertight doors, etc., which are

all galvanized after manufactured so as to resist corrosion. The inside of the casing is lagged, where necessary, with non-conducting material to prevent heat radiation.

The **heater**, which is arranged in two or more sections, is fitted in the top part of the casing, the air supply entering through the mushroom **cowl**. After passing over the heater the air comes into contact with the **ozone generator**, passes through the **fan**, being humidified in the process, and is then discharged into the system of trunks distributing the air throughout the accommodation.

The **apparatus for the production of ozone** is very simple, and consists of a transformer, through the primary circuit of which a low-tension alternating-current passes, producing in the secondary circuit a small current at very high tension. This is connected to the positive and negative plates of the ozone generator, from which a silent brush discharge of electricity takes place into the air passing through the generator. The oxygen contained in the air (O_2) is rapidly changed to ozone (O_3), which is exactly the same as that produced in nature, and which is always present in a bracing atmosphere.

An **ebb and flow of temperature**, in imitation of nature, is accomplished in this system by means of a timed valve which is driven by gearing from the motor spindle. This valve periodically cuts the heater out for short intervals thus allowing a momentary supply of cooler air. The manufacturers claim an invigorating effect for this apparatus.

Power Consumption. The manufacturers claim that one unit will supply sufficient air to ventilate 25 staterooms of 800 to 900 cu. ft. content at an expenditure of 1¾ to 2 B.H.P. depending on the arrangement of air ducts. The current consumption for the ozone apparatus is negligible.

ASH HANDLING MACHINERY

Handling Ashes. The accumulation of ashes in fire-rooms should be wet down and disposed of near the end of each watch at sea and at least once a day in port. **Two general methods of disposal** are employed, (1) hoisting to deck in a bucket and dumping overboard through a chute and (2) pumping directly overboard by means of an ash ejector. Three general **Types of hoisting engines** are in use, (1) steam hoist, (2) electric hoist and (3) steam ram. After hoisting, the ashes may be dumped into a vertical chute at the outboard skin of the ship, or the bucket at the top of the hoist may automatically dump into an inclined chute. In any case the chute discharges a little above the water-line. Fire-room ventilators are used for the hoists, and rails or wire guys are used for guiding the bucket.

Steam Ash Hoist. The engine, Fig. 22, is generally located in the fire-room hatch, and so connected to hoists on both sides by sliding clutch that one engine can serve for two hoists. It is a double cylinder, piston valve engine, *A,A*. The hoisting drum *C* is keyed direct to the crank shaft which is driven by the cranks *B,B*. Valves are driven off the drum shaft by the eccentrics *D,D*. Shaft *E*, which operates the automatic control gear, is geared to the drum shaft as shown. The spur wheel *G* is cast with a sleeve *H* which is in turn threaded to fit the threaded part of *E*. The sleeve *H* has a groove cut in its face, and the collar *L* fits loosely in this groove; this collar operates the valve stem *K* through bell crank.

Operation of Steam Ash Hoist. Turning the handle *N* clockwise will move the spur wheel *G* and collar *L* to the left on the shaft *E*. This will

depress valve stem *K* and start the engine running and driving the drum *C* counter clockwise. If now the handle *N* is held stationary, the gear *G* will move to the right along the threaded shaft until the engine valve is closed. The engine will run in either direction as long as the wheel *N* is kept turning. The nut *P* travels along the threaded shaft *E*. Stops are so set that *P* will bring up on a stop at each end of the desired hoist travel, thus stopping the engine.

The operating wheel *N* can be placed at any desired distant point from the engine and be connected by shafting and gears to shaft *E*. The engine

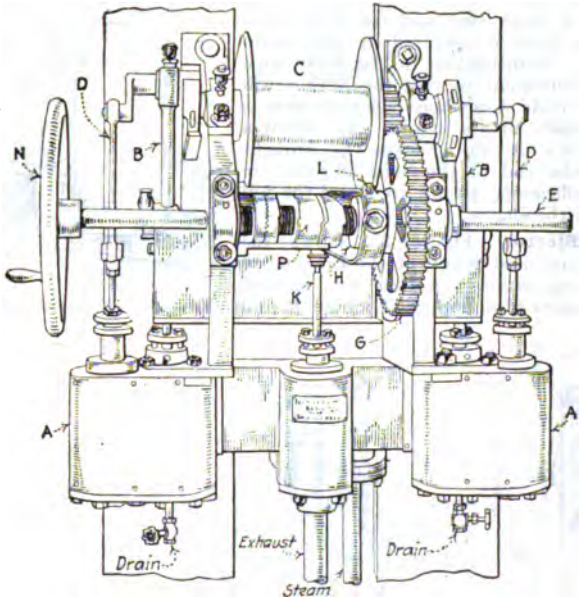

FIG. 22.—Steam Ash Hoist.

described above is used in **Naval vessels and passenger steamers.** For **ordinary merchant work** a similar engine without the follow-up gear, is extensively used for long hoists. The valve stem *K* is operated by hand lever, shaft *E* and its equipment being omitted. In this simple engine the parts are made strong enough to stop the engine if the bucket is brought up against the end of its travel.

Electric Ash Hoist, Fig. 23. Ash hoists are also driven by electric motor. The installation illustrated shows the motor and operating control as a combined unit. The motor speed is geared down to the desired drum speed. In a manner similar to the steam hoist, the drum operates follow-up gear to bring the motor to rest when the operating wheel is stopped. The operating wheel starts the motor and maintains motion of hoist as long as it is oper-

ated, the motor coming to rest when the operating wheel is stopped. A spring weighted brake is held in the *off* position by electric magnets when the hoist is in operation and is held in the *on* position by its springs when the current is broken.

Steam Ram Hoist. The installation shown in Fig. 24 represents the usual design for merchant vessels except for vessels of large size and for long lifts. The wire hoist is operated by the steam ram *A*. Turn-buckles in the hoist rope allow adjustment so that the bucket will be at the ends of the hoist when the ram is at the ends of its stroke. To avoid a break-down if the bucket is brought against the end of the hoist all parts are made sufficiently strong to stand the full power of the engine.

Ash Ejector. Figure 25 shows the arrangement and details of the Emergency Fleet Corporation standard ash ejector. This consists of a hopper *A*, chute *B* and

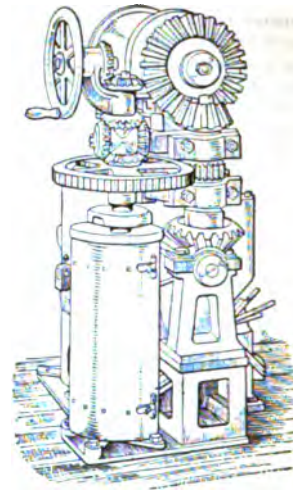

FIG. 23.—Electric Ash Hoist.

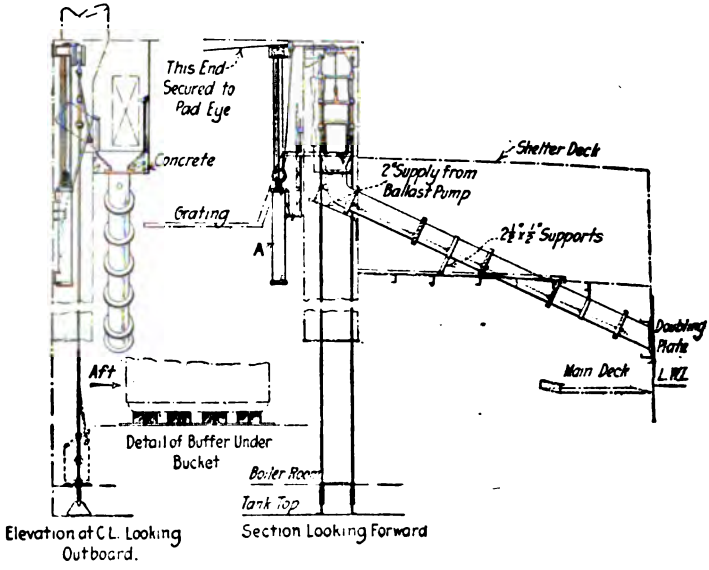

FIG. 24.—Steam Ram Ash Hoist.

pump discharge connection *C*, which contracts to a nozzle *E* (enlarged view shown in detail) and is controlled by stop cock *D*. The upper end of chute can be closed by a flap which may be operated from the fire-room floor by the lever *H*. Ashes exert a very severe scouring action on the upper bend

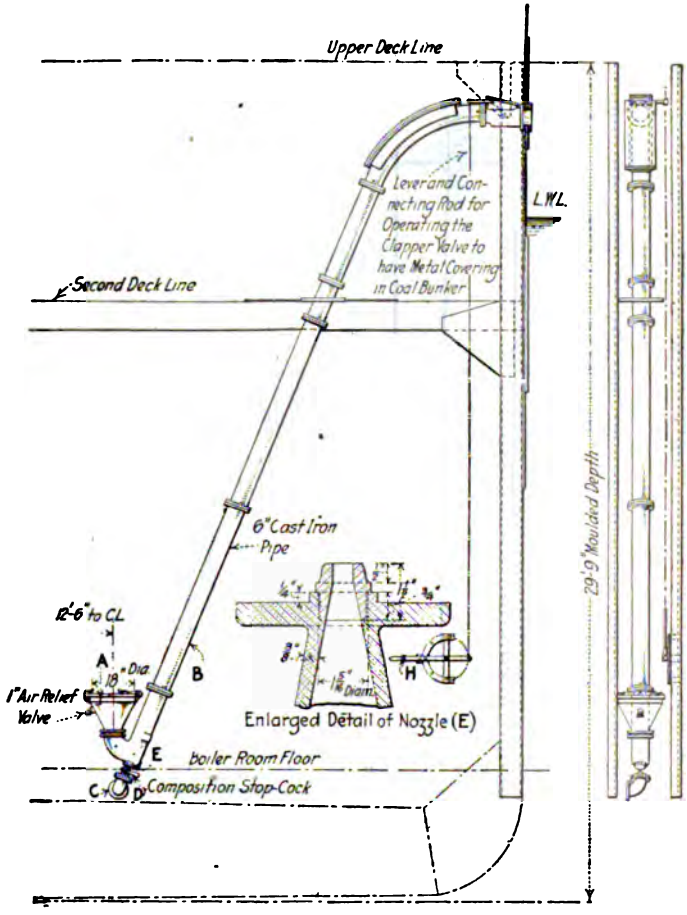

FIG. 25.—Ash Ejector.

of the chute and for this reason this bend is made of extra heavy pipe, or it may take a special form with a liner in the bend. *To operate the ejector*, close the hopper cover and start the water discharge through the nozzle. A high velocity stream will issue from the upper end of the chute. The hopper

may now be uncovered and ashes, when shovelled into the hopper, will be caught in the stream and discharged overboard.

WHISTLE AND SIREN

Steam Whistles are made in single tone type, Fig. 26 and chime type. The former, the more commonly used, consists of a metal bell *F*, open at the bottom, with a thinned lower edge *E*. Steam, admitted through the valve *A*.

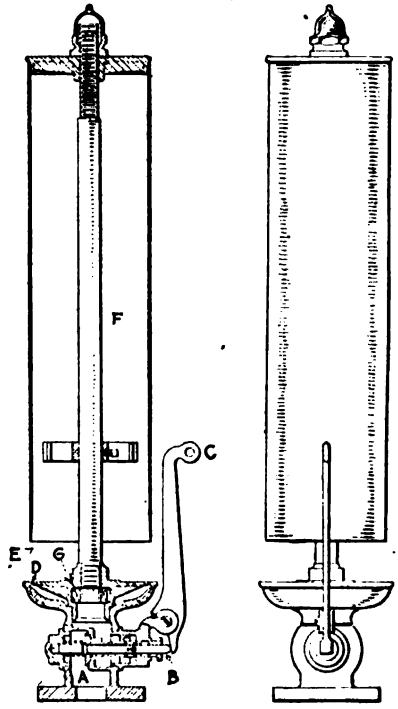

FIG. 26.—Steam Whistle.

slots *G* and passage *D*, impinges on the edge *E* and vibrations thereby set up in this edge are transmitted to the full bell wall. The admission valve is operated from the bridge by a bell pull connected to lever *C*. The whistle may also be electrically operated by solenoid through the valve stem *B*. The bell must be of sufficient length to prevent breaking up the nodes of vibration.

Steam Sirens may be operated by hand or electrically in a manner similar to the steam whistle. In Fig. 27, *A* is a revolving cylinder pierced with slots *C,C*. *B* is a stationary cylinder, also slotted, *D,D*. Steam admitted through

D,*D*, will cause *A* to revolve and the rapid interruption to the steam flow thus created will produce a musical note which will rapidly rise in tone and volume as the velocity of revolution is augmented. The trumpet, shown at the right secures to the flange on *M* and its function is to magnify the sound. Re-

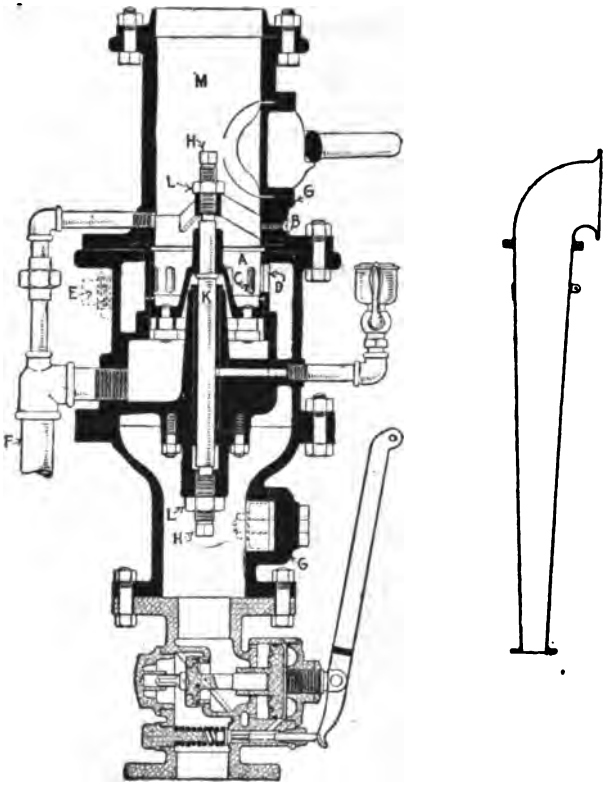

FIG. 27.—Steam Siren.

volving cylinder, *A*, is carried on a spindle *K* working through a long bushed bearing, and is accurately adjusted in position by adjusting screws *H* and lock nuts *L*. Access to these adjustments is had through clamped doors *G*, and inspection is made through plug opening *E*. *F* is the drain, which in both the whistle and siren should lead to a trap.

AIR COMPRESSORS

BY

FRANK W. STERLING

Properties of Air

Air is a mechanical mixture, by volume, of 78 per cent oxygen, 21 per cent nitrogen and 1 per cent argon; ordinarily the atmosphere contains about 0.04 per cent carbon dioxide and varying percentages of moisture.

Specific weight of pure air at standard conditions (32 deg. F. and 29.92 in. of mercury) is 0.0807. The specific weight under other conditions is given by the formula $W = (1.325 \times B) \div T$, where B = barometer reading in inches and T = abs. temp. F. For ranges of pressure and temperature beyond the ordinary meteorologic range the specific weight of pure air at various pressures and temperatures is given in Table 1.

Table 1. Weight of Air at Different Pressures and Temperatures

Temperature of air, deg. F.	Gage pressure, lbs.								
	10	20	40	80	100	150	200	250	300
	Weight in lb. per cu. ft.								
— 20	0.1515	0.2125	0.336	0.580	0.702	1.010	1.318	1.625	1.910
— 10	0.1485	0.2090	0.328	0.567	0.687	0.989	1.288	1.588	1.890
0	0.1455	0.2040	0.321	0.555	0.672	0.968	1.260	1.553	1.838
10	0.1425	0.1995	0.314	0.543	0.658	0.947	1.233	1.520	1.810
20	0.1395	0.1955	0.307	0.533	0.645	0.927	1.208	1.489	1.770
30	0.1366	0.1916	0.301	0.522	0.632	0.908	1.184	1.460	1.735
40	0.1338	0.1876	0.295	0.511	0.619	0.890	1.161	1.431	1.701
50	0.1310	0.1839	0.290	0.501	0.607	0.873	1.139	1.403	1.668
60	0.1283	0.1803	0.284	0.493	0.596	0.856	1.116	1.376	1.636
80	0.1239	0.1738	0.274	0.473	0.572	0.824	1.074	1.325	1.573
100	0.1197	0.1676	0.264	0.455	0.551	0.794	1.035	1.276	1.517
120	0.1155	0.1618	0.255	0.440	0.533	0.767	1.001	1.234	1.465
140	0.1115	0.1565	0.246	0.426	0.516	0.742	0.968	1.194	1.416
150	0.1096	0.1541	0.242	0.419	0.508	0.730	0.953	1.175	1.392
175	0.1054	0.1482	0.233	0.403	0.488	0.701	0.914	1.128	1.337
200	0.1014	0.1427	0.225	0.388	0.470	0.674	0.879	1.084	1.287
225	0.0976	0.1373	0.216	0.374	0.452	0.649	0.846	1.043	1.240
250	0.0941	0.1323	0.208	0.360	0.436	0.627	0.817	1.007	1.197
275	0.0910	0.1278	0.201	0.348	0.421	0.605	0.789	0.972	1.155
300	0.0881	0.1237	0.194	0.336	0.407	0.585	0.762	0.940	1.118
350	0.0825	0.1160	0.183	0.316	0.382	0.549	0.715	0.883	1.048
400	0.0779	0.1090	0.172	0.297	0.360	0.517	0.674	0.831	0.987
450	0.0735	0.1033	0.163	0.281	0.340	0.488	0.637	0.786	0.934
500	0.0696	0.0978	0.154	0.266	0.322	0.463	0.604	0.746	0.885
550	0.0661	0.0930	0.146	0.253	0.306	0.440	0.573	0.709	0.841
600	0.0631	0.0885	0.139	0.241	0.292	0.419	0.547	0.675	0.801

The specific weight of air at various temperatures decreases for each 10 per cent increase of humidity an amount shown in Table 2.

Table 2. D = Decrease of Specific Weight of Air for Each 10 Per Cent Decrease of Humidity, at Various Temperatures

Temperature, F.	32	40	50	60	70	80	90	100
$1000 \times D$	0.019	0.025	0.035	0.051	0.070	0.095	0.127	0.172

The pressure, volume and temperature relations of air are given by the formula $pV = 53.34T$ (see p. 195), where p = abs. press. in lbs. per sq. ft., V = sp. vol. and T = abs. temp. in deg. F.

Mean Specific Heat of pure air, approximately, at constant pressure is 0.242; at constant volume is 0.171.

For the conductance of air films see p. 188.
For the thermal conductivity of air see p. 191.
For the effect of air in condensers see p. 974.

Flow of Air

Discharge of Air. Formulæ for the discharge of air are given on p. 233. The **theoretical discharge** of air into the atmosphere at very low pressures are given in Table 3. The actual velocity can be obtained by multiplying the values in Table 3 by a **coefficient of discharge,** see p. 233, which varies with the form of orifice. For sharp edged orifices in thin plates its value is 0.64; for a plate with rounded orifice on the inside 0.70 to 0.75; for short cylindrical pipe, inner corners not rounded, 0.75 to 0.84; and for a well rounded mouth piece, 0.98. For the **velocity of compressed air discharging into the atmosphere** Hiscox in "Compressed Air" gives the values in Table 4. The **quantity of air discharging into the atmosphere through an orifice** is given in Table 5.

Table 3. Velocity of Efflux of Air Under Low Pressures

(Temperature 62 deg. F. Barometer 30 in.)

Pressure		Theoretical velocity, ft. per sec.	Pressure		Theoretical velocity, ft. per sec.
Inches of water	Lbs. per sq. ft.		Inches of water	Lbs. per sq. ft.	
0.01	0.052	6.61	0.8	4.15	59.1
0.02	0.104	9.35	0.9	4.67	62.7
0.04	0.208	13.2	1.0	5.19	66.1
0.07	0.363	17.4	1.5	7.79	80.9
0.10	0.519	20.9	2.0	10.38	93.5
0.20	1.038	29.5	2.5	12.98	104.0
0.30	1.558	36.2	3.0	15.58	114.0
0.40	2.077	41.8	3.5	18.18	124.0
0.45	2.337	44.3	4.0	20.77	132.0
0.50	2.597	46.7	4.5	23.37	140.0
0.60	3.116	51.2	5.0	25.97	148.0
0.70	3.635	55.3	6.0	31.16	162.0

Table 4. Velocity of Efflux of Compressed Air

Pressure			Theoretical velocity, ft. per sec.	Pressure			Theoretical velocity, ft. per sec.
Atmospheres	Inches of mercury	Lbs. per sq. in.		Atmospheres	Inches of mercury	Lbs. per sq. in.	
0.010	0.30	0.147	94.4	0.680	20.4	10.0	780
0.066	2.10	1.00	246.0	0.809	24.28	12.0	855
0.100	3.00	1.47	299.0	1.0	30.0	14.7	946
0.136	4.08	2.00	348.0	2.0	60.0	29.4	1094
0.204	6.12	3.00	472.0	5.0	150.0	73.5	1219
0.272	8.16	4.00	493.0	10.0	300.0	147.0	1275
0.340	10.20	5.00	552.0	20.0	600.0	294.0	1304
0.408	12.24	6.00	604.0	40.0	1200.0	588.0	1323
0.500	15.00	7.35	673.0	100.0	3000.0	1470.0	1331
0.544	16.32	8.00	697.0	200.0	6000.0	2940.0	1334
0.611	18.34	9.00	741.0				

Table 5. Discharge of Air Through An Orifice

Cu. Ft. Free Air per Min.
(Ingersoll-Rand Co.)

Diameter of orifice, in.	Receiver gage pressure, lbs. per sq. in.								
	2	5	10	15	20	25	30	35	40
1/64	0.038	0.0597	0.0842	0.103	0.119	0.133	0.156	0.173	0.19
1/32	0.153	0.242	0.342	0.418	0.485	0.54	0.632	0.71	0.77
1/16	0.607	0.965	1.36	1.67	1.93	2.16	2.52	2.80	3.07
1/8	2.435	3.86	5.45	6.65	7.7	8.6	10.0	11.2	12.27
1/4	9.74	15.4	21.8	26.7	30.8	34.5	40.0	44.7	49.09
3/8	21.95	34.6	49.0	60.0	69.0	77.0	90.0	100.0	110.45
1/2	39.0	67.0	87.0	107.0	123.0	138.0	161.0	179.0	196.35
5/8	61.0	96.5	136.0	167.0	193.0	216.0	252.0	280.0	306.80
3/4	87.6	133.0	196.0	240.0	277.0	310.0	362.0	400.0	441.79
7/8	119.5	189.0	267.0	326.0	378.0	422.0	493.0	550.0	601.32
1	156.0	247.0	350.0	427.0	494.0	550.0	645.0	715.0	785.40
1 1/4	242.0	384.0	543.0	665.0	770.0	860.0	1000.0	1120.0	1230.0
1 1/2	350.0	550.0	780.0	960.0	1102.0	1240.0	1435.0	1600.0	1770.0
2	625.0	985.0	1395.0	1700.0	1967.0				

Diameter of orifice, in.	Receiver gage pressure, lbs. per sq. in.							
	45	50	60	70	80	90	100	125
1/64	0.208	0.225	0.26	0.295	0.33	0.364	0.40	0.486
1/32	0.843	0.914	1.05	1.19	1.33	1.47	1.61	1.97
1/16	3.36	3.64	4.2	4.76	5.32	5.87	6.45	7.85
1/8	13.4	14.5	16.8	19.0	21.2	23.5	25.8	31.4
1/4	53.8	58.2	67.0	76.0	85.0	94.0	103.0	125.0
3/8	121.0	130.0	151.0	171.0	191.0	211.0	231.0	282.0
1/2	215.0	232.0	268.0	304.0	340.0	376.0	412.0	502.0
5/8	336.0	364.0	420.0	476.0	532.0	587.0	645.0	785.0
3/4	482.0	522.0	604.0	685.0	765.0	843.0	925.0	
7/8	658.0	710.0	822.0	950.0	1004.0	1145.0	1260.0	
1	860.0	930.0	1070.0	1215.0	1360.0	1500.0	1648.0	

Work of Compression

Work of Adiabatic Compression of Air. If air is compressed from a volume v_1 and pressure p_1, to a volume v_2 and pressure p_2, in a non-conducting cylinder without clearance, the work involved in delivering 1 lb. is as follows:

$$\text{Work of compression} = 2.46\, p_1 v_1 \left[\left(\frac{v_1}{v_2} \right)^{0.41} - 1 \right]$$

$$= 2.46 p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{0.29} - 1 \right].$$

$$\text{Work of expulsion} = p_2 v_2 = p_1 v_1 \left(\frac{p_2}{p_1} \right)^{0.29}.$$

Total work is the sum of the work of compression and expulsion less the work, p_1v_1, of the atmosphere done on the piston during admission:

$$\textbf{Total work} = 3.46p_1v_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right].$$

$$\textbf{Mean effective pressure} = 3.46p_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right].$$

$$\textbf{Final temperature} = T_1\left(\frac{p_2}{p_1}\right)^{0.29} \text{ where } T_1 = \text{initial absolute temperature.}$$

Air Compressors

Marine Use. Compressed air is used on board ships for running pneumatic tools, cleaning fuel oil burner tips and for air atomizing fuel oil systems.

Types. Single stage air compressor, similar to a locomotive air compressor, is the one almost universally used for marine work. A cross compound two-stage compressor is used to a small extent, especially in Naval vessels, for ejecting gases from guns.

The piston speed of Marine Reciprocating Compressors is approximately 100 ft. per min.

Description. The Westinghouse Single Stage Compressor, Figs. 1, 2 and 3 is built right and left handed in sizes 8 × 8 × 10, 9½ × 9½ × 10 and 11 × 11 × 12. The first figure denotes diameter of steam cylinder, the second denotes diameter of air cylinder and the third denotes stroke, all in inches. All parts for compressors of same size are interchangeable. It consists of a steam cylinder vertically over an air cylinder, both being double acting. The two are joined by a central casting. Steam and air pistons are mounted on a common piston rod.

Steam admission and exhaust are controlled by steam actuated valve gear in the steam cylinder head as follows: Chambers *A* and *C* are always in free communication with each other and with steam inlet through ports *B*, *B'*. Main exhaust port *D* and chamber *E* at the left of piston *F* always communicate through port *G*, Fig. 2, so that chamber *E* is always free of pressure. When reversing slide valve *K* is in the position shown in Fig. 1, chamber *J* and exhaust passage *D* are connected through port *L*, reversing valve cavity, and port *H*, and there is no pressure at the right of piston *P*.

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When piston *T* reaches the upper end of its stroke, reversing plate *W* strikes shoulder near top of rod *X*, forcing it and slide valve *K* upward sufficiently to expose port *Y*. Steam from chamber *C* then enters chamber *J* through port *Y* and port *L* of the bushing, Fig 2, thus equalising pressure upon the two faces of piston *P* and balancing the piston. The pressure in chamber *A* now acts on piston *F*, forcing it to the left thus moving the reversing valve to the left.

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1/8	2.435	3.86	5.45	6.65	7.7	8.6	10.0	11.2	12.27
1/4	9.74	15.4	21.8	26.7	30.8	34.5	40.0	44.7	49.09
5/16	21.95	34.6	49.0	60.0	69.0	77.0	90.0	100.0	110.45
1/2	39.0	67.0	87.0	107.0	123.0	138.0	161.0	179.0	196.35
9/16	61.0	96.5	136.0	167.0	193.0	216.0	252.0	280.0	306.80
3/4	87.6	133.0	196.0	240.0	277.0	310.0	362.0	400.0	441.79
7/8	119.5	189.0	267.0	326.0	378.0	422.0	493.0	550.0	601.32
1	156.0	247.0	350.0	427.0	494.0	550.0	645.0	715.0	785.40
1 1/4	242.0	384.0	543.0	665.0	770.0	860.0	1000.0	1120.0	1230.0
1 1/2	350.0	550.0	780.0	960.0	1102.0	1240.0	1435.0	1600.0	1770.0
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1/4	53.8	58.2	67.0	76.0	85.0	94.0	103.0	125.0
5/16	121.0	130.0	151.0	171.0	191.0	211.0	231.0	282.0
1/2	215.0	232.0	268.0	304.0	340.0	376.0	412.0	502.0
9/16	336.0	364.0	420.0	476.0	532.0	587.0	645.0	785.0
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When slide valve *R* reaches its extreme left position, steam from chamber *A* is admitted above piston *T*, through port *Z*, forcing *T* downward; at the same time, steam below the piston is discharged to the atmosphere through port *a* chamber of the slide valve, cavity and port *D* and the exhaust *U*.

When piston *T* reaches its lower centre, reversing plate *W* engages reversing button *b*, drawing it and slide valve *K* down to the position shown. This completes the steam cycle.

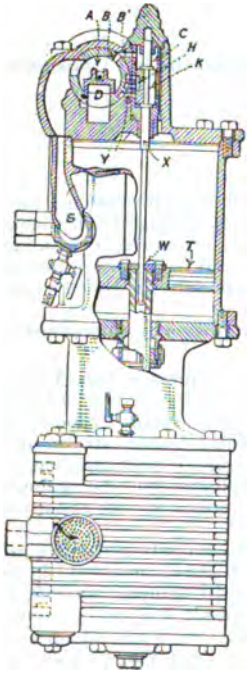


Fig. 1.

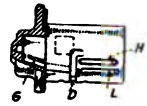


Fig. 2.

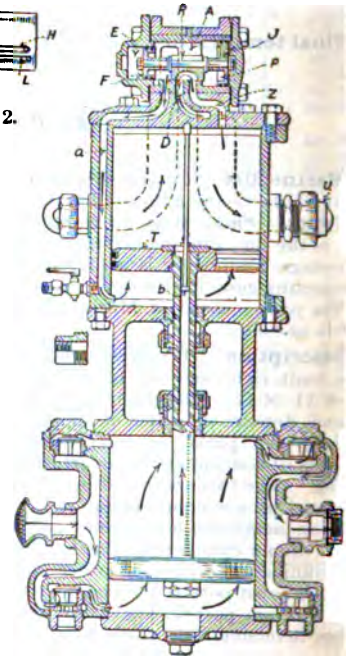


Fig. 3.

Air End. There are two admission air valves on the left, Fig. 3, and two discharge air valves on the right. A strainer is fitted to the suction. On the upward stroke air above the air piston is compressed and discharged into the main reservoir, while the space below it is filled with air drawn through the bottom admission valve from atmosphere On the downward stroke this action is reversed.

Dimensions. Table 6 gives dimensions of air compressors for Marine use. When two compressors are installed in parallel taking steam from and discharging air to common mains the sizes of governors, main and branch pipes are given in Table 7.

Table 6. Dimensions of Air Compressors for Marine Use

Size of Compressors	8-in.	9½-in.	11-in.
Diameter of Steam Cylinder	8″	9½″	11″
Diameter of Air Cylinder	8″	9½″	11″
Stroke, in	10″	10″	12″
Governor	1″	1″	1¼″
Steam Valve	1″	1″	1¼″
Steam Admission Pipe	1″	1″	1¼″
Steam Exhaust Pipe	1¼″	1¼″	1½″
Air Admission Pipe	1½″	1½″	1½″
Air Delivery Pipe	1¼″	1¼″	1¼″
Rated Speed, single strokes per min	120	120	100
Displacement in cu. ft. at rated speed	35	49	66
Overall Dimensions:			
Height	42 5/16″	42 5/16″	51 5/16″
Width	17¾″	18¼″	21½″
Depth	13 3/4″	14 5/16″	15 13/16″
Weight	450 lbs.	550 lbs.	875 lbs
Lift of Air Valves	3/32″	3/32″	3/32″

Table 7. Size of Governors, Steam Valves and Piping for Two Compressor Installation

Size of Compressors	9½-in.	11-in.
Governor	1¼″	1½″
Steam Valve	1¼″	1½″
Steam Admission Pipe:		
Main Pipe	1¼″	1½″
Branch Pipe	1″	1¼″
Steam Exhaust Pipe:		
Main Pipe	2″	2½″
Branch Pipe	1¼″	1½″
Air Admission Pipe	1½″	1½″
Air Delivery Pipe:		
Main Pipe	1½″	1½″
Branch Pipe	1¼″	1¼″

Installation. The compressor must be installed on rigid foundations or pounding will result. Suction must be at such point as clean, cool, dry air can be obtained. Ample air banks must be provided, and same should be drained daily.

Piping. For materials see p. 1254. Bends should be used instead of fittings wherever possible. Before installing, piping should be hammered to loosen scale and dirt, fins should be removed and pipes blown out with steam. Pockets in pipes should be avoided. Screwed joints should be made with shellac or Japan varnish on male threads; *never* use red or white lead.

Testing Air Compressors

Capacity Test. A 9½-in. compressor should maintain 60 lbs. pressure at reservoir against an 1 1/64-in. orifice, at not more than 120 single strokes per min. An 11-in. compressor should not require over 100 single strokes per min. to maintain 60 lbs. at reservoir against a 3/16-in. orifice. Before conducting a capacity test the system should be tested for tightness. The pressure drop due to leakage in the system should not exceed 2 lbs. per min. with the system under a pressure of 60 lbs.

Testing Steam End. Open compressor steam throttle wide and regulate reservoir air pressure by bleeding to atmosphere to value given on curve, Fig. 4 (59 lbs. for 9½ in. and 66 lbs. for 11 in.). Note steam pressure and compressor speed required to maintain this pressure and compare these values with values given by curves. These curves represent what the steam performance should be with steam end in good average condition. If the observed speed is less than that given by the curve it will indicate the compressor is not in good average condition.

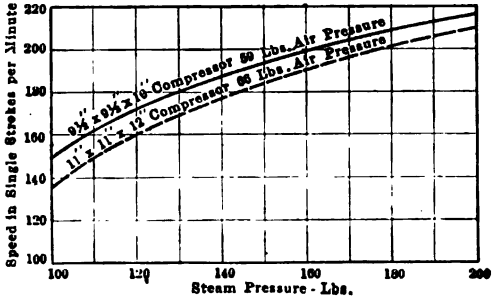


FIG. 4.—Air Compressor, Steam End Performance.

Example: Suppose with steam throttle of 9½ in. compressor opened wide and the reservoir pressure regulated to 59 lbs. by bleeding to atmosphere, a speed of 160 single strokes per min. is attained with a steam pressure of 122 lbs. Fig. 4 shows that with steam pressure of 122 lbs. the compressor speed, when working against 59 lbs. reservoir pressure, should be 174 strokes per min.

Operation and Maintenance of Compressor

Three essentials to maintenance of a compressor are: (1) It must have ample capacity for the required service; (2) good lubrication and general cleanliness and (3) minimum leakage in compressor and air system.

If the **capacity is not ample** continuous running at high speed will cause excessive heating of the air end. This will destroy lubrication, cause groaning and cutting of the air cylinders, produce condensation of moisture in the air system and deposits of burnt oil in the discharge passage. All these will reduce the overall efficiency of the compressor.

If the **compressor refuses to start,** this is probably caused by insufficient lubrication in steam end, leaky piston rings in the small ends of main valve piston, or to accumulation of rust during idle period. To remedy, introduce small quantity of lubricant through reversing valve cap and open throttle quickly.

If the **compressor groans** this is probably due to insufficient lubrication in air end or to dry or binding piston rod. It may also be caused by leaky air piston packing rings or leaky discharge valve.

Uneven strokes of the compressor is probably caused by (1) excessive leakage past air piston packing rings and sticky air valves, (2) improper lift of air valves, (3) clogged discharge valve passages, (4) leaky air valves, or (5) binding or cutting of reversing rod.

If **compressor heats** (1) air passages are clogged, (2) air piston packing rings leak, (3) discharge valves have insufficient lift, or (4) speed is excessive or running time too long. Although heating of the air cylinder during compression is inevitable, yet excessive temperature may and should be guarded against.

If **compressor pounds** (1) air piston is loose, (2) reversing plate is loose, (3) reversing rod or plate is badly worn, or (4) compressor is insecurely fastened.

Erratic action of compressor is generally due to worn condition of valve motion.

Lubrication. Insufficient lubricant is fatal to the compressor. On the other hand, excess lubricant to the air cylinder will result in carbon accumulation that will clog pipes and passages. This may become so serious as to cause explosion. Kerosene should not be introduced to air cylinder for removing carbon deposits. Numerous explosions have been traced to this practice. To clean valves and passages, a hot solution of lye or potash should be circulated through the compressor at routine periods, following this with a thorough rinse of clean hot water. The use of kerosene and naphtha for cleaning may result in dangerous **explosions.**

The best grades of **cylinder oil** should be used for compressor cylinder lubrication. Excess oil feed to air cylinders, or an oil of too high viscosity for the purpose will cause a gradual accumulation of carbon which will cause bad valve action and may even choke air passages and raise the temperature to a dangerous point. The Technical Committee of the Compressed Air Society, New York, in "The Lubrication of Air Compressors" published in National Petroleum News, discuss this subject very fully. The conclusions which are in accord with U. S. Naval Practice, are that an oil of high viscosity is not necessary, unless the conditions demand it; the lighter viscosity oils are far less carbonizing than the heavier viscosity oils and, provided the lighter oils form a suitable lubricating film, preference should be given to the lighter oils.

DECK MACHINERY

BY

FRANK GENTLES

Deck machinery as commonly understood consists of the following equipment: **Steering Gear and Control, Windlasses, Winches, Capstans, Towing Machines, Etc.**

The importance of having this equipment designed to suit the conditions which it is to meet is not generally realized. The steering gear should be given the same careful consideration as the Propelling Machinery and Boiler. Much time can be saved in port and large demurrage charges avoided by having an efficient cargo handling equipment and eliminating stoppages due to the breaking down of the winches. Many vessels could have been saved if the windlasses or ground equipment had not failed at the critical moment.

STEERING GEAR

Steam steering gear for large merchant and passenger vessels consist of one of the following types:

The steam tiller, the steam quadrant steerer, the R. & L. screw gear steering engine and The Electro-hydraulic steerer.

The Steam Tiller is shown in section in Figs. 1. Tiller *A* is keyed to rudder head *B* and at the other end jaw *C* is fitted with a driving pinion *D* which works in toothed segment *E*. The steering engines are carried on the tiller and move around with it. Motion is communicated to pinion *D* through the friction clutch *H* which engages the worm wheel *I*. This wheel is made to fit the worm. Motion is given to the worm by the steam engine. The **friction clutch should be expanded** sufficiently tight to put the rudder hardover at full speed, but should not have sufficient force to hold the connection tight enough to cause fracture of any part of the machinery.

The **advantages** claimed for this gear are: (1) The tiller is keyed direct to rudder head, doing away with the necessity of chains, rods, or wire ropes which carry the full rudder strain; (2) it is very flexible and lets go of the rudder when unduly strained, returning automatically to its former position as soon as the strain has passed; (3) advantage is taken of the limits of the poop deck to make the tiller as long as possible, thereby having a large lever acting directly on the rudder stock, and accordingly reducing the size of the steering engines necessary.

The Quadrant Steering Engine. Two general arrangements of the quadrant steering engine are shown in Figs. 2, 3 and 4. The quadrant as shown is direct connected to the rudder stock, and the motive power from the engine is transmitted through a worm and pinion to the quadrant.

This type of gear is used where the space is limited. It is very flexible. a friction clutch similar to that shown on the tiller being fitted; or the quadrant rack is mounted on a floating frame and attached to a tiller by spring buffers which absorb any undue strain coming on the rudder and transmitted

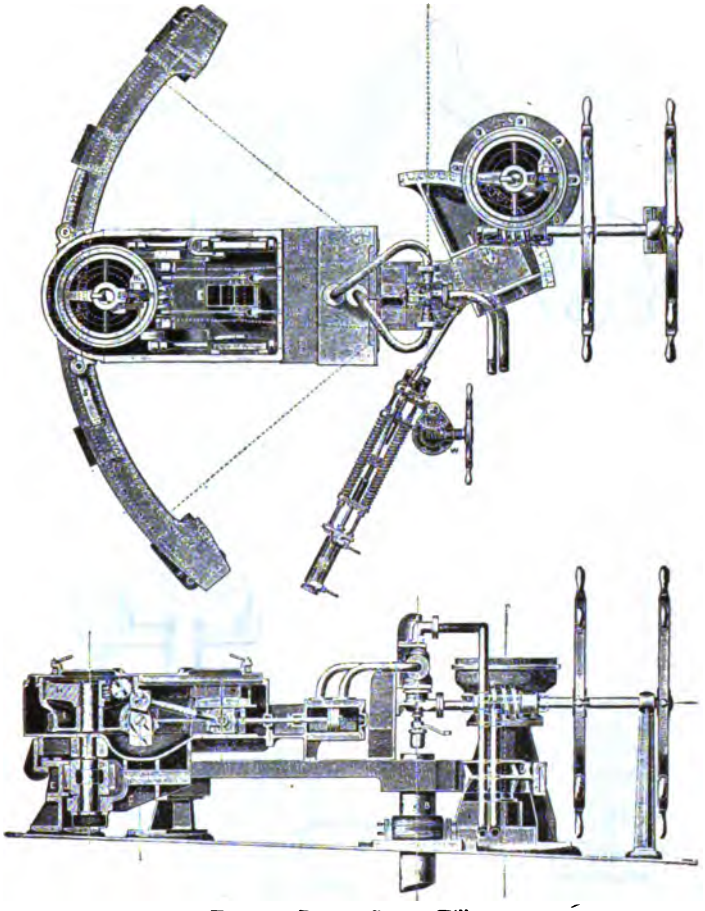


FIG. 1.—Brown Steam Tiller.

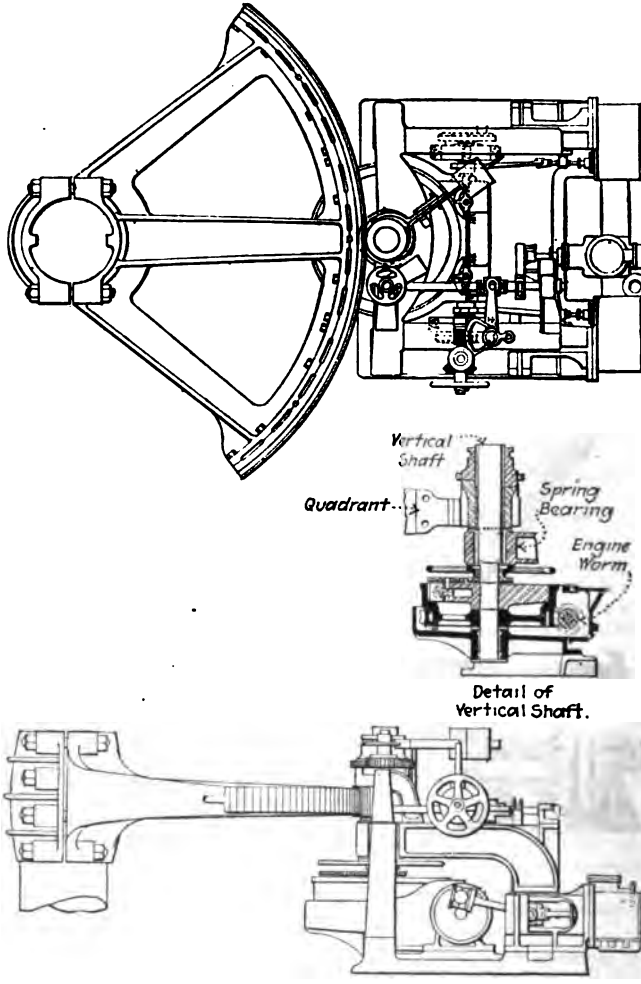
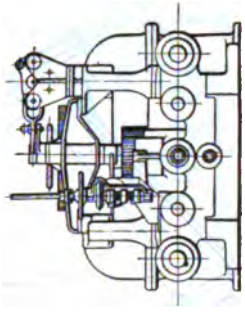
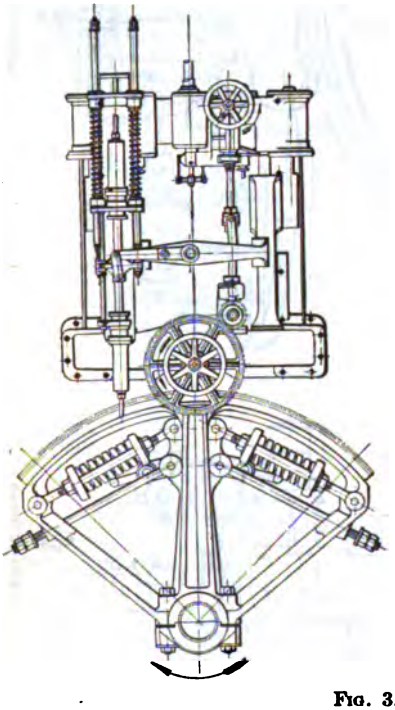

Vertical Shaft
Quadrant
Spring Bearing
Engine Worm
Detail of Vertical Shaft.

FIG. 2

through the rudder stock, Fig. 3. The floor space occupied by the steering gear can be still further reduced by using vertical engines, a very efficient arrangement of which is shown in Fig. 4.

The R. & L. Screw Type Steering Engine consists of a shaft with right and left hand threads cut on it, these threads meet at the center of the shaft. Travelling nuts with sleeves are mounted on each screw. The sleeves run on guide bars and carry links that connect through a crosshead directly with the rudder stock. The motive power of the engine is transmitted through a suitable spur gear reduction to the screw shaft. General arrangements of two types of the R. & L. steerer are shown in Figs. 5 and 6.





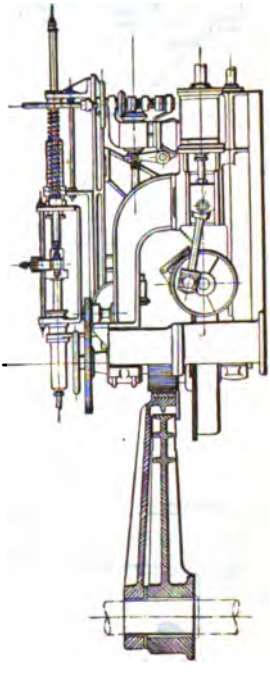


FIG. 3.

This type of gear is used where a powerful gear is required, it is compact and positive. Any rudder shock is taken up in the screw which should be

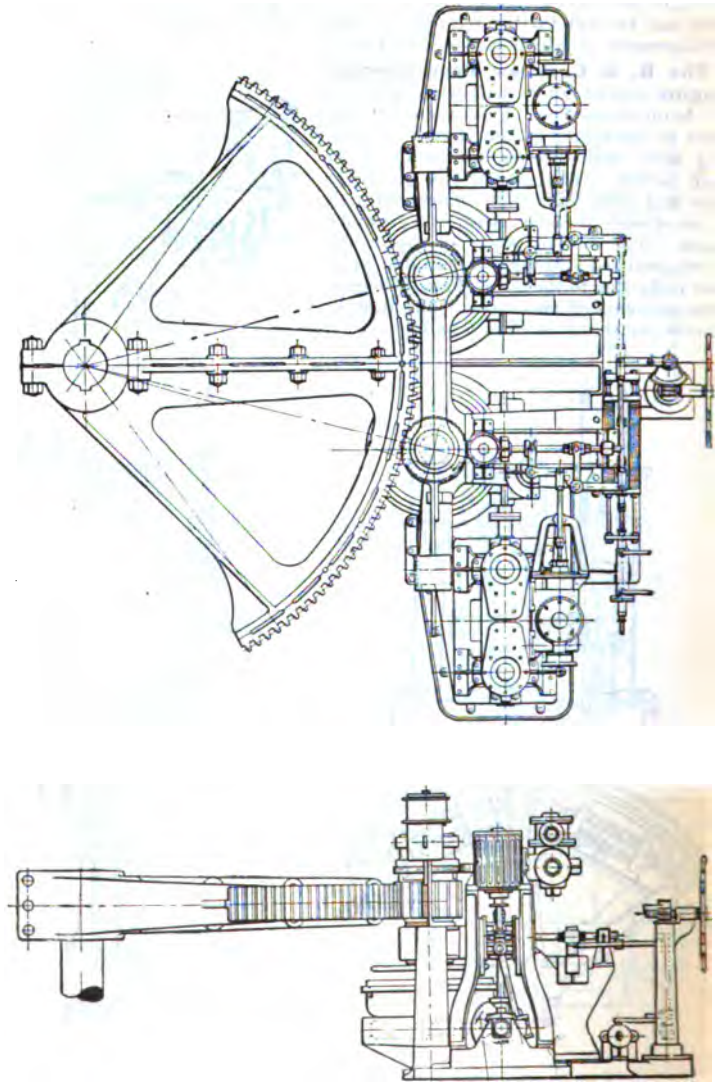


FIG. 4.

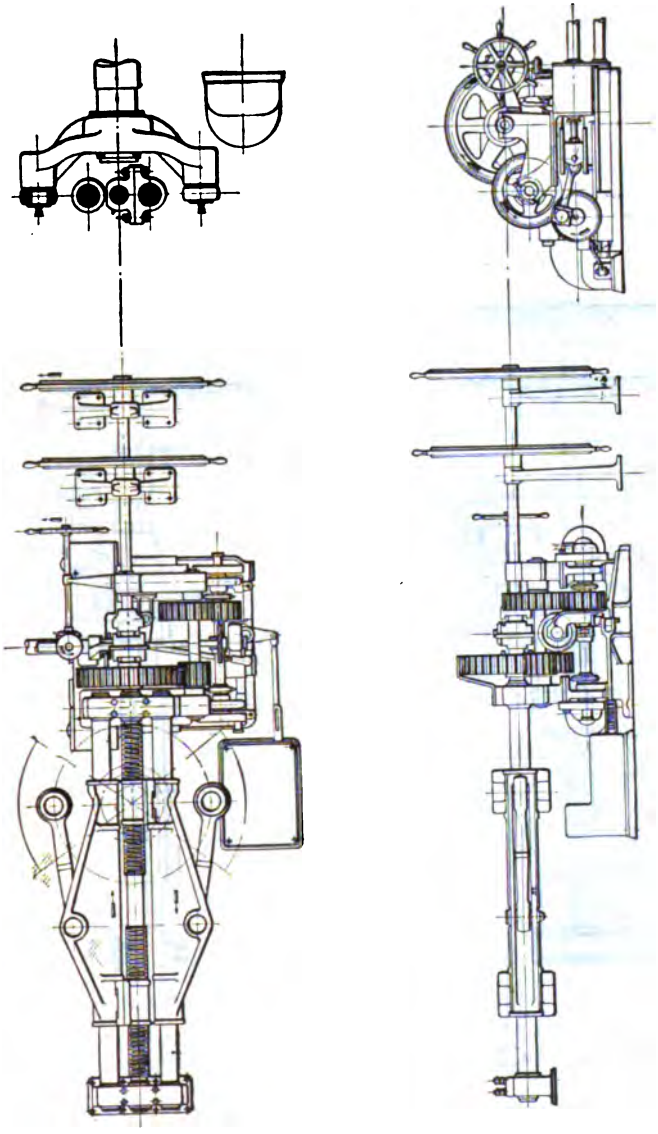
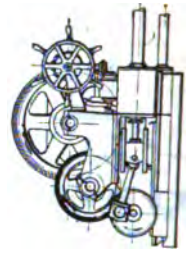
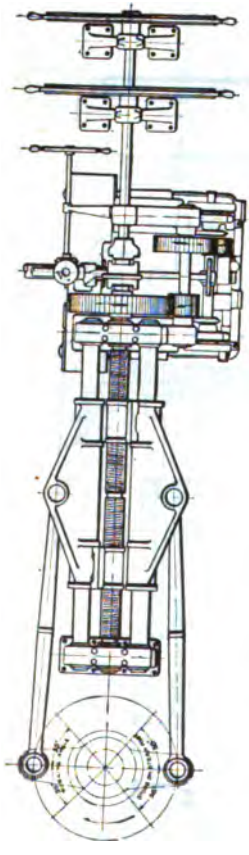


Fig. 5.





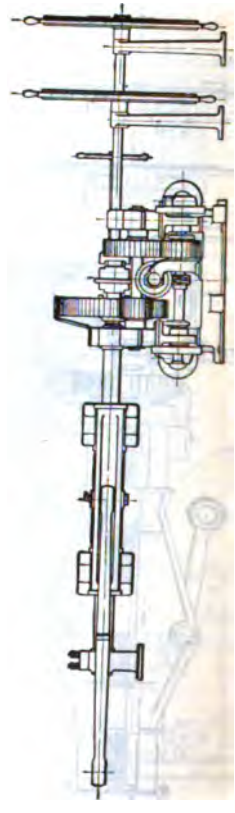


FIG. 6.

designed strong enough to do so without damage. On large and fast vessels the rudders should be of the semi-balanced type in order to keep the strain coming on the links and screw to the smallest limits possible.

The design of the Screw Gear Steerer lends itself readily to quantity production and for this reason was adopted by the Emergency Fleet Corporation for many of its vessels during the war.

It makes a very compact combined hand and screw steerer. The hand steering wheels engaging by means of a clutch with the screw shaft. In merchant vessels a gear reduction of at least three to one should be introduced between the hand steering wheels and the screw gear.

The Drum Type Steerer is used on medium size vessels and tug boats, where the limited space will not permit the steering engine being located aft. A general arrangement of a steam drum type steerer is shown in Fig. 7 and is self explanatory. The tiller ropes or chains pass around the drum and aft on either side of the quadrant. Rudder shock is taken up by buffer springs between the drum and the quadrant.

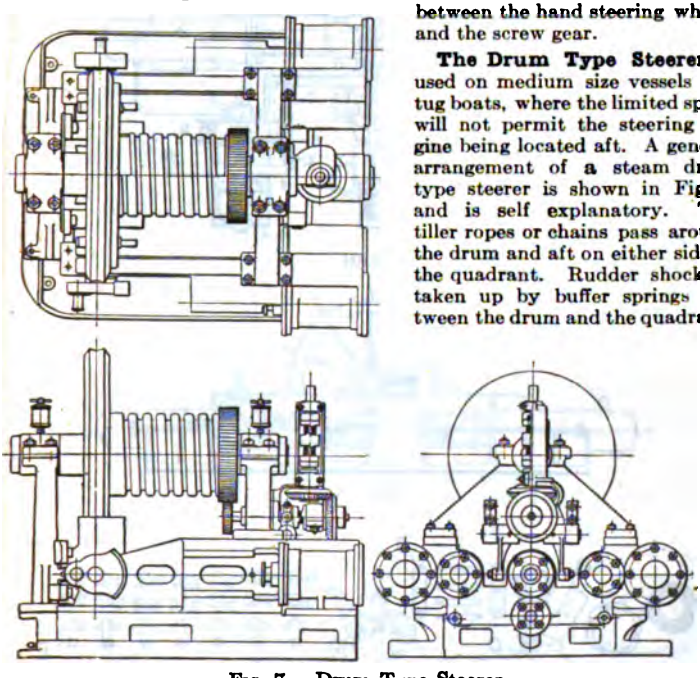


FIG. 7.—Drum Type Steerer.

The steerer is generally located either in the pilot house or in the engine room.

Steam Steering Gear Control. The steam valve of all the types of steam steering gear described is actuated from the steering stations by means of a suitable form of transmission. When the steering gear is located amidship or the vessel is likely to operate in extremely cold climates, the transmission is either obtained by means of shafting and bevel gears or a wire rope control. When located aft a hydraulic telemotor or electric telemotor control is used.

In the tiller type of gear the steam and exhaust valves are mounted on the same centre line as the rudder stock. Any motion of the trick wheel in the

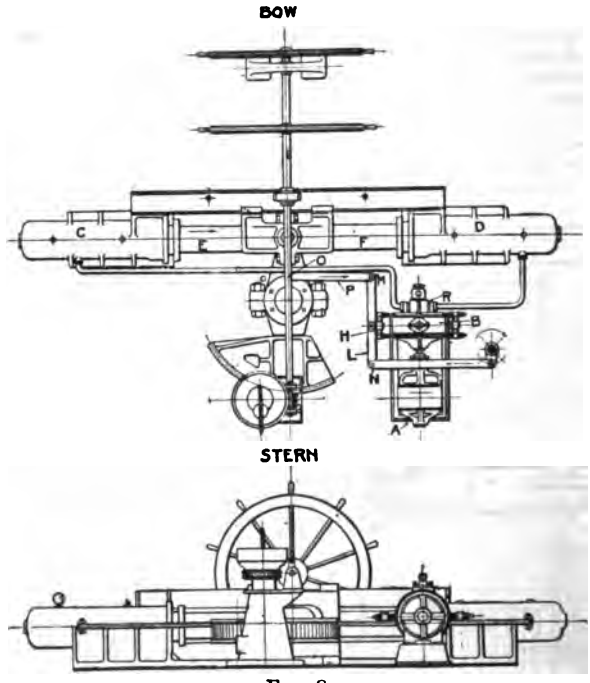


Fig. 8.

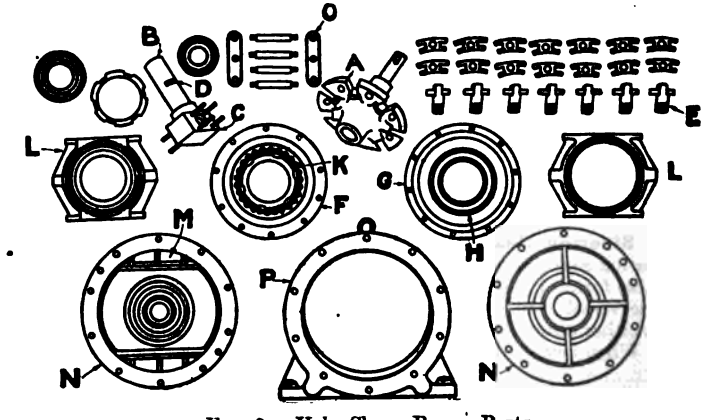


Fig. 9.—Hele Shaw Pump Parts.

pilot house is transmitted by a suitable transmission system to the steam valve, admitting steam to the engine. As the engine rotates the pinion to right or left it enables the tiller to swing to port or starboard. The movement of the tiller closes the steam and exhaust port.

On all the other types of steering some system of follow up gear is used, closing steam valve; the arrangement used by the American Engineering Co., Philadelphia, is shown in Fig. 17 and is described on p. 1218.

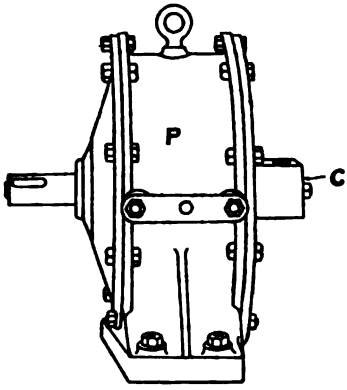


FIG. 10.

Electro-hydraulic Steerer (Hele Shaw) is rapidly coming into favor especially on vessels fitted with Diesel

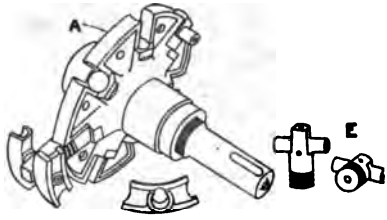


FIG. 11.

engines. A general arrangement and details are shown in Figs. 8 to 16. Like the steam tiller and R. & L. screw gear this steerer is direct connected to the rudder stock, but the torque on the crosshead is transmitted by two hydraulic rams.

In Fig. 8, *A* is the constant running electric motor direct connected to the Hele Shaw pump *B* which will be described in more detail, *C* and *D* are oil cylinders, *E* and *F* the rams driving the rudder crosshead *G* either to port or starboard under an oil pressure of 1000 to 2000 lbs. per sq. in.

The operating crosshead *H* of the Hele Shaw pump is actuated by the transmission from the pilot house through the lever *L* using *M* as fulcrum. The follow up gear connected to the rudder crosshead in *O* will bring the crosshead *H* back in neutral position by levers *P* and *L* now using *N* as fulcrum.

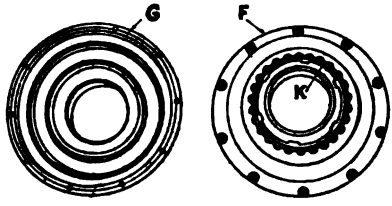


FIG. 12.

In the suction and discharge head *R* of the Hele Shaw pump are located two valves which can be opened by a lever or wheel from the outside in order to make direct communication between suction and discharge in case it would be necessary to use the hand steering gear.

The Hele Shaw Hydraulic Pump is a rotating piston pump; for general arrangement and details see Figs. 9 to 16. Seven cylinders are cast integral on a disk *A* as shown on Fig. 9, giving complete detail of pump. The centre of

this disk is bored to fit the hollow shaft *B*. One end of this shaft extends outside of the housing and has connections for suction and discharge *C*. The transverse section of the shaft is divided in two parts communicating on one end with the suction and discharge connections and on the other end with the two slots *D*. In the bottom of each cylinder is a port which communicates alternately with the two slots in the shaft.

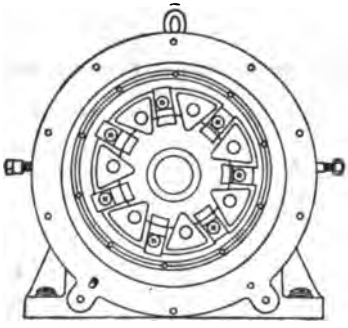


FIG. 13.

FIG. 14.

The pistons are provided with cylindrical supports *E* for guides, one on each side of the piston. Cylinder casting, pistons and guides together are enclosed in a casing consisting of two parts *F* and *G* bolted together, the whole being supported in this casing by the guides which slide in grooves *H* in the inside of the halves. On the outside of each halve is fitted a roller bearing *K* fitting

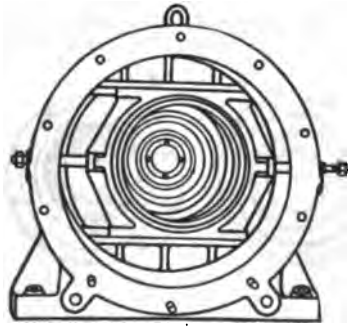


FIG. 15.

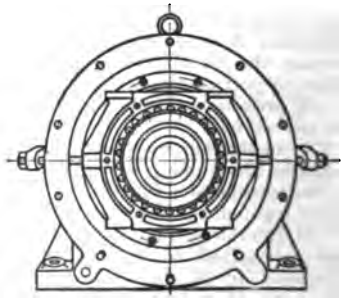


FIG. 16.

in the hexagon frames *L* which slide in guides *M* shown on the inside of the pump frame covers *N*.

These sliding frames are connected on both ends by strips and the whole system can be moved on the guides by a rod brought through either side of the pump frame *P*. When the centre of the sliding frames with the casing and the guided pistons coincides with the center of the shaft, there will

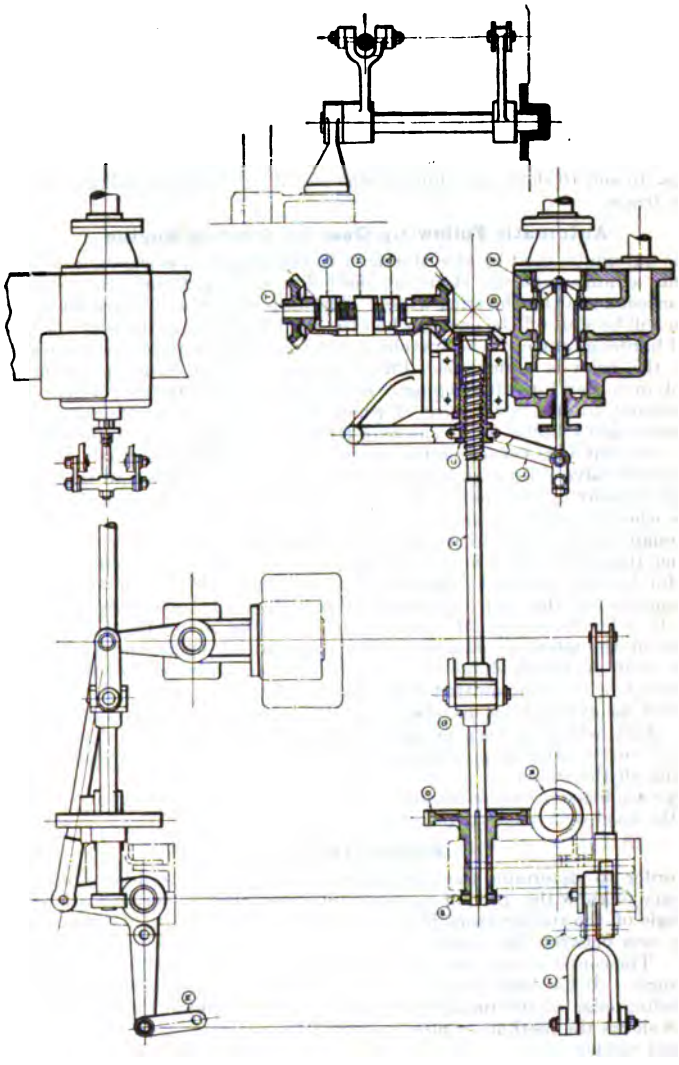


Fig. 17.

be no pumping action; but, as soon as the frame is shifted to one side, the pistons will make a stroke equal to twice the distance the frame is shifted, in order to follow the now eccentrically located grooves for the piston guides. The pump will be reversed by shifting the frame to the other side from the neutral position.

Fig. 10 shows side elevation of the assembled pump.

Figs. 13 and 14 show both sliding frames in place with casing—containing cylinders, pistons, etc. between the two.

Figs. 15 and 16 show one sliding frame and also cylindrical casing in place in pump frame.

Automatic Follow-up Gear for Steering Engine

The automatic used by the American Engineering Co. is shown in Fig. 17. In this arrangement the stop nut shaft T is actuated through a suitable transmission gear by the trick wheel. The stop nuts S_1 and S_2 are set so that steam will be shut off the engine when the rudder arrives at the port and starboard hardover angle. The traveling nut H is set midway between the stops when the helm is amidships. Pin N is removed for hand operation and placed in a rack for safe keeping. Any movement of the bevel gear A is transmitted to the bevel gear B which has a sleeve cast integral with it. The sleeve nut C slides in the sleeve of B and is also fixed so as to turn with it. When the nut C is turned on the screw end of the automatic shaft, it moves the control valve F through lever L out or in, admitting steam to the engine. The movement of the engine, through the worm X on the crank shaft to worm wheel G and automatic shaft K, returns the nut C and control valve F through lever L back to its original position, shutting off the steam.

When the control valve is operated by a telemotor, the pin P is removed and used for locking the trick wheel shaft. The trick wheel is mounted on a continuation of the stop nut shaft. Pin N should be placed in position in lever M. The lever arm M is connected to the telemotor links R and the motion of the telemotor is transmitted through the levers and trunnion Q to the shaft K which slides the sleeve C, in an endwise direction. Any movement of C is transmitted through lever L to control valve F and steam admitted to cylinder. This starts up the engine rotating worm X on the crank shaft which in turn actuates worm wheel G and shaft K returning sleeve C to its original position and through lever L, closing the valve F, shutting off the steam.

Stops are fitted on the telemotor to prevent its operating the control valves after the hard-over angle is reached.

Rudder Torque

In order to determine the twisting movement on the rudder stock it is necessary to know the speed of the vessel, the immersed rudder area, the working angle of the rudder from *H.O.* to *H.O.* (usually 35 deg.), and the length of the arm between the centre of pressure and the centre line of the rudder stock. The centre of pressure shifts as the helm passes from amidship to the *H.O.* angle. It has been found that at 35 deg. it is approximately 0.37 from the leading edge of the breadth of the rectangle equalling the rudder area. Fig. 18 shows the method employed to find the centre of pressure of a semi-balanced rudder.

P = Pressure in lbs.

A = Immersed area of rudder in sq. ft.

V = Speed of vessel in knots per hr. plus 10 per cent for full cargo vessels and 20 per cent for boats of finer lines, to allow for slip.

a = Working angle of helm (taken at 35 deg. for merchant vessels).
p = Pressure per sq. ft. at 1 knot = 3.19 lbs. per sq. ft.
R = Arm in in. from centre of pressure to ℄ of rudder stock.
$P = AV^2 \times \text{Sin } a \times p$.
$T.M.$ = Twisting moment at rudder stock in. lbs. $= P. \times R$.

To find centre of pressure semi-balanced rudder (Fig. 18):

R = Centre of pressure to ℄ of rudder stock.
$A = \Sigma a$ = immersed area of rudder.
JK = Leading edge top section going ahead.
b_1 = Width of top section. h_1 height of top section.
$l_1 = 0.37\, b_1$ = Center of pressure of top section from leading edge going ahead.
e_1 = Distance from leading edge of top section to ℄ of rudder stock.
$r_1 = l_1 + e_1$ = Distance from centre of pressure to ℄ of rudder stock.
$a_1 = h_1 b_1$ = Area top section.
b_2 = Width of bottom section h_2 = height bottom section.
$l_2 = 0.37 b_2$ = Centre of pressure of bottom section from leading edge mn going ahead.
$r_2 = d - l_2$ = Centre of pressure from ℄ of rudder stock.
$a_2 = h_2 b_2$ = Area bottom section.

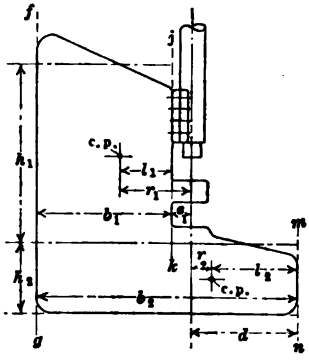


FIG. 18.

Taking moments about rudder stock:

$$A.R. = \Sigma ar = a_1 r_1 + a_2 r_2 + a_n r_n = a_1 r_1 - a_2 r_2 + \text{etc.}$$

$$R = \frac{\Sigma ar}{A}$$

The centre of pressure going astern is found in a similar manner fg being the leading edge.

Size of Steering Gears

In determining the size of the steering engine it should be borne in mind that the maximum power is required only when the rudder is hardover, and that the engine is constantly stopping and starting from rest. The valves of the engine have no lap, the eccentrics being set at 90 deg. with cranks.

The stress and bending moment coming on all parts of the steering gear should be carefully analyzed, special attention being paid to the stress on the gear teeth and keys and the parts made strong enough to stall the engine without injury.

Table 1 gives the details of several installations of steering gears which have proven satisfactory in service. The item Pp was used for comparing various gears and does not take into account any efficiencies:

Pressure on piston: Pp	
Moment on rudder stock in lbs.-in.	$= M$
Radius of crosshead in in.	$= R$
Pressure on crosshead pin in lbs. $\frac{M}{R}$	$= Px$
Travel driving nut HO–HO approx. equal to travel of crosshead pin parallel to screw shaft $= 2R \times \text{Sin } A$	$= Tx$
$Px \times Tx$	$= Wx$
Stroke	$= S$

Table 1. Steering Gears

Length ship	D. W. tons	Speed knots loaded	Size	Rudder area sq. ft.	Distance C pressure to ℄ pintles ahead	Radius quadrant	Diam. drum	Ratio worm and gear	No. rev. HO to	No. rev Eng. HO to HO	Pressure on rudder lbs.	Mom. on rudder stock in lbs. M	Travel piston HO to HO in.	Pressure piston sq. in. Pp.
						Drum Type Steering Gear								
150′	Ocean Tug	12	6× 6	68.0	2′ 0″	5′ 2″	12½″	60: 1	2.2	132	28,800	691,200	1,585″	21.7
369′	7,500	10½	10× 8	161.20	30.6″	6′ 8″	18″	51: 1	1.725	88	47,000	1,438,200	1,760″	19.8
						R. & L. Screw Type Steering Gear								
						Gear and screw	Rev. screw or pin shaft	Eng. rev.	Rudder stock	Rad. crosshead				
335′ 6″	5,000	10½	7× 7	117.00	27″	5.56: 1 1¼P	17.90	99.500	9″	19½″	34,000	917,500	1,390″	19.75
423′ 9″	8,800	10½	8× 7	143.00	38″	5.56: 1 1¼P	19.5	108	10½″	19″	46,500	1,770,000	1,510″	29.8
422′ 0″	8,800	10¾	8× 8	173.00	31.3″	7.5: 1screw 1″P	25.7	193	10¾″	20″	59,300	1,838,300	3,090″	15.2
427′ 0″	8,800	10¾	8× 8	173.00	31.3″	4.7: 1	15.25	71.6	....	19″	59,300	1,860,000	1,145″	39.5
423′ 9″	8,800	11½	8× 8	134.00	28	4.26: 1	20.6	87.500	10″	20″	52,300	1,465,000	1,400″	26.8
450′ 0″	10,000	10¾	9× 9	178.00	36	5.56: 1	22	122	12¾″	24″	54,000	1,942,000	2,200″	16.00
416′ 0″	9,500	11	9× 9	172.00	38	5.56: 1	22	122	12½″	24″	55,000	2,090,000	2,200″	17.20
535′ 0″	13,000	18ahd 12ast	15×12	289.52	3′6″ahd 7′ 11″ast	4.8: 1	19.8	95	....	2′ 10½″	ahd ast	9,260,000 9,300,000	2,280″	26.7
535 ′0″	13,000	18ahd 12ast	16×12	289.52	42″	2″lead 3.56: 1 2	19.5	69.328	20″	2′ 10″	ahd ast	9,260,000 9,300,000	1,665″	31.6
						Tiller Type Steering Gear								
						Rad. rack	Rev. pinion	Worm ratio	Rev. engine					
404′ 0″	7,500	11	7½×12	153.40	36.8″	20′ 0″	3.8	25.1	95		52,500	1,940,000		25.6

Diam. of cylinder in in. = Dc
Area of piston in sq. in. = Ap
Revolutions of engine from hardover to hardover = Re
Piston travel (in in.) HO to $HO - Re \times 2 \times S$ = Tp
Piston pressure per sq. in. (figuring 1 cylinder doing all work at 100 per cent efficiency) = Pp

Then $Ap \times Pp \times Tp = Wp$ and $Wp = Wx$

$$Pp = \frac{Wx}{Ap \times Tp}.$$

Formula to Determine Approximate Size of Screw-gear Type Steam Steering Engine for Given Twisting Moment on Rudder Stock

M = Twisting moment on rudder stock (in lbs.-in.).
R = Radius of rudder crosshead (in in.).
A = Working angle of rudder each side of ℄ of ship.
L = Lead of thread on screw shaft (in in.).
G = Gear ratio (screw shaft to crankshaft).
D = Diam. of engine cylinder (in in.).
S = Stroke of piston (in in.) assume $S = D$.
P = Effective pressure on piston (in lbs. per sq. in.).
= Abs. steam pressure − abs. exhaust pressure. Assume 100 lbs. per sq. in.
E = Total efficiency of gear, assume 0.135.
$E = e_1 \times e_2 \times e_3 \times e_4$ where
e_1 = Efficiency of rudder and step bearings, assume 0.75.
e_2 = Efficiency of screw gear and parts, assume 0.25.
e_3 = Efficiency of two cut spur gear reductions, assume 0.90.
e_4 = Efficiency of engine running parts, assume 0.80.
T = Torque on crankshaft required by gear.
T_e = Engine torque.

$$T = \frac{ML}{REG\ 2\pi \cos A} \qquad T = T_e.$$

$T_e = 0.3927PSD^2$ and when $S = D$ $T_e = 0.3927PD^3$.

$$D = \sqrt[3]{\frac{ML}{2.467REGP \cos A}}.$$

Using values given for E and P, and $A = 35$ deg.

$$D = \sqrt[3]{\frac{ML}{27.28\ RG}}.$$

Example 1. To determine size of a screw gear type steam steering engine, suitable for a vessel having a 9 in. diameter, steel rudder stock, with a twisting moment of 917,500 lbs.-in. (In this case it is desired to have the bore and stroke of the engine the same.)

D = Diameter of cylinder = $\sqrt[3]{\frac{ML}{2.467\ REG \cos A}}$.
M = Twisting moment (given above) = 917,500 lbs.-in.
L = Lead of thread. This may be assumed to be 1¼ in.
R = Radius of crosshead pins. Assume to be 2.17 × diameter of rudder stock = 2.17 × 9 in. = 19½ in.
E = Total efficiency. Assume 13.5 per cent for this type of gear.
G = Gear ratio. Assume 5.56.
P = Effective steam pressure. Assume 100 lbs. per sq. in.
A = Working angle of rudder. This is usually 35 deg.

$$D = \sqrt[3]{\frac{917{,}500 \times 1.25}{2.467 \times 19.5 \times 0.135 \times 5.56 \times 100 \times 0.819}} = 7.3 \text{ or } 7 \text{ in.}$$

Using an engine having the same bore and stroke.

A 7 × 7 engine will be suitable for this gear.

Example 2. In case it is desired to have the bore and stroke of the engine different, i.e., the stroke 0.875 × the diameter the formula $Te = 0.3927\,PSD^2$ becomes $Te = 0.3927\,P\,0.875\,D^3 = 0.344\,PD^3$.

Again using values given for E and P and $A = 35$ deg.

$$D = \sqrt[3]{\frac{ML}{23.87\,RG}}$$

For same values of M, L, R and S as in example 1.

$$D = \sqrt[3]{\frac{917{,}500 \times 1.25}{23.87 \times 19.5 \times 5.56}} = 7.62 \text{ in.}$$

$S = 0.875 \times 7.62 = 6.67$ in.

Use $D = 7\frac{5}{8}$ in. and $S = 6\frac{3}{4}$ in.

When the approximate size of the engine has been determined, the power of the engine with the efficiencies allowed, should be checked up to see if it is capable of putting the helm "*HO*" to "*HO*" in 30 sec.; this being the time usually specified for Merchant vessels.

Installation of Steering Gears

In new installations the following points should be carefully noted:

1. The quadrant should mesh accurately with pinion on worm gear shaft of the engine without excessive backlash.

2. The extreme angle of rudder and working angle each side of ℄ should be clearly marked on quadrant, in the L. & R. gear the marks should be made on the guide bars.

3. The engine should be run over slowly to determine that stops of the automatic follow-up gear are set properly and that steam is cut off the engine when extreme working angle is reached both sides of mid position.

4. When a spring quadrant is used the floating parts are to have clearance for movement when the springs are compressed or released.

5. Where a friction clutch of the expanding type is fitted, this clutch should be tightened so that it will not slip when the engine is exerting its maximum power. In case of a severe storm or a heavy sea coming astern and causing great pressure on the rudder, care should be used to see that the friction clutch does not slip, continuously.

6. When it is desired to operate the hand steering gear, it is always advisable to lock and unlock the pinion when the quadrant is in mid position and the engine is at rest, thereby avoiding any possibility of disarranging the follow-up gear and causing unequal travel to starboard or port positions of the helm angle.

Steering Gear Control

Shafting Control. On small vessels, and on vessels where the steering gear is located amidship the control is generally obtained by means of shafting and gears. Where the leads are direct and not too long this is a very satisfactory and positive control. Care should be taken to install a sufficient number of bearings and to have the shafting truly lined up and easily accessible for lubricating and repairs. The gears should be of bronze and have machine cut teeth. Connection is made direct to the automatic stop nut shaft on the steering engine.

Wire Rope Control. When direct leads cannot be obtained a very satisfactory control can be obtained by using wire rope. The wire rope can be further protected by running it through pipes, which are sometimes run through the holds. The classification societies prefer them to be run on deck where they are more accessible, though more liable to damage and cor-

rosion. With this control provision should be made for automatically taking up the stretch of the wire. Fig. 19 shows a Hanscom drum which is made up of two parts with adjustable springs and has been designed with this end in view.

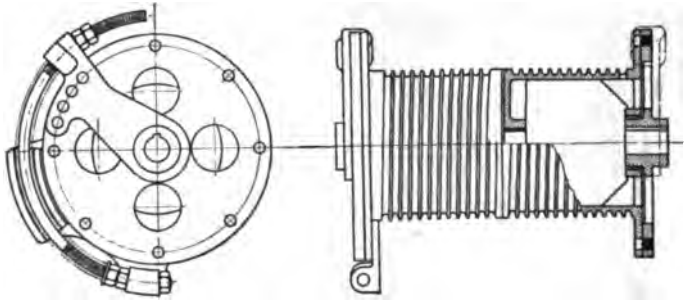


FIG. 19.

Telemotor Control. On large vessels which have their steering gears located aft the friction developed in the shaft and wire rope control becomes excessive, and it is necessary on this type vessel to install a telemotor control.

A general arrangement of a **Brown type hydraulic telemotor** is shown in Fig. 20. This equipment consists of two hydraulic cylinders *A* and *K*, one located in the pilot house, and the other close to the steering gear. The piston *B* in cylinder *A* is displaced by the trick wheel *S* working through the gearing as shown. This displacement is communicated to the piston *N* in cylinder *K* and the movement communicated through pin *P* and suitable levers to steam valve on the steering engine. Any movement of *P* will compress or elongate the springs *Q*. As soon as the trick wheel *S* is released the springs will return piston *N* to neutral position and this displacement will in turn be communicated to the forward piston returning it also to neutral position.

Should the zero position of the steering wheel not correspond with the zero position of the helm, the wheel should be put to the zero position this is all that is required in some makes of telemotor, in others it is necessary to open a by-pass valve which is kept open until the helm reaches the neutral position.

The pump for charging the system with liquid is shown at *Y*.

Troubles. The principal source of trouble met with in the hydraulic telemotor has been due to air in the system. The instructions issued with the various makes of telemotor for getting rid of the air should be consulted.

Leaks in the piping connections are another source of trouble, various types of couplings which have been successfully used to avoid this trouble are shown in Fig. 21.

The fluid pressure need never exceed 250 lbs. per square inch.

The piping should not be run in locations where there are great variations in temperature, and sharp bends, or pockets which are likely to form air traps should be avoided.

Mixture. In tropical climates the system should be filled with clean water but in colder climates a mixture of water with glycerine or telemotor oil should be used.

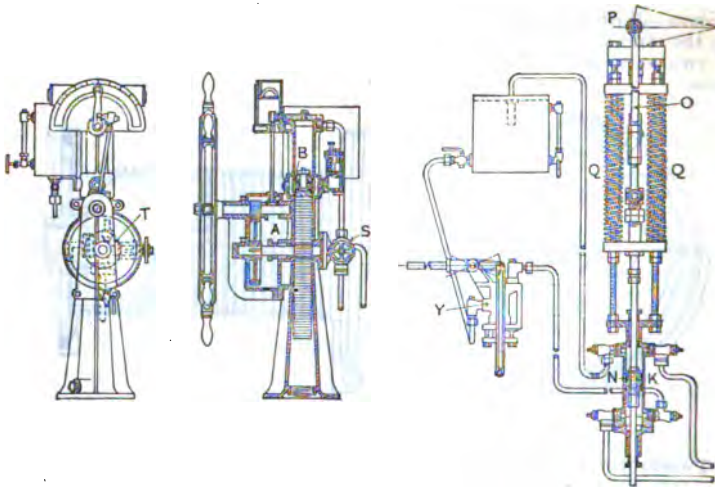


FIG. 20.—Brown Hydraulic Telemotor.

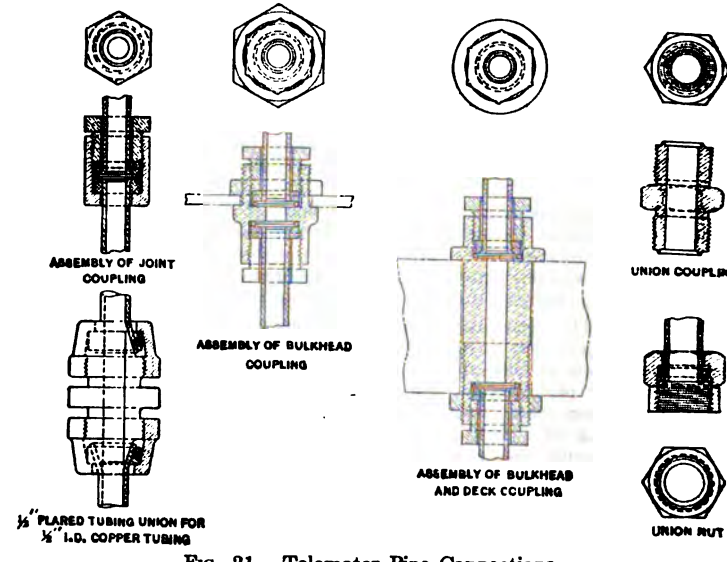

ASSEMBLY OF JOINT COUPLING
ASSEMBLY OF BULKHEAD COUPLING
ASSEMBLY OF BULKHEAD AND DECK COUPLING
UNION COUPLING
½" FLARED TUBING UNION FOR ½" I.D. COPPER TUBING
UNION NUT

FIG. 21.—Telemotor Pipe Connections.

Telemotor Non-freezing Mixture of Water and Glycerine

Per cent glycerine in mixture	25	33	50	60
Safe working temp., deg. F	+13	+10	−20	−30

Any mixture over 60 per cent glycerine is too thick to operate properly. Telemotor oil starts to congeal at about 15 deg. F.

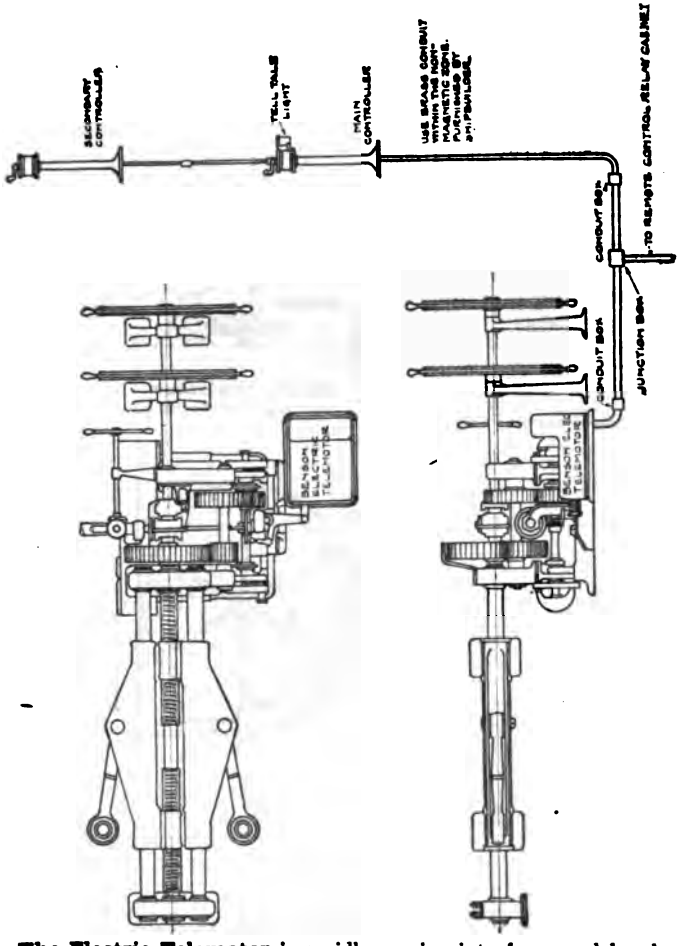


FIG. 22.

The Electric Telemotor is rapidly coming into favor and has been installed on quite a number of vessels with very satisfactory results. Figs. 22 to 25 show the general application and the wiring diagrams of the Benson

electric telemotor. The elements of this telemotor consist of (1) a main controller located in the wheel house, (2) a secondary controller on the bridge (optional), (3) an after telemotor, follow-up disk and motor connected to the steering engine through suitable gears and crosshead, (4) a remote control

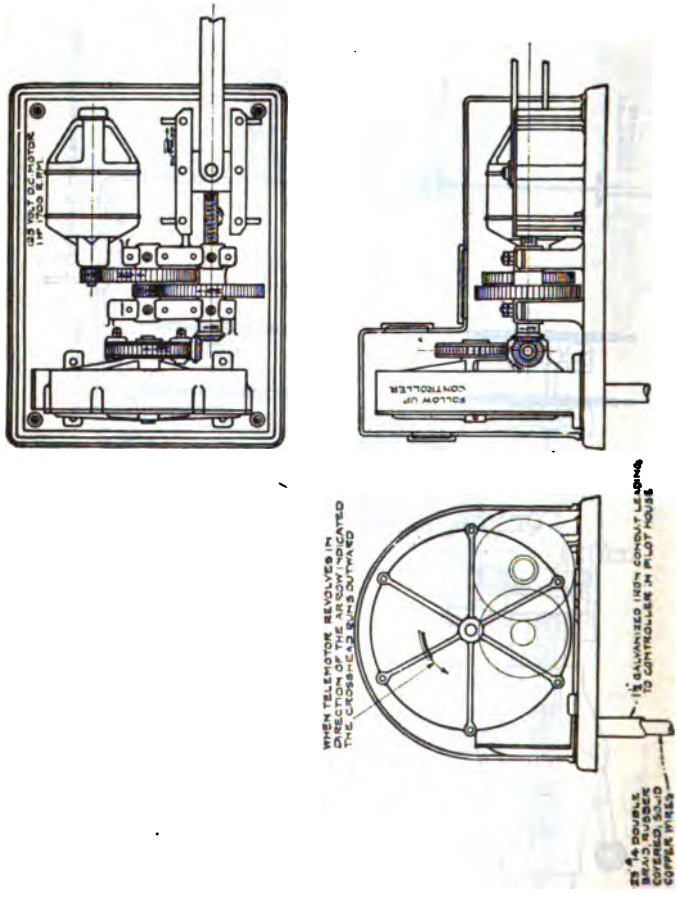

WHEN TELEMOTOR REVOLVES IN DIRECTION OF THE ARROW INDICATED THE CROSSHEAD MOVES OUTWARD
¾" GALVANIZED IRON CONDUIT LEADING TO CONTROLLER IN PILOT HOUSE
23 #18 DOUBLE BRAID RUBBER COVERED SOLID COPPER WIRES

FIG. 23.

relay cabinet, located near the main switch board, (5) cable and conduit necessary to connect the various elements electrically.

Rudder movement is obtained by rotating the controller lever thus making contact between the various points on the contactor and the common

return wire. There are fourteen contact points on the contact disk, seven (7) on either side of midship, corresponding to the following rudder angles:

Contact No.	Degrees of travel	Total angle
1	3	3
2	3	6
3	3	9
4	5	14
5	10	24
6	10	34
7	10 or H.O.	44 or H.O.

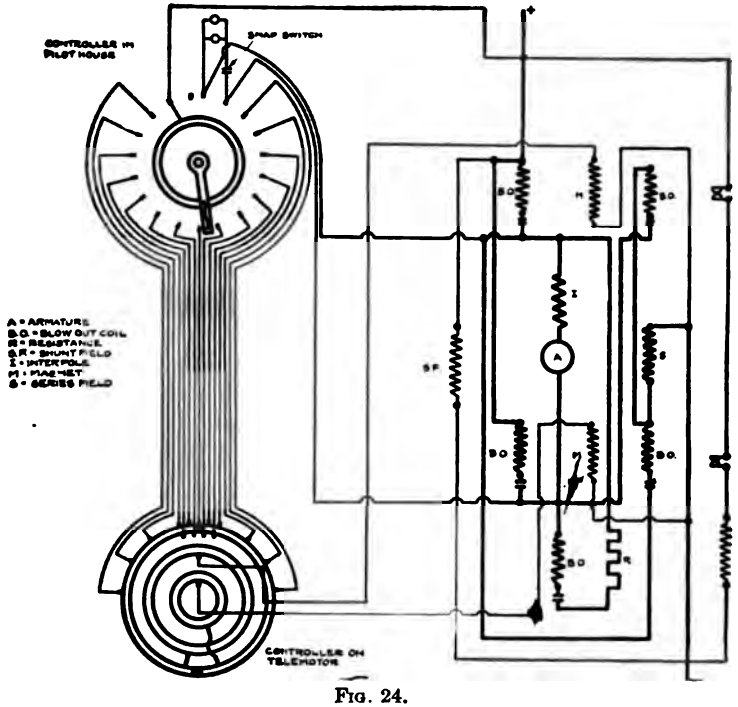


FIG. 24.

The entire mechanism is so arranged that the rudder is always on the same side of the ship's centre line as the controller handle. The controller handle can be rotated from any position to any other desired position on the contact disk, it may be rotated from hardover to hardover but the rudder will always follow until it has reached a position corresponding to the position of the controller handle. No reversing switch or other mechanical mechanism is required to reverse the direction of travel of the rudder.

From the base of the controller, conduit is run to the after telemotor and to the relay cabinet through a junction box.

The after telemotor has a heavy cast iron base, on this base are cast crosshead ways, gear bearings and pads for a motor. A 1-H.P. motor is mounted on this base and drives a crosshead by means of a train of gears and a screw shaft. The same gears also drive a worm shaft which engages a worm gear that in turn rotates the follow-up disk in the after telemotor.

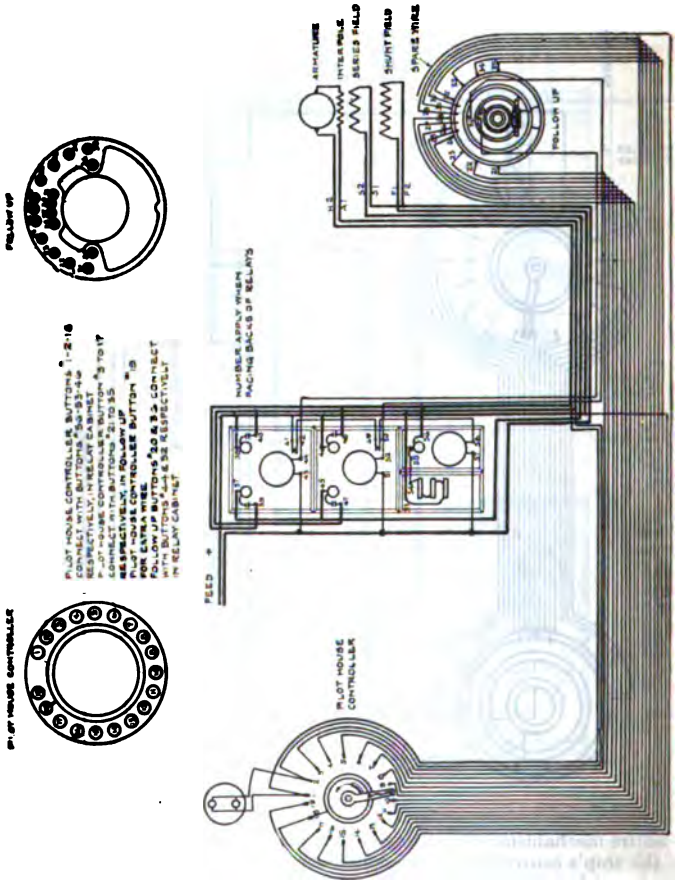


FIG. 25.

The crosshead of the telemotor is connected to the reverse valve of the steering engine by a suitable rocker arm.

A relay cabinet is placed in any convenient place in the engine room. In this cabinet are two double pole and one single pole remote control relays.

When the controller lever is placed on any contact in the controller an electrical circuit is completed through a contact on the follow-up disk to a

controller ring in the follow-up casing and from there to one of the remote control relays selected by the direction in which the lever is moved. This immediately closes the primary circuit through the motor, starting it running

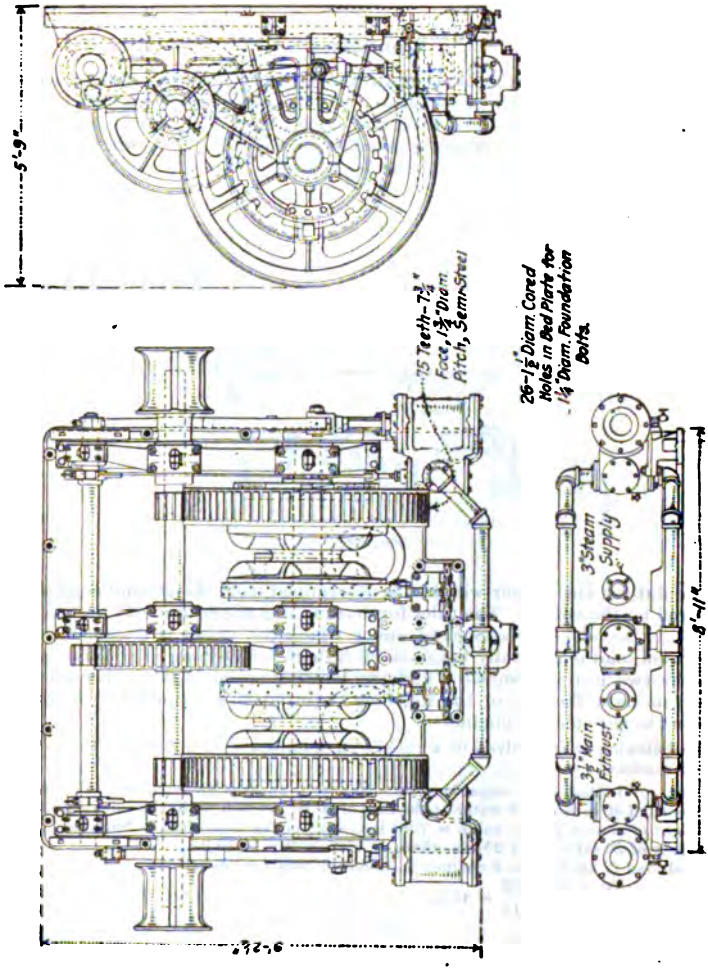

5'-9"
75 Teeth-7½" Face, 1½" Diam. Pitch, Semi-Steel
26-1½" Diam. Cored Holes in Bed Plate for 1¼" Diam. Foundation Bolts
3" Steam Supply
3½" Main Exhaust
8'-11"
9'-2½"

FIG. 26.

in that direction. As the motor runs it operates the crosshead and also the follow-up disk. When the follow-up disk reaches the position corresponding to the controller lever the circuit is open through the relay causing the motor and the crosshead block to stop at this position. At the same

time the single pole relay closes and places a dynamic brake on the motor to insure its stopping instantly at the right place. It stays in this position until the controller lever is shifted when the operation is repeated.

The motor is an interpolar compound wound machine, running equally well in either direction.

Windlasses

Windlasses are generally of the **spur geared type** either compound with a gear ratio of about 30:1 or triple geared with a ratio of 50:1; Fig. 26 shows a **compound geared** windlass with two separate wildcat shafts and gears; Figs. 27 and 28 show a **triple geared** windlass; Fig. 29 a **worm driven** windlass with capstan engines on the deck below.

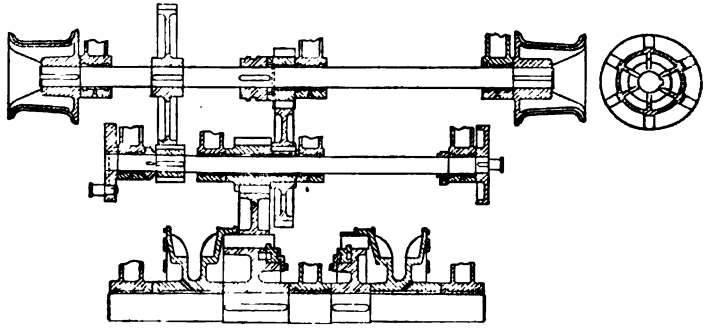


FIG. 27.

The duty of the anchor windlass is determined from the ground equipment required by the vessel. The rules for determining the equipment tonnage are given in the rules of the Society under which the vessel is classified. The maximum load is generally taken as 60 fathoms of the size of chain required and the two heaviest anchors, and the hoisting speed at 30 ft. per minute, with this data the size of the machine required is determined in a similar manner to the steering engine.

The following is an analysis of a vessel's equipment and of a 10 in. × 14 in. triple geared windlass.

NOTE: Windlass to be capable of handling both anchors and chains simultaneously in a depth of 30 fathoms of water at the rate of 30 ft. per min. at 100 lbs. steam pressure.

Weight of anchor 2¾-in. cable = 7084 lbs. + 25 per cent for drag = 8855 lbs.

Weight of 30 fathoms of 2⅝-in. chain = 4725 × 2 = 9450 lbs.

Total load (2 anchors + 2 chains) = 2(8855 + 9450) = 36,610 lbs.

Gear ratio $\frac{60 \times 38 \times 62}{14 \times 17 \times 15} = 45.5.$

Pitch diameter of five lug wildcat = $\frac{10 \times 4T}{\pi} = 12.732T.$

R.P.M. of wildcat $\frac{30 \times 12}{10(14\frac{1}{4} - 4\frac{3}{4})} = 3.79.$

R.P.M. of engine 3.79 × 45.5 = 172.4.

Piston speed of engine $\frac{172.4 \times 2 \times 14}{12} = 402$ ft. per. min.

Steam pressure, 100 lbs.; back pressure, 5 lbs.; card factor, 0.6.
M.E.P. = 0.6 (100 − 5) = 57 lbs. per sq. in.

$$\text{H.P. of engine} = \frac{78.54 \times 402 \times 2 \times 57}{33{,}000} = 109.$$

Assuming the efficiency of cut gear and bearings as 0.95 and of cast gears and bearings as 0.90.

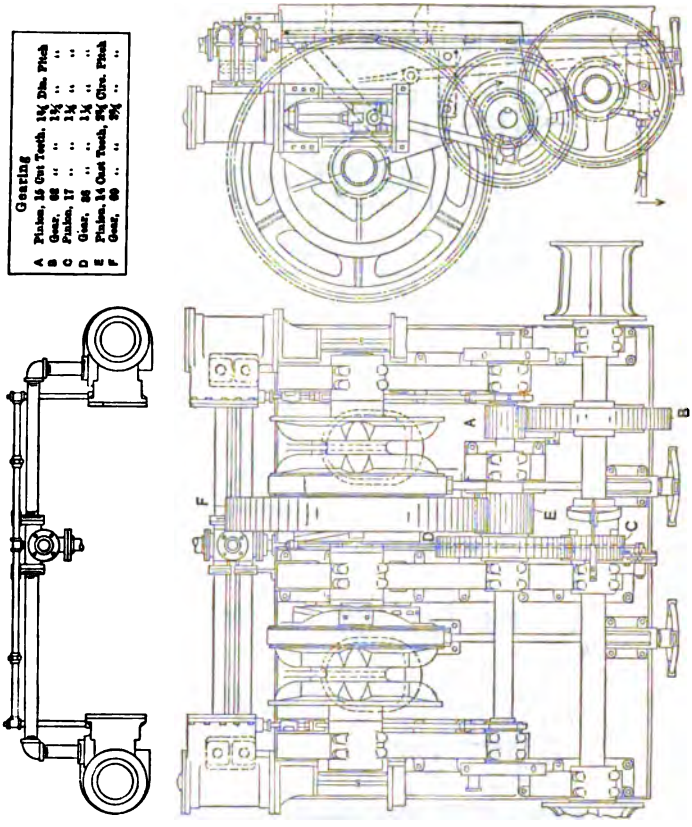

Gearing
A Pinion, 15 Cut Teeth, 1¼ Dia. Pitch
B Gear, 68 " " 1½ " "
C Pinion, 17 " " 1¼ " "
D Gear, 86 " " 1½ " "
E Pinion, 14 Cast Teeth, 2½ Circ. Pitch
F Gear, 60 " " 2½ " "
A
B
C
D
E
F

FIG. 28.

Efficiency of machine 0.95 × 0.95 × 0.90 × 81.27.
Efficiency of hawse pipes (assumed) 50 per cent.
Total efficiency = 0.50 × 0.812 = 40.6 per cent.

$$\text{Capacity of machine beyond hawse pipes} = \frac{109 \times 33{,}000 \times 0.406}{30} = 48{,}680 \text{ lb. at 30 ft. per. min.}$$

$$\text{Excess capacity of windlass} = \frac{48{,}680 - 36{,}610}{36{,}610} \times 100 = 33 \text{ per cent.}$$

The strength of the teeth on the gears should be calculated by the "Lewis Formula" and also the combined bending and torque stress on the shafts. All parts should be made strong enough to stall the engine at the maximum

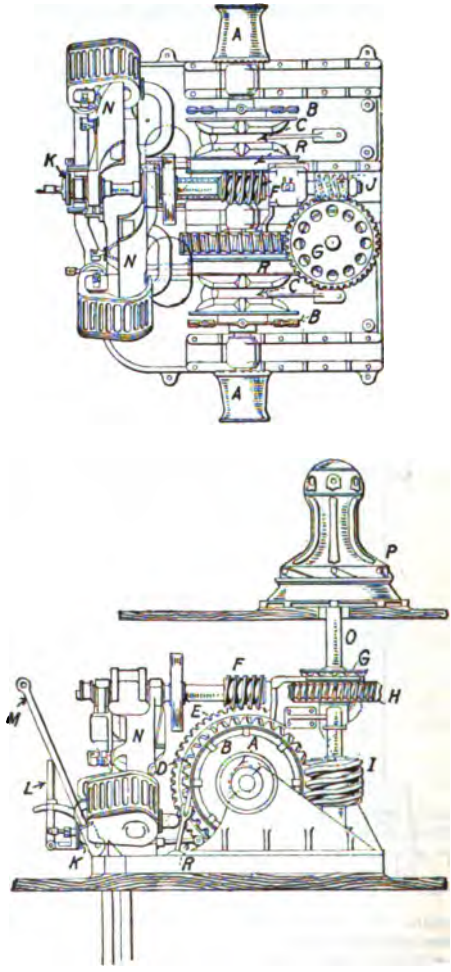


FIG. 29.

steam pressure without failure. Up to 2¼-in. chain the wildcats and gears should be of 15 per cent semi-steel, above 2¼ in. the wildcats and gears should be of cast steel with machine cut teeth in the crank shaft and inter-

mediate gears. The wildcat should be inspected to see that it has lugs of the correct pitch and shape to take the chain without riding the lugs or excessive snapping. If this is not taken care of the side frames are subjected to severe strains. Locking heads must clear the wildcat and also must be readily accessible and easy to lock and unlock.

Table 2 is a table of windlasses which have proven satisfactory in service.

Table 2.

Type	Size	Gear	Rev. wildcat	Rev. engine	Winch head	Diam. chain, in.	No. and size wildcat head, in.	Wildcat pitch diam., in.	Weight anchors	Weight chain	Total load	Rev. pis. travel 1 Fm. chain	Pressure on piston	
													Total	per sq. in.
Comp.	9×12	$\frac{69}{15}\times\frac{78}{13}$	1	27.6	6.00	2	2-14	24.0	5,600	30,100	38,700	4.4-104	4,060	64.0
Comp.	11×12	$\frac{57}{12}\times\frac{75}{13}$	1	27.4	5.77	2¼	2-13½	28.5	7,150	38,200	45,350	3.61-87	6,250	64.1
Trip.	10×10	$\frac{70}{14}\times\frac{44}{22}\times\frac{70}{14}$	1	50.0	5.00	2¼	2-15	28.38	7,150	38,200	45,350	6.73-134	4,060	51.8
Trip.	10×12	$\frac{70}{14}\times\frac{44}{22}\times\frac{70}{14}$	1	50.0	10.00	2½	2-18	28.5	10,045	52,700	62,745	6.71-161	4,675	59.5

Emergency Fleet Corporation Specification for Standard Steam Windlass

Size of Engines, 10 in. × 14 in. Chain, 2¼ in. to 2⅜ in. Stud Link.

Windlass to be of the design and dimensions as shown on drawings. To be of ample strength in all parts for properly handling chains from 2¼ in. diameter to 2⅜ in. diameter inclusive. The windlass proper to be mounted on three bitts and to carry two wildcats, locking heads and locking gear complete and to be driven through three sets of spur gears by a horisontal engine, self contained with windlass on one bedplate.

The windlass under 100 lbs. working pressure to be capable of handling both anchors and chains simultaneously in a depth of 30 fathoms of water, at the rate of 30 feet of chain per minute.

The cylinders and engine parts to be of ample strength, however, to stand full boiler pressure in case of accident to the reducing valve.

Gypsy Heads to be fitted on each end of the intermediate shaft for the quick handling of lines, gypsy heads 16 in. diameter barrel by 18 in. overall.

Bedplate to be of cast iron with raised pads accurately ma-

chined to receive parts bolted thereon. The underside of the bedplate to be strongly ribbed, planed true on the bottom. The bedplate to be provided with a sufficient number of bolt holes for holding down bolts. These holes are to be drilled to template so that all bedplates of windlasses should be interchangeable for securing to deck. The bosses for the bolts are to be spot faced on top for receiving the nuts.

Cylinders to be of cast iron having 10 in. diameter bore by 14 in. stroke. Walls of cylinders to be of ample thickness for reboring. The cylinders are to be securely bolted to the outboard bitts of windlass, cylinders to be tested in shop to 400 pounds hydraulic pressure.

Bearings for supporting the shafts of windlass to be of cast iron lined with white metal composition. They will be securely bolted to the windlass bedplates.

Shafting of windlass to be made of best quality mild steel finished all over.

Wildcats of windlass to be of 15 per cent semi-steel with grabs made to accurately fit the size of chain to be used, chain dimensions to be approved by the American Bureau and Lloyd's Register of Shipping. Each wildcat to be provided with friction bands of ample size and of sufficient strength in all parts to ride by with windlass unlocked.

Locking heads of windlass to be steel castings securely keyed to windlass shaft. Each locking head to be provided with a locking ring and block key arranged for operating by means of a lever.

Gearing. All gears and pinions on the windlass to be steel castings. The gears and pinions on the crank and intermediate shaft to have machine cut teeth.

Chain Stoppers. Two cast iron chain stoppers to be furnished with each windlass to be used for holding chain when heaving in anchor.

All parts of windlass where bolted together should be faced true and spot faced in way of bolt heads and nuts.

All bushings and babbitt used to be grooved for oil or grease. The bushings to be carefully secured by pins or screws.

Guards about ¼ in. thick to be fitted over all gearing of windlass and securely fastened, same to be taken from work.

Automatic drains are to be fitted on the steam cylinders, and all grease cups, and drip cocks are to be provided where called for.

The windlass to be tested under steam at the works of the maker in the manner described in the regulations issued by the Emergency Fleet Corporation for the inspection of windlasses. These regulations to be binding in their entirety and to be followed in the inspection of material during the construction of the windlass.

Before shipment all unfinished surfaces are to be given a coat of iron filler rubbed down smooth and painted with oil paint. All finished parts to be protected by a coat of rust preventing mixture.

Cargo Winches

American practice would seem to favor winches of the **reverse valve type**, as shown in Figs. 30 and 31, while European practice favors the **link motion** type. **Friction drum** winches are also in favor in some quarters, though the advantages claimed for this type are disputed and the British practice of keying the drums on the shaft is growing in favor. The ratio of gearing

in the single winches is about 5:1 and in the compound geared winches 8 to 10:1.

It is essential that the cargo handling equipment be considered as a whole; that is, the size, type and location of winches, size of hatches, method of handling cargo, cargo booms, rigging and falls, and loading cycle.

A loading or unloading cycle may be divided into the following elements, drag, hoist, swing, lowering, return of hook. The equipment should be de-

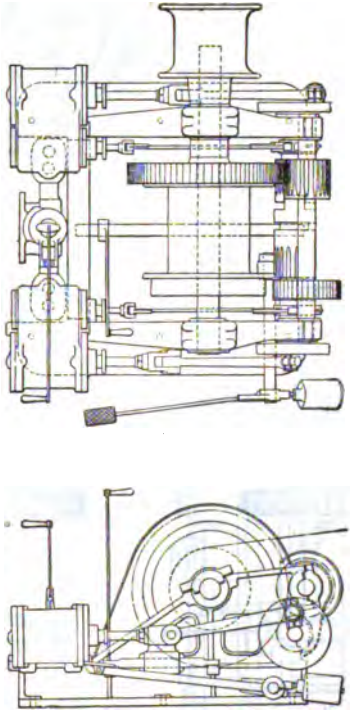


FIG. 30.

signed to cut down the number of men and the time required for each of these operations to the lowest limit. Figs. 32 and 33 show the actual loading cycles attained on two tests.

The highest cargo handling speeds are made with sling loads of about 2000 lbs. which should be hoisted at a speed of between 200 and 250 feet per minute with a working steam pressure of 100 lbs. per square inch, this is as fast as the cargo landed can be stowed away.

In general cargo work the sling loads vary from about 800 lbs. as a minimum to 5000 lb. as a maximum on a single whip, and the diameter of wire rope from ½ in. diam. to ¾ in. diam.

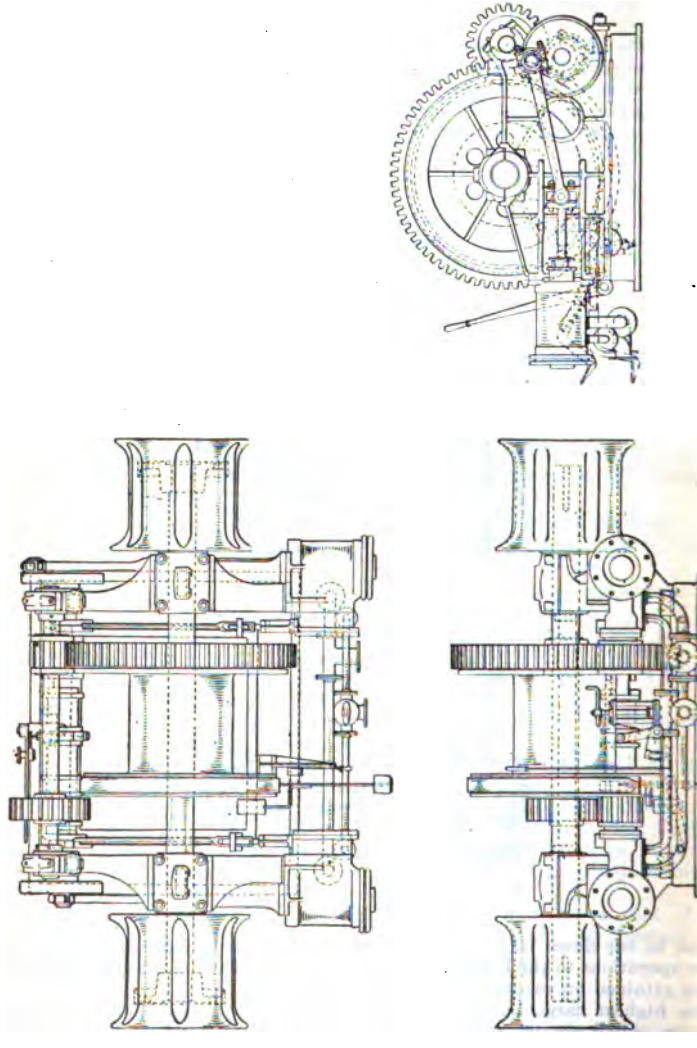


Fig. 31.

Provision is generally made at least at one hatch for hoisting extra heavy loads, by installing larger or compound geared winches, and blocks with the necessary number of falls to take special care of the maximum load; in the case of special loads, speed is generally a minor consideration.

All parts of the winch should be designed strong enough to stall the machine at maximum steam pressure.

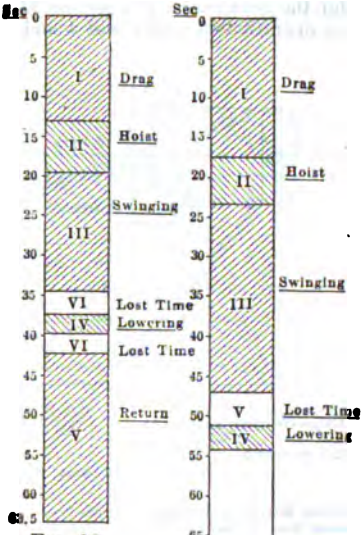


FIG. 32.

FIG. 33.

Graphs of Unloading Cycle. Graphs Fig. 32 and Fig. 33 represent two actual unloading cycles.

Fig. 32 is the cycle timed as an average for unloading corn with two 8¼ × 10 winches using the Burtonning method of cargo handling. The arrangement of winches, booms and blocks is shown in Fig. 34.

Average total time of cycle was 63.5 sec. made up of:

1. Drag of load in hold—13.2 sec. or 20.8 per cent of cycle.
2. Hoist of load, about 26 feet—6.65 sec. or 10.5 per cent of cycle.
3. Swinging outboard of load including time required to hook on the swing line—14.74 sec. or 23.2 per cent of cycle.
4. Lowering of hook about 5 feet—2 sec. or 3.1 per cent of cycle.
5. Return of hook for the next load—21.2 sec. or 33.4 per cent of cycle.
6. Lost time between operations 3 and 4 and 4 and 5—5.7 sec. or 9 per cent of cycle.

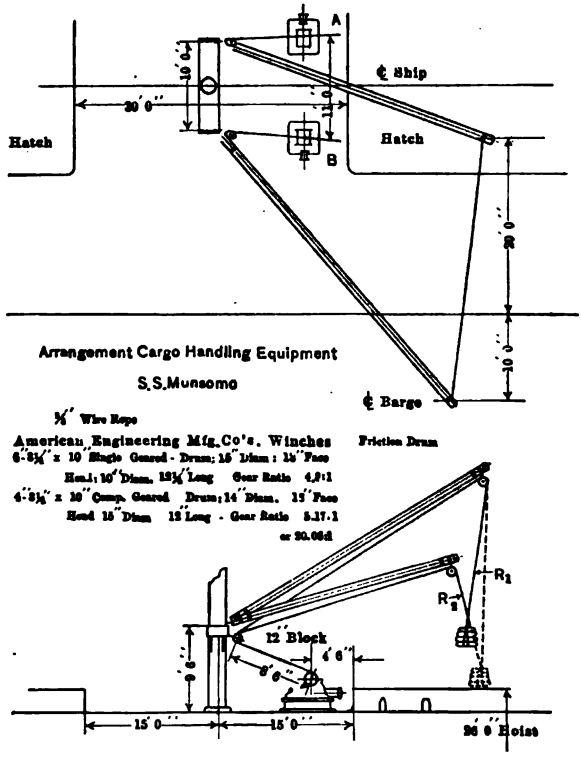


FIG. 34.

With an average load of about 1560 lbs. the total tonnage handled per hatch per hour per two winches derived from the above is 44 tons.

Fig. 33 gives the cycle for unloading linseed with two 8½ × 8 winches.

Fig. 35 shows arrangement of winches and booms.

Total time of cycle was 104.9 sec. made up of:

1. Drag of load—17.2 sec. or 16.4 per cent of cycle.
2. Hoist about 23 ft.—5.8 sec. or 5.54 per cent of cycle.

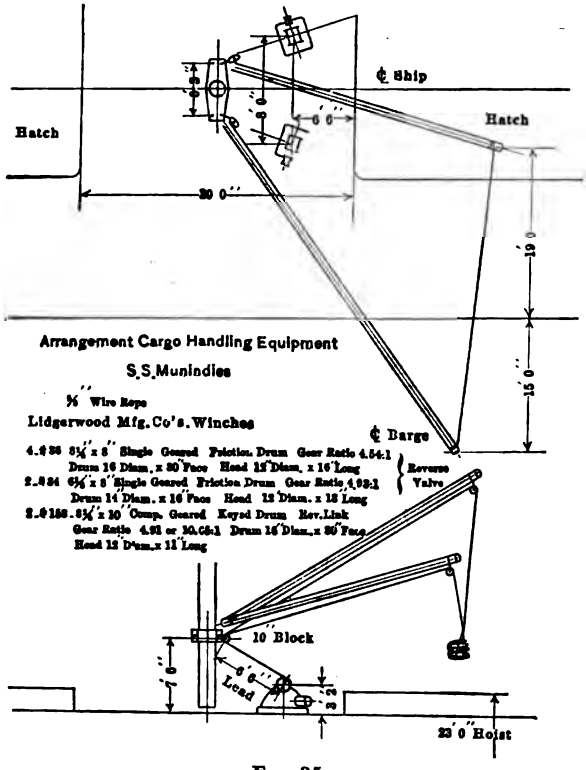

℄ Ship
Hatch
Hatch
6 0"
30 0"
19 0"
15 0"
Arrangement Cargo Handling Equipment
S.S.Munindies
⅝" Wire Rope
Lidgerwood Mfg. Co's. Winches
℄ Barge
4.#86 8½" x 8" Single Geared Friction Drum Gear Ratio 4.54:1
Drum 16 Diam. x 30 Face Head 12 Diam. x 16 Long
Reverse
2.#84 6½" x 8" Single Geared Friction Drum Gear Ratio 4.93:1
Valve
Drum 14 Diam. x 16 Face Head 12 Diam. x 15 Long
2.#188 8½" x 10" Comp. Geared Keyed Drum Rev. Link
Gear Ratio 4.91 or 10.05:1 Drum 16 Diam. x 30 Face
Head 12 Diam. x 11 Long
10" Block
6' 6" Lead
23' 0" Hoist

FIG. 35.

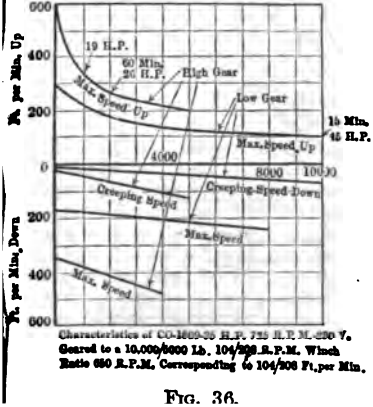


FIG. 36.

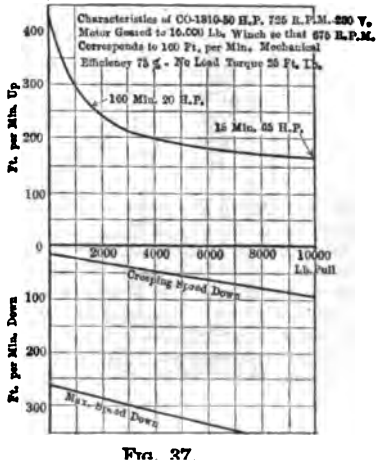


FIG. 37.

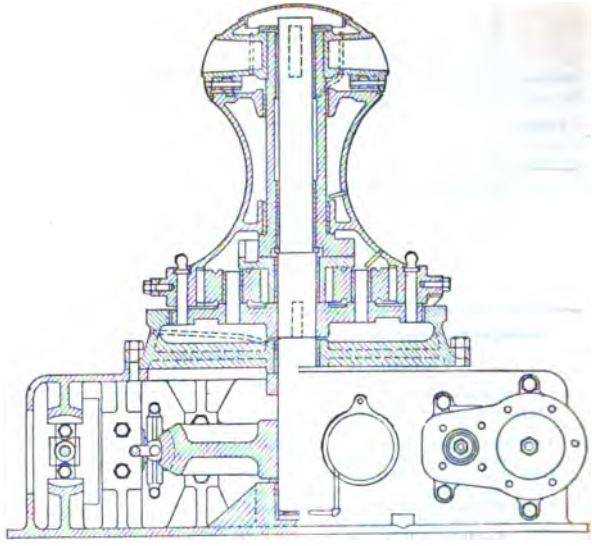


FIG. 38.

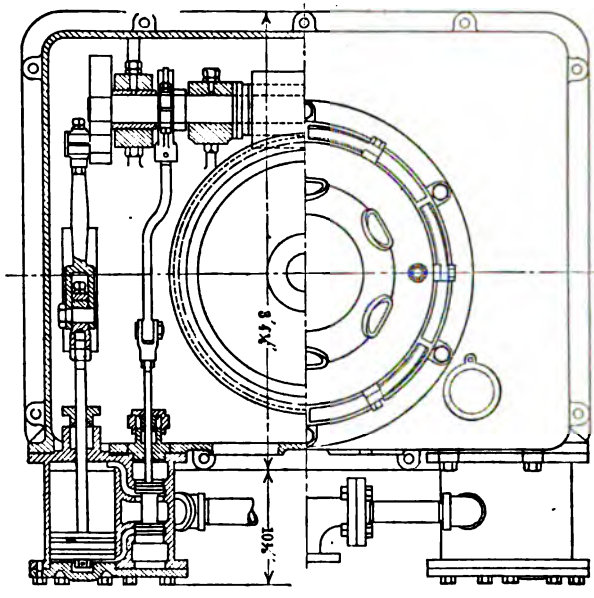


FIG. 39.

3. Swinging including time required to hook on swing line—23.75 sec. or 22.7 per cent of cycle.
4. Lowering of load—3 sec. 2.86 per cent of cycle.
5. Return of hook and lost time 49.75 sec. 52 per cent of cycle.

With an average load of 2240 lbs. the total tonnage handled per hour per hatch per two winches is 37 tons.

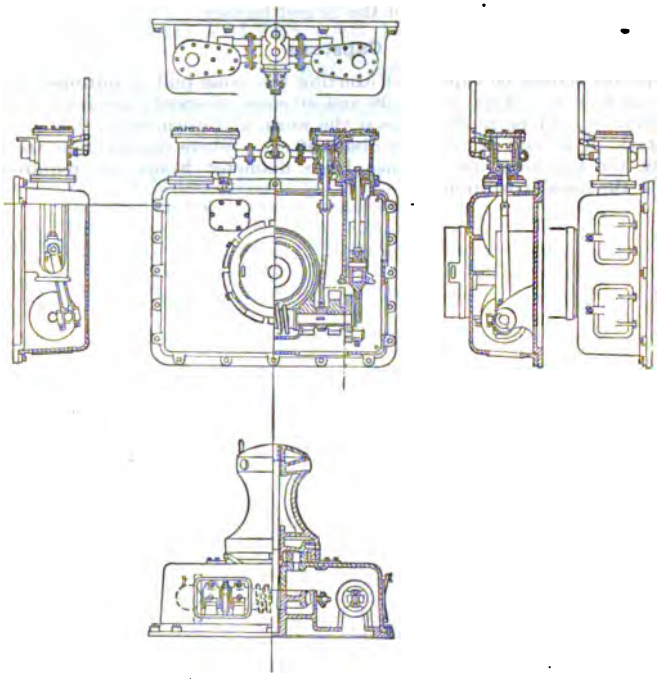


FIG. 40.

Electric Winches

Electric winches are displacing steam winches thus doing away with the littering of the decks with steam pipes. Unfortunately in some of the early installations the motors were underpowered creating the impression that electric winches were slow in operation. This is not correct as the electric winch is more flexible than the steam winch, and it is only a question of designing a motor to meet the duty required. For general cargo handling a motor of at least 40 H.P. 30 minute rating capable of hoisting 6000 lbs. at 200 ft. per minute should be installed. Figs. 36 and 37 show the characteristics of two electrically driven winches designed by the General Electric Company.

PIPINGS, VALVES AND FITTINGS

BY

FRANK W. STERLING

PIPING

REFERENCES: "National" Pipe Standards, 1917, National Tube Co. Valve World, Crane Co.

Piping plans should be carefully prepared with a view to securing the greatest simplicity, directness of lead, and least radiating surface, together with proper security and interchangeability. Special attention should be paid to securing proper **accessibility for repairs** and overhauling, especially for joints and flanges. Fuel-oil piping must be particularly accessible, and, where possible, above floor plates. **Cut-out valves** should be placed where necessary to enable various parts to be overhauled while maintaining boilers, engines, and important auxiliaries in operation. To this end important services should be so arranged that they can take steam from each side of the vessel. Special care will be taken to have pipes properly aligned in their supports without being forced into place, and that flanges are parallel before joint is made.

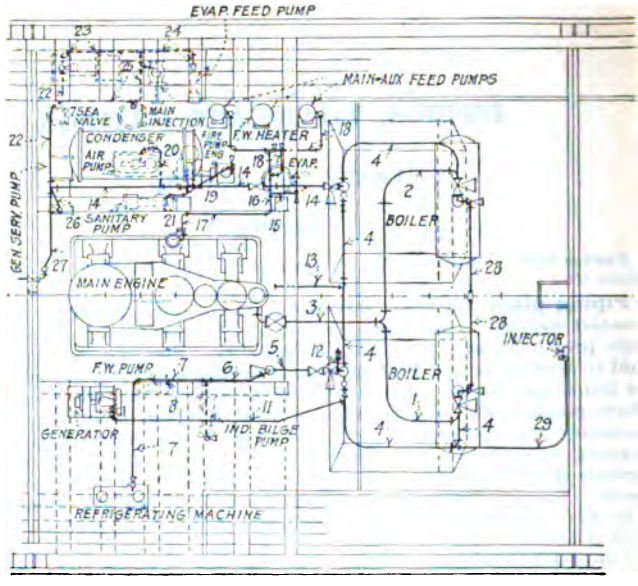
Leads should be direct and bends (except such as are employed to allow for expansion) few and easy. Careful attention should be paid to the location of **anchorages**, so that proper allowance for expansion may be made and any tendency to vibration checked. **Joints** should be as few as is consistent with proper facilities for removal; the lengths of sections of piping should be standardized to the greatest possible extent.

Great care should be taken to insure the possibility of **removing any section** of any pipe without displacing others. The **interlocking** of one pipe with another is to be avoided.

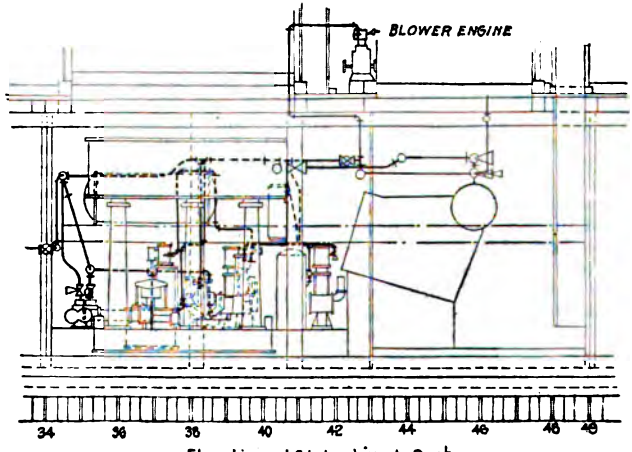
In steam piping, **pockets where water may collect** are to be avoided and precautions should be taken that the lead is such that drainage water will flow back to boilers or to separators. The location of steam-pipe drain connections must be such that pipes can be freed of all collecting water. Particular attention should be paid to the **accessibility of expansion joints** and to the length of sleeves and depth of stuffing boxes to insure that the end of pipe will not draw back within the packing.

Feed suction pipes and bilge suction pipes should be arranged for the best possible **flow of water to pumps**, for absolute tightness of joints, for accessibility and ease of inspection. The location of bilge strainers must be most carefully considered with a view to their accessibility for cleaning. Where possible, feed pipes should be placed well above floor plates. Care should be taken that **all copper pipes in bilges** are led sufficiently high to keep them out of the bilge water under ordinary circumstances. They must not rest in contact with any of the iron or steel work of the vessel.

Steam and water pipes, when it can be avoided, should not lead through distribution rooms. In case any such pipes are led through these spaces, flanges or other connections must not be located over or in the vicinity of switchboards.



Plan
Steam Pipes

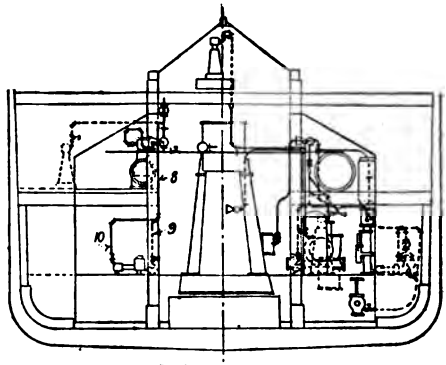


Elevation at C.L. Looking to Port
Steam Pipes

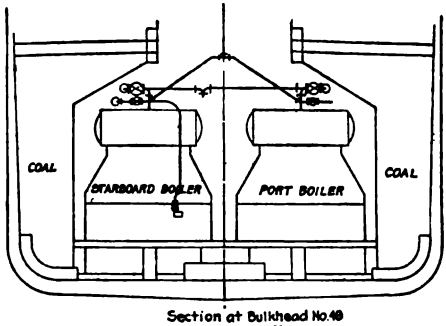
FIG. 1.—Steam Piping in Engine and

List of Pipes

Mark	Size, nom. in.	Service	Mark	Size, nom. in.	Service
1	4½	Main steam starb'd boiler.	16	1¼	Steam to reverse engine.
2	4½	Main steam port boiler.	17	1	Steam to reverse engine.
3	6	Main steam to engine.	18	1½	Steam to feed pump.
4	3½	Aux. steam.	19	1½	Steam to main circ. pump.
5	4	Steam to deck mach. & starb'd auxil's.	20	1¼	Steam to main air pump.
			21	1	Steam to sanitary pump.
6	1½	Steam to starb'd auxil's.	22	2	Steam to fire & bilge pump.
7	1½	Steam to ice mach.	23	1	Steam to sea valve & evap. feed pump.
8	1¼	Steam to fresh water & indep. bilge pump.	24	½	Steam to evap. feed pump.
9	½	Steam to fresh water pump.	25	1	Steam to sea valve
10	1	Steam to indep. bilge pump.	26	2	Steam to gen. service pump.
11	1½	Steam to generator.	27	2	Steam to steer'g. eng. & capstan aft.
12	1½	Steam to smothering line.			
13	1½	Steam to blower engine.	28	2	Steam to whistle.
14	3½	Aux. steam port.	29	1½	Steam to injector.
15	1½	Steam to evaporator.			

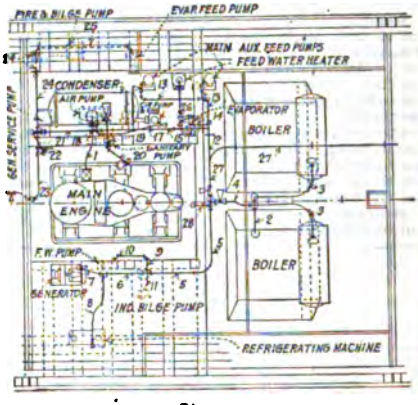


Section at Frame No.43
Looking Aft
Steam Pipes

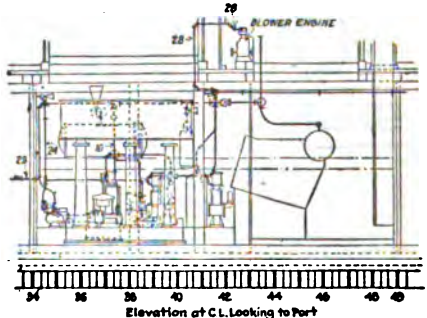


Section at Bulkhead No.49
Looking Aft
Steam Pipes

Boiler Rooms, 3500 ton D. W. Steamer.



Plan
Exhaust Pipes

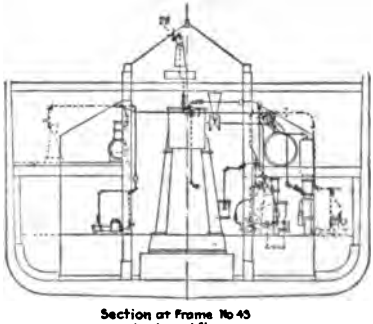


Elevation at C.L. Looking to Port
Exhaust Pipes.

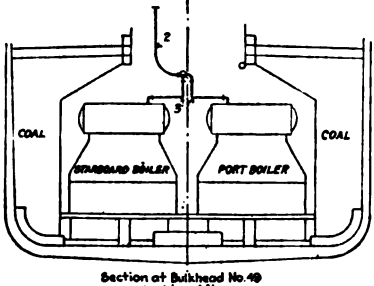
FIG. 2.—Exhaust Piping in Engine and

List of Pipes

Mark	Size, nom. in.	Service	Mark	Size, nom. in.	Service
1	18	Main exhaust to condenser.	16	3	Aux. exhaust port.
2	7	Escape pipe.	17	2	Exhaust from main circ. pump.
3	4½	Safety valve escape.			
4	6	Aux. exhaust to atmosphere.	18	1½	Exhaust from main air pump.
5	3	Exhaust from starb'd auxiliaries.	19	1½	Exhaust from sanitary pump.
			20	1	Exhaust from reverse engine.
6	3	Exhaust from generator & ice mach'y.	21	4½	Aux. exhaust.
			22	2½	Exhaust from gen'l service pump.
7	2	Exhaust from generator.			
8	1½	Exhaust from ice mach'y.	23	2½	Exhaust from steer'g eng. & capstan aft.
9	1½	Exhaust from F.W. & Indp. bilge pump.	24	2½	Exhaust from fire & bilge pump.
10	¾	Exhaust from fresh water pump.	25	¾	Exhaust from evap. feed pump.
11	1½	Exhaust from indp. bilge pump.	26	5	Aux. exhaust to feed water heater.
12	6	Aux. exhaust.	27	4	Exhaust from forw'd deck mach'y.
13	2	Exhaust from feed pumps.			
14	3	Vapor from evaporator.	28	2	Exhaust from blower engine.
15	6	Aux. exhaust port.			

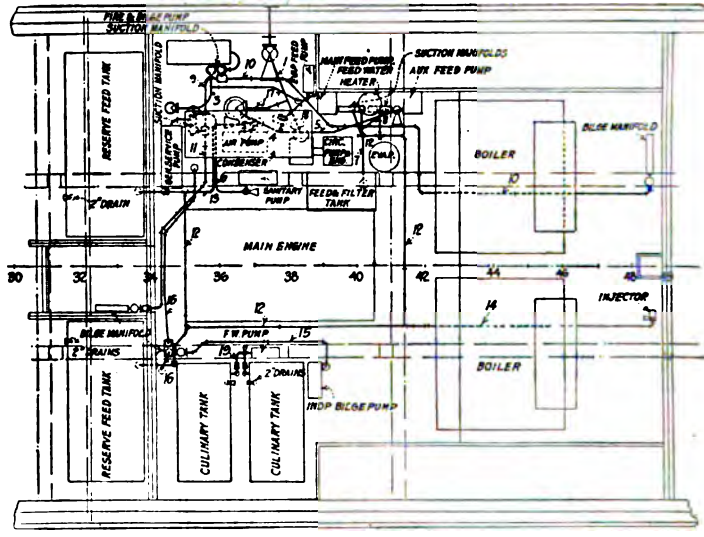


Section at Frame No 43
Looking Aft
Exhaust Pipes.



Section at Bulkhead No.49
Looking Aft
Exhaust Pipes.

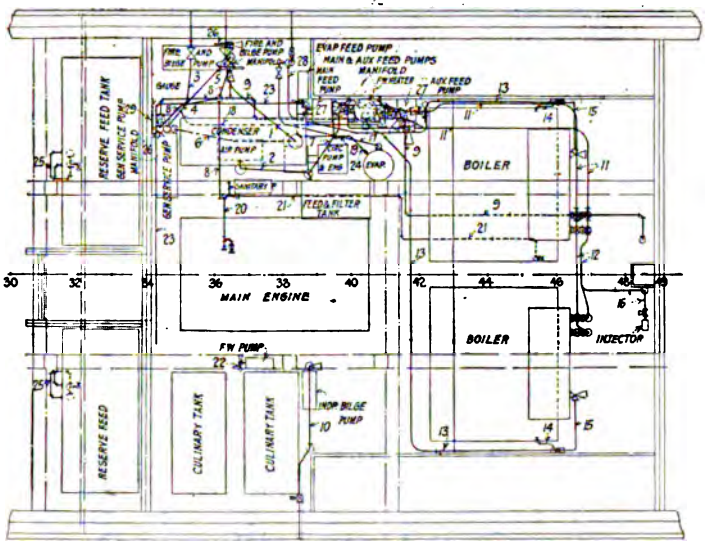
Boiler Rooms, 3500 Ton D. W. Steamer.



List of Pipes

Mark	Size, nom. in.	Service
1	10	Main circ. pump sea suction.
2	6	Bilge injection.
3	7	Gen. service & fire & bilge pumps sea suction.
4	7	Air pump suction from condenser.
5	4	Aux. feed pump suction from condenser.
6	3	Sanitary pump sea suction.
7	4	Main feed pump suction from feed & filter tank.
8	4	Aux. feed pump suction from feed & filter tank.
9	4	Gen. service & fire & bilge pumps suction from bilge.
10	3½	Gen. service & fire & bilge pumps suction from for'd bilge.
11	3½	Gen. service & fire & bilge pumps suction from aft bilge.
12	3	Feed pumps suction from reserve feed tank.
13	1¼	Make up feed suction from reserve feed tank.
14	1½	Injector suction from reserve feed tank.
15	4	Indp. bilge pump suction.
16	3	Bilge suction.
17	2	Evaporator feed pump sea suction.
18	2	Evaporator feed pump suction from condenser overb'd disch.
19	2	Fresh water pump from culinary tanks.

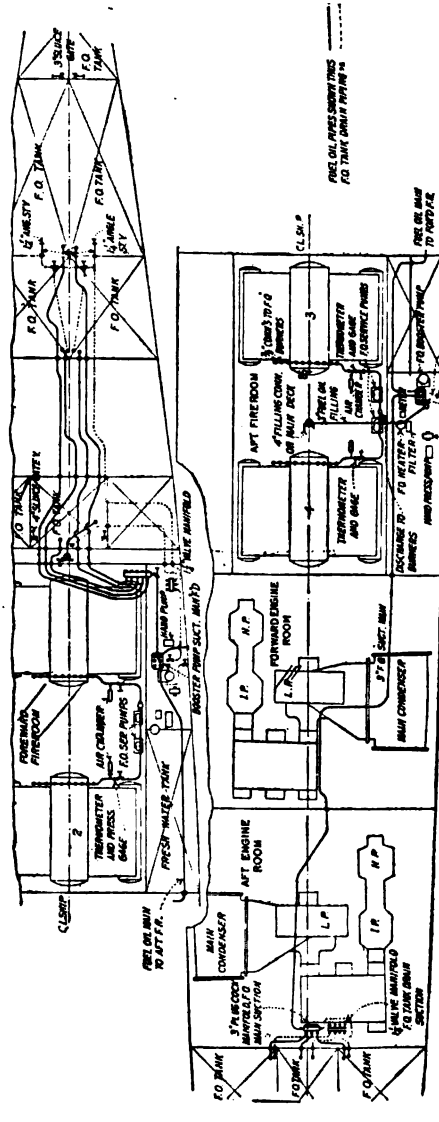
Fig. 3.—Suction Piping in Engine and Boiler Rooms, 3500 Ton, D. W. Steamer.



List of Pipes

Mark	Size, nom. in.	Service	Mark	Size, nom. in.	Service
1	10	Overboard discharge.	15	1½	Surface flow.
2	6	Air pump disch. to feed & filter tank.	16	1½	Injector disch.
3	5	Gen. service-fire & bilge pumps disch. overboard.	17	2	Evaporator blow overboard.
4	4	Gen. service pump disch. overboard.	18	1½	Feed pump disch. overboard.
5	4	Fire & bilge pump disch. overboard.	19	1½	Feed pump disch. to reserve feed tank.
6	4	Gen. service pump disch. to circ. water.	20	2½	Sanitary pump disch. to water service.
7	3	Feed pump disch. to feed & filter tank.	21	1¼	Sanitary pump disch. to ash wetting.
8	3½	Gen. service-fire & bilge pumps disch. to fire main.	22	1½	F. W. pump disch. to F. W. system.
9	3	Gen. service-fire & bilge pumps disch. to ash ejector.	23	1¼	Evaporator feed pump disch. to ice machine.
10	3	Indp. bilge pump disch. overboard.	24	1	Evaporator feed pump disch. to evaporator.
11	3	Feed pump disch. to boiler.	25	2½	Filling & air pipes.
12	2	Feed pump disch. to boiler.	26	2	Gen. service-fire & bilge pumps relief.
13	1½	Boiler blows overboard.	27	2	Main & aux. feed pumps relief.
14	1½	Bottom blow.	28	2	Boiler blows overboard.
			29	¼	Firemain gauge.

FIG. 4.—Discharge Piping in Engine and Boiler Rooms, 3500 Ton, D. W. Steamer.


FORE PEAK TK.
F.O. TANK
F.O. TANK
F.O. TANK

Pipes should be led in such a manner that no **stiffeners** on **bulkheads** or beams on decks will be cut. Pipes should also be clear of butts and laps, where practicable. Watertight, removable plates should be fitted on bulkheads, where necessary, to allow flange bolts to be properly set up.

Where pipes pass through watertight bulkheads the holes should be made watertight by stuffing boxes, flanges, or other approved means. In no case should any of the watertight structure of the vessel form a part of the piping.

Where pipes pass through wooden decks they should be fitted with a hollow casting and a stuffing box, so arranged that non-conducting material may be fitted between movable and fixed parts.

All **branches leading from the steam pipes** to crank engines should be from the top or side of the pipe and in no case from the bottom. When

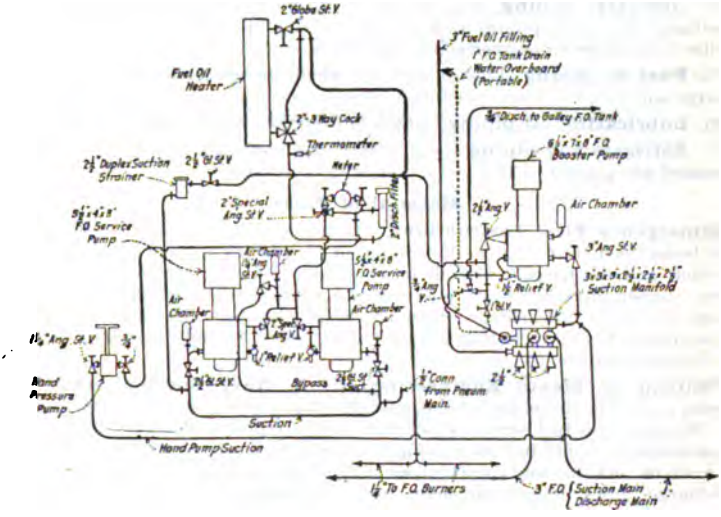


FIG. 5*a*.—Fuel Oil Piping Arrangement in After Fireroom.

the branch leads to a lower level, the stop valve on the branch should be placed as high as possible, so that with the engine standing idle there will be no opportunity for water to collect in the vertical pipe leading to it. Branches leading from steam pipes to direct-acting pumps may lead from the bottom of the pipes.

Each sea injection valve should have a steam connection leading from the auxiliary steam line to the injection pipe outside the injection valve, with a valve at each end, for cleaning out and thawing the strainer. The following sizes are recommended:

Size Sea Valve.....	6 in. and less.	6–9 in. inc.	9–12 in. inc.	Above 12 in.
Size Steam Conn.....	¾ in.	1 in.	1¼ in.	1½ in.

The piping systems on shipboard may be classified as:

1. **Steam piping,** Fig. 1, which includes the main and auxiliary steam lines; branch steam lines to all auxiliary machinery including generators and deck machinery; branch steam to smothering apparatus; steam to heating system, sea valves, whistle, injectors, galley and wash rooms, lubricating oil settling tank and fuel oil tank.

2. **Exhaust piping,** Fig. 2, which includes main exhaust to condensers; auxiliary exhaust lines; branch exhaust lines from all auxiliary machinery including generators and deck machinery; boiler escape pipes and machinery drains.

3. **Suction piping,** Fig. 3, which includes all salt and fresh water pump and injector suction lines from the sea, tanks and bilges.

4. **Discharge piping,** Fig. 4, which includes all salt and fresh water discharge lines from pumps and injectors; filling pipes and air vents to tanks; boiler blow-down lines overboard, and fire main.

5. **Fuel oil piping,** Figs. 5 and 5*a*, which includes all suction and discharge and other piping in the fuel oil system.

6. **Lubricating oil piping,** which is described on pp. 663 to 667.

7. **Refrigerating piping** for conveying refrigerating medium, and **compressed air piping** The last is not always fitted.

Material of Pipes

Emergency Fleet Instructions. Copper or brass pipe should be used for boiler feed, boiler blow-off, condenser circulating system and air pump suction and discharge. For all other service seamless steel pipe, wrought pipe, copper pipe and brass pipe may be used. All wrought pipe of 2 in. diam. should be lap-welded. Standard weight wrought pipe may be used for pressures up to and including 125 lbs.; extra heavy wrought pipe may be used for higher pressure.

Bureau of Steam Engineering, U. S. Navy Specifications. All steam pipes 2 in. diam. and above, seamless drawn steel; copper where directed by Bureau. Oil-fuel service pipes and branch pipes to oil-fuel burners, steel seamless drawn. Oil-fuel suction pipes, lap-welded steel. Steam pipes less than 2 in. diam., exhaust and escape pipes 10 in. diam. and less, all feed suction and discharge pipes, pneumatic main from air compressor, refrigerating plant water and brine pipes, copper seamless drawn. Main exhaust pipes, galvanized sheet steel. Other exhaust and escape pipes above 10 in. diam., distiller circulating pump suction and discharge and evaporator feed pipes, copper. Pipes to reserve feed tanks, suction from drainage, lap-welded wrought iron or steel tubing. All fresh water pipes of evaporating and distilling plant, galvanized wrought iron. Pipes for forced lubrication and water service pipes, copper or brass, polished where exposed. Evaporator drains to tanks, and brine and blow pipes, copper, iron pipe size. Cooling pipes in scuttle butts, copper, well tinned on outside. Cooling pipes in refrigerator room, wrought iron. All internal pipes in boilers, steel, 83 mils thick.

All wrought-iron, steel, and copper pipes intended to convey salt water will be protected on the inside by a coating composed of two parts of lead and one of tin.

Comparison of Materials. Rear Admiral C. W. Dyson, U. S. Navy, in *Practical Marine Engineering*, states that: "The **advantages of copper** are its great ductility, freedom from corrosion, and the readiness with which it may be used to make pieces of an irregular form, such as the elbows, junctions, etc., referred to above. Its disadvantages are, greater cost, low tensile

strength, the possibility of damage to the quality of the material in the process of pipe manufacture, and the possibility of a loss of ductility in service by repeated strains due to the expansions and contractions which result from changes in temperature."

"The metal close about a flanged joint seems especially liable to lose its strength in this way. This is probably due to the concentration of the strains due to expansions and contractions in the vicinity of a rigid connection such as a flanged joint, and to the development in this way of a line of weakness running around the pipe."

"The chief **advantages of wrought iron and steel pipes** are, less cost, greater tensile strength, less liability of the material to damage in quality in the processes of manufacture, and less liability to lose strength or ductility in service. Their disadvantages are, greater liability to corrosion, and greater weight of cast junctions and fittings than for copper."

Dimensions and Weights of Pipe and Tubing

Comparative transverse areas of pipes of various diameters is given in Table 1.

Table 1. Comparative Transverse Areas of Pipe

	⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	9	10
⅛	1.0																		
¼	1.8	1.0																	
⅜	3.3	1.8	1.0																
½	5.3	2.9	1.5	1.0															
¾	9.3	5.1	2.7	1.7	1.0														
1	15.0	8.3	4.5	2.8	1.6	1.0													
1¼	26.1	14.3	7.8	4.9	2.8	1.7	1.0												
1½	35.3	19.5	10.6	6.7	3.8	2.4	1.4	1.0											
2	58.6	32.2	17.5	11.0	6.3	3.9	2.2	1.6	1.0										
2½	83.5	45.9	24.4	15.7	9.0	5.5	3.2	2.3	1.4	1.0									
3	129.0	71.0	33.3	24.2	13.8	8.6	5.0	3.6	2.2	1.5	1.0								
3½	172.6	90.6	51.6	32.4	18.5	11.4	6.6	4.8	2.9	2.1	1.3	1.0							
4	222.1	122.3	66.3	41.8	23.9	14.8	8.5	6.2	3.8	2.7	1.7	1.3	1.0						
5	348.8	192.0	104.2	65.6	37.5	23.2	13.4	9.8	6.0	4.1	2.7	2.0	1.6	1.0					
6	504.1	277.5	150.7	94.8	54.1	33.5	19.3	14.2	8.6	6.0	3.9	2.9	2.3	1.4	1.0				
7	676.6	372.1	202.1	127.0	72.6	44.9	25.9	19.0	11.2	8.1	5.2	3.9	3.0	1.9	1.3	1.0			
8	873.2	480.7	261.1	164.9	93.8	58.0	33.4	24.5	14.9	10.5	6.8	5.2	3.9	2.5	1.8	1.3	1.0		
9	1094.7	602.6	327.2	205.8	117.6	72.7	41.9	30.8	18.7	13.1	8.5	6.3	4.9	3.1	2.1	1.6	1.3	1.0	
10	1375.8	757.3	411.3	258.6	147.8	92.0	52.7	38.7	23.5	16.5	10.7	7.9	6.2	3.9	2.8	2.0	1.6	1.3	1.0

Weight of steel pipe or tubes per linear foot is given by the formula

$$W = 10.680158\ (D - t)t,$$

where W = weight of steel tube in pounds per foot;
D = outside diameter of tube in inches;
t = thickness in inches.

This formula is based on weight of 1cu. in. of steel being 0.2833 lb. The dimensions and weights, respectively, of **standard, extra strong and double extra strong** wrought steel pipe are given in Tables 2, 3 and 4. The outside diameter of all classes of pipe of the same size is the same as the outside diameter of standard pipe, the extra thickness being on the inside. All weights in tables are for black pipe and couplings; galvanized pipe and couplings will be slightly heavier.

Pipe and tubing are known by their nominal inside diameters from ⅛ in. to 15 in.; above 15 in. by their outside diameter. When ordering pipe, designate weight or thickness desired, but not both. All material will be cut to length when so ordered, with extreme variation not exceeding ¼ in. When length is not designated pipe is furnished in random lengths. When ordering threaded pipe Briggs Standard gages are used, see p. 1271.

Table 2. Dimensions and Weights of Standard Wrought Pipe

Size, in.	Diameters		Nominal thickness, in.	Internal transverse area, sq. in.	Length of pipe in ft. per sq. ft. of		Nominal weight, lbs. per ft.		Number of threads per in. of screw
	External, in.	Approximate internal, in.			External surface	Internal surface	Plain ends	Threaded and coupled	
⅛	0.405	0.269	0.068	0.057	9.431	14.199	0.244	0.245	27
¼	0.540	0.364	0.088	0.104	7.073	10.493	0.424	0.425	18
⅜	0.675	0.493	0.091	0.191	5.658	7.747	0.567	0.568	18
½	0.840	0.622	0.109	0.304	4.547	6.141	0.850	0.852	14
¾	1.050	0.824	0.113	0.533	3.637	4.635	1.130	1.134	14
1	1.315	1.049	0.133	0.864	2.904	3.641	1.678	1.684	11½
1¼	1.660	1.380	0.140	1.495	2.301	2.767	2.272	2.281	11½
1½	1.900	1.610	0.145	2.036	2.010	2.372	2.717	2.731	11½
2	2.375	2.067	0.154	3.355	1.608	1.847	3.652	3.678	11½
2½	2.875	2.469	0.203	4.788	1.328	1.547	5.793	5.819	8
3	3.500	3.068	0.216	7.393	1.091	1.245	7.575	7.616	8
3½	4.000	3.548	0.226	9.886	0.954	1.076	9.109	9.202	8
4	4.500	4.026	0.237	12.730	0.848	0.948	10.790	10.889	8
4½	5.000	4.506	0.247	15.947	0.763	0.847	12.538	12.642	8
5	5.563	5.047	0.258	20.006	0.686	0.756	14.617	14.810	8
6	6.625	6.065	0.280	28.891	0.576	0.629	18.974	19.185	8
7	7.625	7.023	0.301	38.738	0.500	0.543	23.544	23.769	8
8	8.625	8.071	0.277	51.161	0.442	0.473	24.696	25.000	8
8	8.625	7.981	0.322	50.027	0.442	0.478	28.554	28.809	8
9	9.625	8.941	0.342	62.786	0.396	0.427	33.907	34.188	8
10	10.750	10.192	0.279	81.585	0.355	0.374	31.201	32.000	8
10	10.750	10.136	0.307	80.691	0.355	0.376	34.240	35.000	8
10	10.750	10.020	0.365	78.855	0.355	0.381	40.483	41.132	8
11	11.750	11.000	0.375	95.033	0.325	0.347	45.557	46.247	8
12	12.750	12.090	0.330	114.800	0.299	0.315	43.773	45.000	8
12	12.750	12.000	0.375	113.097	0.299	0.318	49.562	50.706	8

Permissible variation in weight ± 5 per cent. Weight per foot of pipe with threads and couplings is based on a length of 20 ft., including the coupling.

Table 3. Dimensions and Weights of Extra Strong Wrought Pipe

Diameter			Nominal thickness, in.	Internal transverse area, sq. in.	Length of pipe in ft. per square foot of		Nominal weight per foot plain ends, lbs.
Nominal internal, in.	External, in.	Approximate internal, in.			External surface	Internal surface	
⅛	0.405	0.215	0.095	0.036	9.431	17.766	0.314
¼	0.540	0.302	0.119	0.072	7.073	12.648	0.535
⅜	0.675	0.423	0.126	0.141	5.658	9.030	0.738
½	0.840	0.546	0.147	0.234	4.547	6.995	1.087
¾	1.050	0.742	0.154	0.433	3.637	5.147	1.473
1	1.315	0.957	0.179	0.719	2.904	3.991	2.171
1¼	1.660	1.278	0.191	1.283	2.301	2.988	2.996
1½	1.900	1.500	0.200	1.767	2.010	2.546	3.631
2	2.375	1.939	0.218	2.953	1.608	1.969	5.022
2½	2.875	2.323	0.276	4.238	1.328	1.644	7.661
3	3.500	2.900	0.300	6.605	1.091	1.317	10.252
3½	4.000	3.364	0.318	8.888	0.954	1.135	12.505
4	4.500	3.826	0.337	11.497	0.848	0.998	14.983
4½	5.000	4.290	0.355	14.455	0.763	0.890	17.611
5	5.563	4.813	0.375	18.194	0.686	0.793	20.778
6	6.625	5.761	0.432	26.067	0.576	0.663	28.573
7	7.625	6.625	0.500	34.472	0.500	0.576	38.048
8	8.625	7.625	0.500	45.663	0.442	0.500	43.388
9	9.625	8.625	0.500	58.426	0.396	0.442	48.728
10	10.750	9.750	0.500	74.662	0.355	0.391	54.735
11	11.750	10.750	0.500	90.763	0.325	0.355	60.075
12	12.750	11.750	0.500	108.434	0.299	0.325	65.415

See footnotes, Table 2.

Table 4. Dimensions and Weights of Double Extra Strong Wrought Pipe

Diameter					Length of pipe in ft. per square foot of		Nominal weight per foot plain ends, lbs.
Nominal internal, in.	External, in.	Approximate internal diameter, in.	Nominal thickness, in.	Internal transverse area, sq. in.	External surface	Internal surface	
½	0.840	0.252	0.294	0.050	4.547	15.157	1.714
¾	1.050	0.434	0.308	0.148	3.637	8.801	2.440
1	1.315	0.599	0.358	0.282	2.904	6.376	3.659
1¼	1.660	0.896	0.382	0.630	2.301	4.263	5.214
1½	1.900	1.100	0.400	0.950	2.010	3.472	6.408
2	2.375	1.503	0.436	1.774	1.608	2.541	9.029
2½	2.875	1.771	0.552	2.464	1.328	2.156	13.695
3	3.500	2.300	0.600	4.155	1.091	1.660	18.583
3½	4.000	2.728	0.636	5.845	0.954	1.400	22.850
4	4.500	3.152	0.674	7.803	0.848	1.211	27.541
4½	5.000	3.580	0.710	10.066	0.763	1.066	32.530
5	5.563	4.063	0.750	12.966	0.686	0.940	38.552
6	6.625	4.897	0.864	18.835	0.576	0.780	53.160
7	7.625	5.875	0.875	27.109	0.500	0.650	63.079
8	8.625	6.875	0.875	37.122	0.442	0.555	72.424

See footnotes, Table 2.

Table 5. Weight per Foot of Seamless Steel Tubing

Based on 1 cu. in. Steel = 0.2833 lb.
Compiled from tables issued by the National Tube Company, Pittsburg, Pa., in its "Book of Standard," 1917

Outside diameter, in.	Thickness in B.W. gage and fractions of an inch																
	22 0.028	20 0.035	18 0.049	1/16	5/32	1/8	5/32	3/16	7/32	1/4	5/16	3/8	1/2	5/8	3/4	7/8	1
1/8	0.141	0.174	0.236	0.292	0.407	0.501	0.57										
3/16	0.179	0.221	0.301	0.376	0.532	0.668	0.78	0.88									
1/4	0.216	0.267	0.367	0.459	0.657	0.834	0.99	1.13	1.24	1.33							
5/16	0.253	0.314	0.432	0.542	0.782	1.00	1.20	1.38	1.53	1.67							
1	0.291	0.361	0.498	0.626	0.907	1.17	1.41	1.63	1.83	2.00	2.30						
1 1/8	0.328	0.407	0.563	0.709	1.03	1.34	1.62	1.88	2.12	2.34	2.71	3.00					
1 1/4	0.365	0.454	0.629	0.793	1.16	1.50	1.83	2.13	2.41	2.67	3.13	3.50					
1 3/8	....	0.501	0.694	0.876	1.28	1.67	2.03	2.38	2.70	3.00	3.55	4.01					
1 1/2	....	0.548	0.759	0.960	1.41	1.84	2.24	2.63	2.99	3.34	3.96	4.51	5.34				
1 3/4	....	....	....	1.13	1.66	2.17	2.66	3.13	3.58	4.01	4.80	5.51	6.68				
2	....	....	....	1.29	1.91	2.50	3.08	3.63	4.16	4.67	5.63	6.51	8.01	9.18			
2 1/4	....	....	....	1.46	2.16	2.84	3.49	4.13	4.75	5.34	6.47	7.51	9.35	10.85			
2 1/2	....	....	....	1.63	2.41	3.17	3.91	4.63	5.33	6.01	7.30	8.51	10.68	12.52			
2 3/4	....	....	....	1.79	2.66	3.50	4.33	5.13	5.91	6.68	8.14	9.51	12.02	14.18			
3	....	....	....	....	2.91	3.84	4.75	5.63	6.50	7.34	8.97	10.51	13.35	15.85	18.02	19.86	21.36
3 1/4	....	....	....	....	3.16	4.17	5.16	6.13	7.08	8.01	9.80	11.51	14.69	17.52	20.03	22.19	24.03
3 1/2	....	....	....	....	3.41	4.51	5.58	6.63	7.67	8.68	10.64	12.52	16.02	19.19	22.03	24.53	26.70
3 3/4	....	....	....	....	....	4.84	6.00	7.13	8.25	9.35	11.47	13.52	17.36	20.86	24.03	26.87	29.37
4	....	....	....	....	....	5.17	6.41	7.64	8.83	10.01	12.31	14.52	18.69	22.53	26.03	29.20	32.04
4 1/4	....	....	....	....	....	....	6.83	8.14	9.42	10.68	13.14	15.52	20.03	24.20	28.04	31.54	34.71
4 1/2	....	....	....	....	....	....	7.25	8.64	10.00	11.35	13.98	16.52	21.36	25.87	30.04	33.88	37.38
4 3/4	....	....	....	....	....	....	7.67	9.14	10.59	12.02	14.81	17.52	22.70	27.53	32.04	36.21	40.05
5	....	....	....	....	....	....	8.08	9.64	11.17	12.68	15.64	18.52	24.03	29.20	34.04	38.55	42.72
5 1/4	....	....	....	....	....	....	8.50	10.14	11.75	13.35	16.48	19.52	25.37	30.87	36.05	40.88	45.39
5 1/2	....	....	....	....	....	....	8.92	10.64	12.34	14.02	17.31	20.53	26.70	32.54	38.05	43.22	48.06
5 3/4	....	....	....	....	....	....	9.34	11.14	12.92	14.69	18.15	21.53	28.04	34.21	40.05	45.56	50.73
6	....	....	....	....	....	....	9.75	11.64	13.51	15.35	18.98	22.53	29.37	35.88	42.05	47.89	53.40

Table 6. Weight per Foot of Seamless Brass Tubes

(Condensed from Manufacturers' Standard Tables, 1915)

Wall*	A.W.G.	2	4	6	8	10	12	16	20	24
	in.	0.2576	0.2043	0.1620	0.1285	0.1019	0.0808	0.0508	0.0320	0.0201
	Diam.†									
	1/8	....	....	....	....	....	.....	0.044	0.034	0.024
	5/32	....	....	....	....	....	.....	0.080	0.058	0.039
	3/16	....	....	....	....	0.175	0.158	0.117	0.081	0.053
	7/32	....	....	....	....	0.248	0.217	0.154	0.104	0.068
	1/4	....	....	....	0.376	0.322	0.275	0.191	0.127	0.083
	5/16	....	....	0.634	0.562	0.469	0.392	0.264	0.173	0.112
	3/8	1.10	0.994	0.868	0.748	0.617	0.509	0.338	0.219	0.141
	7/16	1.47	1.29	1.10	0.934	0.764	0.626	0.411	0.266	0.170
	1/2	1.84	1.59	1.34	1.12	0.911	0.743	0.485	0.312	0.199
	1	2.21	1.88	1.57	1.31	1.06	0.859	0.558	0.358	0.228
	1 1/16	2.59	2.18	1.81	1.49	1.21	0.976	0.632	0.404	0.257
	1 1/8	2.96	2.47	2.04	1.68	1.35	1.09	0.705	0.450	0.286
	1 3/16	3.33	2.77	2.27	1.86	1.50	1.21	0.779	0.497	0.315
	1 1/4	3.70	3.06	2.51	2.05	1.65	1.33	0.852	0.543	0.344
	1 3/8	4.45	3.65	2.98	2.42	1.94	1.56	0.999	0.635	0.402
	2	5.19	4.24	3.45	2.79	2.24	1.79	1.15	0.728	0.460
	2 1/4	5.94	4.84	3.91	3.16	2.53	2.03	1.29	0.820	0.519
	2 1/2	6.68	5.43	4.38	3.54	2.83	2.26	1.44	0.913	0.577
	2 3/4	7.43	6.02	4.85	3.91	3.12	2.50	1.59	1.01	0.635
	3	8.17	6.61	5.32	4.28	3.42	2.73	1.73	1.10	0.693
	3 1/4	8.92	7.20	5.79	4.65	3.71	2.96	1.88	1.19	0.751
	3 1/2	9.66	7.79	6.26	5.02	4.01	3.20	2.03	1.28	0.809
	3 3/4	10.4	8.38	6.73	5.39	4.30	3.43	2.18	1.37	0.867
	4	11.2	8.97	7.19	5.77	4.60	3.66	2.32	1.47	0.926
	4 1/4	11.9	9.56	7.66	6.14	4.89	3.90	2.47	1.56	0.984
	4 1/2	12.6	10.2	8.13	6.51	5.19	4.13	2.62	1.65	1.04
	4 3/4	13.4	10.7	8.60	6.88	5.48	4.37	2.76	1.74	1.10
	5	14.1	11.3	9.07	7.25	5.78	4.60	2.91	1.84	1.16
	5 1/4	14.9	11.9	9.54	7.62	6.07	4.83	3.06	1.93	
	5 1/2	15.6	12.5	10.0	8.00	6.36	5.07	3.20	2.02	
	5 3/4	16.4	13.1	10.5	8.37	6.66	5.30	3.35	2.11	
	6	17.1	13.7	10.9	8.74	6.95	5.53	3.50	2.21	
	6 1/4	17.9	14.3	11.4	9.11	7.25	5.77	3.65		
	6 1/2	18.6	14.9	11.9	9.48	7.54	6.00	3.79		
	6 3/4	19.4	15.5	12.3	9.85	7.84	6.24	3.94		
	7	20.1	16.1	12.8	10.2	8.13	6.47	4.09		
	7 1/4	20.8	16.7	13.3	10.6	8.43	6.70	4.23		
	7 1/2	21.6	17.2	13.8	11.0	8.72	6.94	4.38		
	7 3/4	22.3	17.8	14.2	11.3	9.02	7.17	4.53		
	8	23.1	18.4	14.7	11.7	9.31	7.40	4.67		
	8 1/4	23.8	19.0	15.2	12.1	9.61	7.64	4.82		
	8 1/2	24.6	19.6	15.6	12.5	9.90	7.87	4.97		
	8 3/4	25.3	20.2	16.1	12.8	10.2	8.11	5.12		
	9	26.1	20.8	16.6	13.2	10.5	8.34	5.26		
	9 1/4	26.8	21.4	17.0	13.6	10.8	8.57	5.41		
	9 1/2	27.6	22.0	17.5	13.9	11.1	8.81	5.56		
	9 3/4	28.3	22.6	18.0	14.3	11.4	9.04	5.70		
	10	29.0	23.2	18.4	14.7	11.7	9.27	5.85		

* Thickness in inches. † Outside diameter, inches.

Seamless brass tubes are made from 1/8 in. to 1 in. outside diameter, varying by 1/16 in., nd from 1 1/8 in. to 10 in. outside diameter, varying by 1/8 in., and in all gages from No. 2 o No. 24 A.W.G. within the limits of the above table. To determine the weight per foot f a tube of a given inside diameter, add to the weights given above the weights given below, nder the corresponding gage numbers.

A.W.G.	2	4	6	8	10	12	16	20	24
Lb. per ft.	1.54	0.966	0.607	0.382	0.240	0.151	0.060	0.024	0.009

For copper tubing add 5 per cent to the weights given above.

Weight of pipes of materials other than steel may be obtained by using the factor in Table 7, where W = weight of steel pipe as obtained by formula on page 1255 or from Table 2.

Table 7. Weights of Pipe of Materials Other Than Steel

Weight of Wrought Iron Pipe... = W − 2 per cent.
Weight of Cast Iron Pipe..... = W − 8 per cent.
Weight of Wrought Copper Pipe = W + 13 per cent.
Weight of Wrought Brass Pipe = W + 7 per cent.
Weight of Wrought Nickel Pipe = W + 10½ per cent.
Weight of Lead Pipe..... = W + 45 per cent.
Weight of Tin Pipe..... = W − 6 per cent.
Weight of Cast Aluminum Pipe = W − 67½ per cent.

Table 8. Seamless Brass and Copper Tube, Iron Pipe Sizes

Nominal size, in.	Weight per foot, lbs.		Nominal size, in.	Weight per foot, lbs.	
	Brass	Copper		Brass	Copper
⅛	0.246	0.259	3	8.314	8.741
¼	0.437	0.459	3½	10.85	11.41
⅜	0.612	0.644	4	12.29	12.93
½	0.911	0.958	4½	13.74	14.44
¾	1.235	1.298	5	15.40	16.19
1	1.740	1.829	6	18.44	19.39
1¼	2.557	2.698	7	23.92	25.15
1½	3.037	3.193	8	30.05	31.60
2	4.017	4.224	9	36.94	38.84
2½	5.830	6.130	10	43.91	46.17

Table 9. Tin-lined and Lead-lined Iron Pipe

(United Lead Co., New York, 1915)

Size, in.	Wt. per ft., lbs.		Size, in.	Wt. per ft, lbs.		Size, in.	Wt. per ft., lbs.		Size, in.	Wt. per ft., lbs. lead lined
	Lead lined	Tin lined		Lead lined	Tin lined		Lead lined	Tin lined		
½	1.37	1.00	2	6.12	5.25	4½	18.00	16.00	9	66
¾	1.62	1.37	2½	8.50	7.50	5	21.50	26.10	10	75
1	2.50	2.25	3	11.50	10.16	6	29.75	19.16	12	88
1¼	3.50	3.00	3½	14.50	12.80	7	36.00			
1½	4.37	3.75	4	15.66	14.16	8	47.00			

Weight of Seamless Steel Tubing is given in Table 5.
Weight of Seamless Brass Tubing is given in Tables 6 and 8.
Weight of Copper Tubing is given in Table 8.
Weight of Tin-lined and Lead-lined Iron Pipe is given in Table 9.
Weight of Lead Pipe is given in Table 10.

Table 10. Lead and Tin-lined Lead Pipe

United Lead Company, New York, 1915

Inside diam., in.	Letter	Weight per ft. lbs.	Thickness	Inside diam., in.	Letter	Weight per ft. lbs.	Thickness	Inside diam., in.	Letter	Weight per ft. lbs.	Thickness
3/16	D	0.63	0.09	5/8	AA	2.75	0.21	1 1/16	AA	5.75	0.25
3/16	C	0.75	0.10	5/8	AAA	3.50	0.26	1 1/16	AAA	6.75	0.28
3/16	B	1.00	0.13	3/4	E	1.00	0.08	1 1/8	E	3.00	0.12
3/16	A	1.25	0.15	3/4	D	1.25	0.09	1 1/8	D	3.50	0.14
3/16	AA	1.50	0.17	3/4	C	1.75	0.13	1 1/8	C	4.25	0.16
3/16	AAA	1.75	0.20	3/4	Special	2.00	0.14	1 1/8	B	5.00	0.19
1/4	.....	0.81	0.10	3/4	B	2.25	0.16	1 1/8	A	6.50	0.24
1/4	.....	1.00	0.12	3/4	A	3.00	0.20	1 1/8	AA	7.50	0.27
1/4	E	0.56	0.06	3/4	AA	3.50	0.23	1 1/8	AAA	8.50	0.30
1/4	D	0.75	0.08	3/4	AAA	4.75	0.29	1 3/16	D	4.00	0.14
1/4	C	1.00	0.11	1	E	1.50	0.09	1 3/16	C	5.00	0.17
1/4	B	1.25	0.13	1	D	2.00	0.12	1 3/16	B	6.00	0.20
1/4	Special	1.50	0.15	1	C	2.50	0.14	1 3/16	A	7.00	0.23
1/4	A	1.75	0.17	1	B	3.25	0.18	1 3/16	AA	8.50	0.27
1/4	AA	2.00	0.19	1	A	4.00	0.21	1 3/16	AAA	10.00	0.31
1/4	Special	2.50	0.22	1	AA	4.75	0.25	2	D	4.75	0.14
1/4	AAA	3.00	0.26	1	AAA	6.00	0.30	2	C	6.00	0.18
5/16	E	0.75	0.07	1 1/4	E	2.00	0.10	2	B	7.00	0.21
5/16	D	1.00	0.09	1 1/4	D	2.50	0.12	2	A	8.00	0.23
5/16	C	1.50	0.13	1 1/4	C	3.00	0.14	2	AA	9.00	0.26
5/16	B	2.00	0.16	1 1/4	B	3.75	0.17	2	AAA	11.75	0.33
5/16	A	2.50	0.20	1 1/4	A	4.75	0.21				

Weight of lead is taken as 0.4106 lb. per cu. in. The safe working strength of lead is about 1/4 the elastic limit, or 225 lbs. per sq. in.

Thickness of Pipes

Emergency Fleet Corporation formulæ for thickness of pipes, where T = thickness in in., P = working pressure in lbs. per sq. in. and D = inside diam. of pipe in in. are as follows:

For copper pipe $T = \frac{P \times D}{6000} + 0.625$.

For seamless steel pipe $T = \frac{P \times D}{10{,}000} + 0.125$.

For wrought pipe thicknesses, see Tables 2, 3, and 4.

Bureau of Steam Engineering, U. S. Navy, formulæ for thickness of pipes, where T = thickness of pipe, in in., D = inside diam. of pipe in in., and P = pressure above atmosphere in lbs. per sq. in. as taken from Table 11, follows:

For straight copper pipe $T = \frac{P \times D}{8000} + 1/16$ in.

For straight steel pipe $T = \frac{P \times D}{10{,}000} + 1/16$ in.

Table 11

Pipes	P =	Pipes	P =
Steam, bleeder, blowpipes from boilers	Boiler press.	Exhaust pipes above 4½ in. diam.	50.0
Feed discharge pipes	1.5 boiler press.	Pneumatic main from air compressor	200.0
Fire-main connections	150.0	Vapor and escape pipes	30.0
Feed suction pipes	100.0	Oil-fuel service pipes	300.0
Other water pipes without pressure	50.0	Oil-fuel suction pipes	50.0
Exhaust pipes 4½ in. diam. and less	20.0	Forced-lubrication system pipes	40.0

Strength of Pipes

Barlow's Formula is the one best suited for all ordinary calculations pertaining to the bursting strength of commercial tubes, pipes and cylinders. The theoretical error on the side of safety resulting from its use will generally not exceed the actual combined error on the side of danger when using other formulæ due to the ordinary range of variation in the thickness of wall, strength of the material, etc., when applied to the ordinary commercial product.

This is true, at least up to the yield point of the material, for any ratio of thickness of wall to outside diameter less than three-tenths. In this respect Barlow's formula is very superior to the common approximate formula which gives errors that are absurdly large on the side of danger for very thick walls.

For certain classes of seamless tubes and cylinders and for critical examinations of welded pipe, where the least thickness of wall, yield point of material, etc., are known with accuracy and close results are desired, Clavarino's formula and Birnie's equations may be used. They do not find application in marine engineering.

For all ordinary calculations pertaining to the bursting strength of commercial tubes, pipes and cylinders use Barlow's Formula, which is

$$\frac{p}{t} = \frac{2t}{D};\ p = \frac{2ft}{D};\ t = \frac{Dp}{2f};\ f = \frac{Dp}{2t}$$

where D = outside diameter, inches;
t = average thickness of wall, inches;
p = internal fluid pressure, pounds per square inch;
f = working or safe fiber stress, pounds per square inch.

When n = safety factor as based on ultimate strength then
$f = 40{,}000/n$ for butt-welded steel pipe;
$= 50{,}000/n$ for lap-welded steel pipe;
$= 60{,}000/n$ for seamless steel tubes;
$= 28{,}000/n$ for wrought iron pipe.

These average values of f are based upon the values of bursting pressures of commercial tubes and pipes shown in Table 12. They are intended for substitution in Barlow's Formula in case more exact data for the working fiber stress are not at hand.

Table 12. Bursting Tests of Commercial Tubes and Pipes

Tests made by *National Tube Company*

Size	Number of pieces burst	Nominal external diam. in.	Average thickness of walls, in.	Bursting pressures, lbs. per sq. in. Minimum	Maximum	Average	Head condition	See note below	Average fiber stress by Barlow's formula	Class of material
Steel—butt-welded	10	0.405	0.066	11,840	17,320	14,266	*C*	1	44,011	Standard pipe
	10	0.540	0.085	8,830	14,680	12,206	*C*	1	38,645	Standard pipe
	10	0.675	0.088	15,850	13,030	10,330	*C*	1	39,272	Standard pipe
	10	0.840	0.101	11,380	16,310	14,038	*C*	0	58,163	Standard pipe
	10	1.050	0.109	7,150	9,150	8,020	*C*	0	38,057	Standard pipe
1	10	1.315	0.131	4,500	8,800	6,990	*C*	0	35,085	Standard pipe
1¼	10	1.660	0.139	4,400	7,300	5,808	*C*	0	34,603	Standard pipe
1¼	15	1.660	0.140	5,500	11,900	7,700	*C*	1	45,215	Redrawn
1½	10	1.900	0.143	3,000	6,100	4,960	*C*	0	33,031	Standard pipe
2	11	2.375	0.149	3,830	6,060	4,951	*C*	0	40,485	Standard pipe
2½	10	2.875	0.198	4,310	5,740	5,134	*C*	0	37,351	Standard pipe
3	10	3.500	0.204	4,650	6,370	5,398	*C*	0	46,234	Standard pipe
1¼	10	1.660	0.180	7,910	14,280	10,514	*C*	0	48,922	Extra strong
2	10	2.375	0.213	7,250	8,940	8,238	*C*	0	45,935	Extra strong
2	10	2.375	0.220	6,160	8,920	7,661	*C*	0	41,347	Extra strong
2	10	2.375	0.445	8,500	18,314	14,992	*C*	0	40,023	XX strong
						General average			41,686	
Steel—lap-welded	10	2.375	0.155	4,890	7,940	6,645	*C*	1	50,962	Standard pipe
	10	2.375	0.182	4,860	10,060	7,361	*C*	0	47,889	Standard pipe
	10	3.500	0.210	3,830	8,200	6,368	*C*	7	53,560	Standard pipe
	10	4.500	0.232	4,810	5,680	5,249	*C*	1	51,462	Standard pipe
	10	5.563	0.258	3,410	5,260	4,538	*C*	1	48,882	Standard pipe
	5	6.625	0.275	2,450	5,210	4,088	*C*	0	49,286	Standard pipe
	5	6.625	0.275	3,170	4,760	3,666	*B*	0	44,106	Standard pipe
	5	10.750	0.349	3,560	4,730	4,290	*C*	1	66,080	Standard pipe
	5	10.750	0.347	2,770	3,940	3,396	*B*	2	52,692	Standard pipe
	10	2.375	0.218	2,500	9,870	7,909	*C*	0	43,254	Extra strong
	10	2.000	0.108	5,100	6,560	6,062	*C*	7	55,607	Boiler tubes
	10	3.000	0.112	3,220	4,860	3,967	*C*	1	52,957	Boiler tubes
	5	4.000	0.135	3,640	4,070	3,840	*C*	2	56,978	Boiler tubes
	5	4.000	0.136	3,720	4,040	3,914	*B*	1	57,440	Boiler tubes
						General average			52,225	
Steel—seamless	10	2.000	0.098	5,420	6,590	6,052	*C*	10	61,530	Boiler tubes
	10	3.000	0.112	3,940	4,730	4,272	*C*	10	57,075	Boiler tubes
	6	4.000	0.134	4,160	4,440	4,318	*C*	6	64,450	Boiler tubes
	4	4.000	0.134	4,250	4,440	4,328	*B*	4	64,488	Boiler tubes
						General average			61,886	
Iron—butt-welded	10	1.660	0.136	2,880	6,290	5,283	*C*	3	32,126	Standard pipe
	10	1.660	0.136	3,640	5,680	4,891	*C*	1	29,817	Standard pipe
	10	2.375	0.156	2,930	4,250	3,687	*C*	2	28,051	Standard pipe
	10	1.660	0.188	2,770	7,330	5,895	*C*	1	26,678	Extra strong
						General average			29,168	
Iron—lap-welded	10	2.375	0.152	2,400	3,940	3,213	*C*	1	25,122	Standard pipe
	10	2.375	0.207	5,530	7,120	6,349	*C*	8	36,461	Extra strong
						General average			30,792	

The column marked "See note below" gives the number burst by failure of material not at weld.

C—Clavarino conditions.

B—Birnie conditions.

The strength of pipe welds is given in Table 13, which is compiled from tests conducted by the *National Tube Co.* Of the steel tubes and pipes used in the preceding tables, 95 per cent were made by that Company, while 86

per cent of the wrought iron pipe tested was obtained by purchase in the open market.

The average ultimate tensile strength of pipe steel is 57,000 lbs. per sq. in., whether taken in the direction of rolling or transversely thereto, while that of the seamless steel tested is 60,000 lbs. per sq. in. No tensile tests were made of the material of the wrought iron pipes.

An examination of these tables will lead to the conclusion that the relative strengths of steel pipes and tubes, when using Barlow's Formula and basing the calculations on average diameter, thickness of wall and ultimate tensile strength of material, are as follows: For butt-welded steel pipe, 73 per cent; for lap-welded steel pipe, 92 per cent; and for seamless steel tubes, approximately 100 per cent. In steel pipe, then, the strength of the butt-weld is about 80 per cent of that of the lap-weld.

The relative strengths of wrought iron and steel pipe, from Table 12, are as follows: Butt-welded wrought-iron pipe is 70 per cent as strong as similar butt-welded steel pipe; and lap-welded wrought iron pipe is 60 per cent as strong as similar lap-welded steel pipe.

Table 13. Strength of Weld of Commercial Tubes and Pipes

(*National Tube Co.*)

Size	Number burst in weld*	Average fiber stress by Barlow's formula	Class of material
		Steel—butt-welded	
⅛	9	43,938	Standard pipe
¼	9	37,777	Standard pipe
⅜	9	38,954	Standard pipe
½	10	58,163	Standard pipe
¾	10	38,657	Standard pipe
1	10	35,085	Standard pipe
1¼	10	34,603	Standard pipe
1¼	14	45,643	Redrawn
1½	10	33,031	Standard pipe
2	11	40,485	Standard pipe
2½	10	37,351	Standard pipe
3	10	46,234	Standard pipe
1¼	10	48,922	Extra strong
2	10	45,935	Extra strong
2	10	41,347	Extra strong
2	10	40,023	XX strong
General average		41,634	
		Steel—lap-welded	
2	9	50,052	Standard pipe.
2	10	47,889	Standard pipe
3	3	54,510	Standard pipe
4	9	51,019	Standard pipe
5	9	48,852	Standard pipe
6	10	47,826	Standard pipe
10	7	59,537	Standard pipe
2	10	43,254	Extra strong
2	3	56,933	Boiler tubes
3	9	51,980	Boiler tubes
4	7	57,521	Boiler tubes
General average		51,688	

* These only are included in averages.

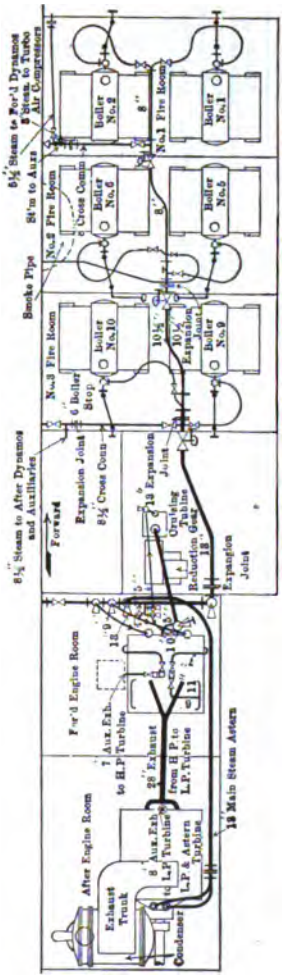


FIG. 7.—Installation Illustrating Practical Pipe Bends.

When applying these formulæ and tables to practice it should be remembered that a suitable factor of safety should be applied. The selection of a proper safety factor in any particular case should be left to the judgment of one who is quite familiar with the conditions under which the tube is to be used.

Ordinarily a safety factor of five is sufficient when the stresses due to actions other than a constant fluid pressure are more or less trivial. In case there are repeated fluctuations of the fluid pressure, vibration shock, internal strain due to unequal heating, etc., then a larger safety factor of from six to twelve or more should be used, depending upon the severity of these actions.

(3) The apparent fibre stress under which the different tubes failed varied from about 7000 lbs. for the relatively thinnest to 35,000 lbs. per sq. in. for the relatively thickest walls. Since the average yield point of the material was 37,000 and the tensile strength 58,000 lbs. per sq. in., it would appear that the strength of a tube subjected to a fluid collapsing pressure is not dependent alone upon either the elastic limit or the ultimate strength of the material constituting it.

The Board of Supervising Inspectors of the Steam Boat Inspection Service require that working pressures and corresponding minimum thicknesses of wall for long, plain, lap-welded and seamless steel flues, 7 to 18 inches diameter, subjected to external pressure only, shall be determined by Table 14.

Table 14

Outside diameter of flue, inches	Working pressure in pounds per square inch						
	100	120	140	160	180	200	220
	Thickness of flue in inches. Safety factor, 5						
7	0.152	0.160	0.168	0.177	0.185	0.193	0.201
8	0.174	0.183	0.193	0.202	0.211	0.220	0.229
9	0.196	0.206	0.217	0.227	0.237	0.248	0.258
10	0.218	0.229	0.241	0.252	0.264	0.275	0.287
11	0.239	0.252	0.265	0.277	0.290	0.303	0.316
12	0.261	0.275	0.289	0.303	0.317	0.330	0.344
13	0.283	0.298	0.313	0.328	0.343	0.358	0.373
14	0.301	0.320	0.337	0.353	0.369	0.385	0.402
15	0.323	0.343	0.361	0.378	0.396	0.413	0.430
16	0.344	0.366	0.385	0.404	0.422	0.440	0.459
17	0.366	0.389	0.409	0.429	0.448	0.468	0.488
18	0.387	0.412	0.433	0.454	0.475	0.496	0.516

Thicknesses in this table were calculated by formula (*B*), p. 1265.

Pipe Bends and Expansion of Pipes

Expansion of Pipes. The linear expansion or contraction of pipe, due to differences of temperature of the fluid carried and the surrounding air, must be cared for by suitable expansion joints or bends. Table 15 gives the increase in length of pipes 100 ft. long at various temperature differences. The expansion for any length of pipe may be found by taking the difference in increased length at the minimum and maximum temperatures, dividing by 100 and multiplying by the length of the line under consideration.

Table 15. Expansion of Pipe

Increase in Length—Inch per 100 Feet

(Crane Co.)

Temperature deg. F.	Steel	Wrought iron	Cast iron	Brass and copper
0	0	0	0	0
20	0.15	0.15	0.10	0.25
40	0.30	0.30	0.25	0.45
60	0.45	0.45	0.40	0.65
80	0.60	0.60	0.55	0.90
100	0.75	0.80	0.70	1.15
20	0.90	0.95	0.85	1.40
40	1.10	1.15	1.00	1.65
60	1.25	1.35	1.15	1.90
80	1.45	1.50	1.30	2.15
200	1.60	1.65	1.50	2.40
20	1.80	1.85	1.65	2.65
40	2.00	2.05	1.80	2.90
60	2.15	2.20	1.95	3.15
80	2.35	2.40	2.15	3.45
300	2.50	2.60	2.35	3.75
20	2.70	2.80	2.50	4.05
40	2.90	3.05	2.70	4.35
60	3.05	3.25	2.90	4.65
80	3.25	3.45	3.10	4.95
400	3.45	3.65	3.30	5.25
20	3.70	3.90	3.50	5.60
40	3.95	4.20	3.75	5.95
60	4.20	4.45	4.00	6.30
80	4.45	4.70	4.25	6.65
500	4.70	4.90	4.45	7.05
20	4.95	5.15	4.70	7.45
40	5.20	5.40	4.95	7.85
60	5.45	5.70	5.20	8.25
80	5.70	6.00	5.45	8.65
600	6.00	6.25	5.70	9.05
20	6.30	6.55	5.95	9.50
40	6.55	6.85	6.25	9.95
60	6.90	7.20	6.55	10.40
80	7.20	7.50	6.85	10.95
700	7.50	7.85	7.15	11.40
20	7.80	8.20	7.45	11.90
40	8.20	8.55	7.80	12.40
60	8.55	8.90	8.15	12.95
80	8.95	9.30	8.50	13.50
800	9.30	9.75	8.90	14.10

For Expansion Joints, see p. 1271.

Pipe Bends may be used to provide flexibility in steam piping and to **compensate for expansion and contraction.** The amount of expansion or contraction compensated by various types of bends of different radii for the common sizes of pipes can be obtained from Fig. 6, which was published in *The Valve World*, Oct., 1915. This manner of compensating for expansion is finding increasing use in Shore and Naval Practice, see Fig 7 showing the steam piping of the U. S. S. Nevada. Pipe bends are also used to **reduce the number of joints and fittings** in piping; they also will **reduce friction of flow** in pipes by providing long sweeping bends instead of the shorter turns provided by fittings. Bends are also used to avoid obstructions such as ships, beams and columns, machinery foundations, etc.

The following is quoted from *The Valve World*, Oct., 1915:

"In our tests we compared the merits of pipe bends to equivalent structures

made from straight pipe using regular extra heavy fittings, as it is not always convenient to install pipe bends and the user may have to resort to the pipe and fitting construction.

Our tests showed that bends are superior to the ordinary pipe and fitting structure, because the bends are more flexible and do not throw all the load on the joints. The built-up structures invariably leaked at the flanged joints, and in many instances this leakage occurred before the pipe or the fitting had been stressed near the maximum allowable fibre stress."

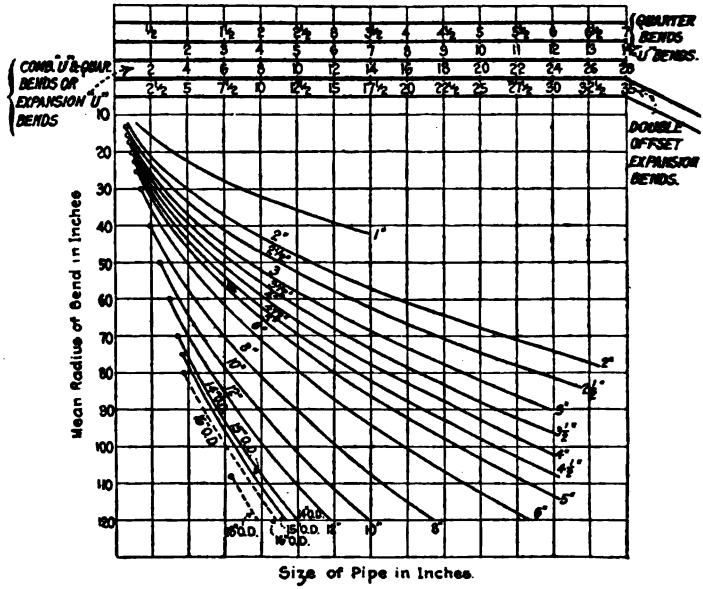

QUARTER BENDS
U BENDS.
COMB. U & QUAR. BENDS OR EXPANSION U BENDS
DOUBLE OFFSET EXPANSION BENDS.
Mean Radius of Bend in Inches
Size of Pipe in Inches.

FIG. 6.—Expansion Compensated by Different Pipe Bends; for Dimensions of Bends see Table 16, p. 1270.

The U. S. Navy Specifications require that no **bend in copper pipes** will be allowed of which the radius is less than one and one-half times the bore of the pipe. **Bends in steel pipes** will be as approved. The thinnest part of a bent pipe must be at least equal to the thickness required by the formulæ, p. 1261.

The Crane Co. for many years has standardised the dimensions of Pipe Bends. Table 16 illustrates the **minimum advisable radius** for pipe bends. For expansion work, they suggest the use of the largest possible radius which can be used, depending upon the installation.

Expansion or Slip Joint. When expansion and contraction in piping is not compensated by the method described under "pipe bends" expansion

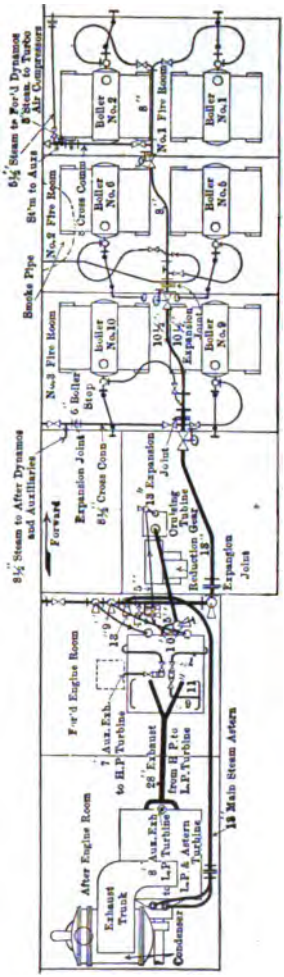
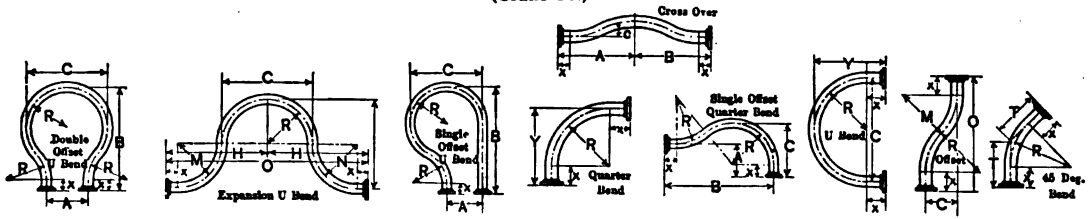


FIG. 7.—Installation Illustrating Practical Pipe Bends.

Table 16. Pipe Bends Made From Lap Welded Steel Pipe

(Crane Co.)



Size of pipe, in.	R-M-N advisable radius of bends, in.	T Centre to end or face of flanges, ft. in.	X Length of tangents or straight pipe on each bend, in.	Y Centre of bends to face of flanges or ends of pipe, ft. in.	Lineal feet of pipe in each quarter bend, ft. in.	Lineal feet of pipe in each U bend, ft. in.	Lineal feet of pipe in each 45° bend, ft. in.	Minimum radius to which bends can be made from extra strong pipe only, in.
2½	12½	0—9¾₁₆	4	1—4½	2—3¾	3—11¼	1—5⅞	7
3	15	0—10¼	4	1—7	2—7⅜	4—7⅛	1—7⅜	8
3½	17½	1—0¼	5	1—10½	3—1½	5—5	1—11¾	10
4	20	1—1¼	5	2—1	3—5½	6—1	2—1¾	12
4½	22½	1—3¹¹₁₆	6	2—4½	3—11½	6—10¾	2—5⅜	14
5	25	1—4⅜	6	2—7	4—3¼	7—6⅝	2—7½	15
6	30	1—7¹¹₁₆	7	3—1	5—1½	9—0¼	3—1½	20
7	35	1—10½	8	3—7	5—11	10—6	3—7⅜	24
8	40	2—1¹⁵₁₆	9	4—1	6—9	11—11¾	4—1½	28
9	45	2—5⅝	11	4—8	7—8¾	13—7⅜	4—9¼	35
10	50	2—8¾	12	5—2	8—6½	15—1⅛	5—3¼	40
12	60	3—2⅞	14	6—2	10—0¼	18—0¼	6—3¼	50
14	70	3—9	16	7—2	11—10	21—0	7—3	65
15	75	3—11¹¹₁₆	16	7—7	12—6		7—7	70
16	80	4—3⅛	18	8—2	13—5¾		8—2¾	78
18	108	5—2¾	18	10—6	17—1¾		10—0⅞	88
20	120	5—7¾	18	11—6	18—8½		10—10¼	104
22	132	6—0¾	18	12—6	20—3		11—7¾	132
24	144	6—5¾	18	13—6	21—10		12—0⅛	144

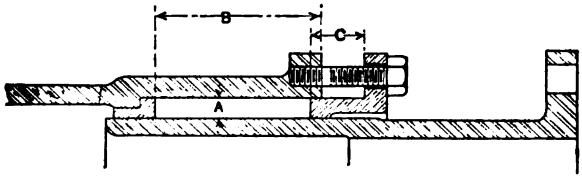
joints must be fitted for this purpose. In this connection *Mr. W. T. Dimm* of the *Newport News Shipbuilding Co.* states:

"For all Naval vessels we use the Bureau of Steam Engineering standard dimensions for H.P. pipe expansion joints, as given in Table 17. For low pressure joints we reduce somewhat the depth of the packing space. For merchant vessels we use practically the same dimensions as for Government work for both high pressure and low pressure."

In **Naval practice** expansion or slip joint fittings are made of class B cast steel or composition G and the gland of the latter material. Specifications require that steam, exhaust, and feed piping not fitted with expansion or slip joints will be run as approved, with ample bends, to provide for expansion. Such pipes will be so fitted as to put them under stress when cold by allowing for every 10 feet of length when bolting up a clearance between flanges of $\frac{3}{16}$ inch for steam pipes and $\frac{3}{32}$ inch for exhaust and feed pipes.

Table 17. Bureau Steam Engineering Standard Expansion Joint

Packing Space for Expansion Joints



Internal diameter of pipe	Packing space		Depth of gland	Internal diameter of pipe	Packing space		Depth of gland
	Width	Depth			Width	Depth	
	A in in.	B in in.	C in in.		A in in.	B in in.	C in in.
3	½	4	1½	11	⅞	7	2¾
3½	½	4	1½	11½	⅞	7	2¾
4	½	4	1½	12	⅞	7	2¾
4½	½	4	1½	12½	⅞	7	2¾
5	⅝	5	1¾	13	⅞	7	2¾
5½	⅝	5	1¾	13½	⅞	7	2¾
6	⅝	5	1¾	14	⅞	7	2¾
6½	⅝	5	1¾	14½	1	8	2¾
7	⅝	5	1¾	15	1	8	2¾
7½	⅝	5	1¾	15½	1	8	2¾
8	¾	6	2	16	1	8	2¾
8½	¾	6	2	16½	1	8	2¾
9	¾	6	2	17	1	8	2¾
9½	¾	6	2	17½	1	8	2¾
10	¾	6	2	18	1	8	2¾
10½	¾	6	2				

PIPE CONNECTIONS, PIPE FITTINGS

Pipe Threads

Briggs Standard Pipe Thread as adopted by the Committee of Manufacturers on Standardization of Fittings and Valves and the A.S.M.E. in 1913, is the one used in this country; see Table 18. When cutting threads on pipes care must be exercised not to make the thread longer than shown in column *E*.

The Briggs length of thread as shown in this column is sufficient to allow making a tight joint, even should a large percentage of the thread be stripped when being cut. In fact, two or three perfect threads are sufficient to insure making an absolutely tight joint.

When a thread is cut longer than the Briggs Standard, it is necessary with practically all dies on the market to run the thread through the die. The thread which projects through the die is consequently cut straight. This straight thread does not help but in most cases interferes with making a satisfactory joint, as illustrated in Figs. 8 and 9.

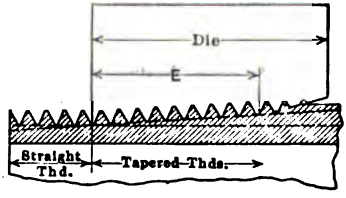


FIG. 8.

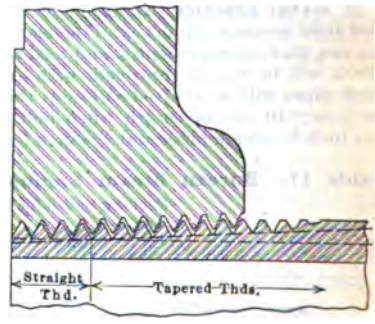


FIG. 9.

Screwed pipe connections are allowed by the Emergency Fleet Corporation on sizes 3 in. and smaller, for pressures not to exceed 250 lbs. For higher pressures, sizes 1 in. and smaller may be screwed. The Bureau of Steam Engineering, U. S. Navy, does not permit screwed joints in steel steam piping.

Flanges, Flanged Connections

The Emergency Fleet Corporation allows **screwed flanges** for pressures not to exceed 125 lbs., in sizes 12 in. and smaller; and for pressures not to exceed 250 lbs., in sizes 4 in. and smaller. Flanges of the **Von Stone type** or of the **Peened type** may be used for any pressure and service. For main and auxiliary steam, flanges must be wrought iron, forged steel or cast steel. For other services, any of the materials described on p. 1254 may be used. The Emergency Fleet Corporation Instructions further require that:

For pressures up to and including 125 lbs. all iron and steel flanged connections shall conform to the **American Standard, "125 pound"** dimensions, Table 19. For pressures up to and including 150 lbs., all non-ferrous flanged connections shall conform to the **1914 Brass Standard "Heavy"** dimensions, Table 21.

For pressures up to and including 250 lbs., all flanged connections shall conform to the **American Standard "250 pound"** dimensions, Table 20, or the **1914 Brass Standard "Extra Heavy"** dimensions, Table 22.

For pressures 251 to 300 lbs. inc., all flanged connections shall conform to the American Standard "Extra Heavy" dimensions, Table 22, except that bolts will be ⅛ in. larger.

For higher pressure flanged connections shall conform to **Hydraulic American Standard**, Table 30, recommended by the American Society of Mechanical Engineers as published in A.S.M.E. Journal of October, 1918.

The Bureau of Steam Engineering, U. S. Navy, Specifications require that: Flanges for all steel steam pipes and oil-fuel pipes will be of Class B forged steel or Class B boiler plate; flanges for wrought iron or steel

Table 18. American Briggs Standard for Taper and Straight Pipe Threads and Locknut Threads

A equals $G - (0.05G + 1.9) \times \frac{1}{N} + \frac{0.8}{N}$ E equals $(0.8G + 4.8) \times \frac{1}{N} + \frac{2}{N}$

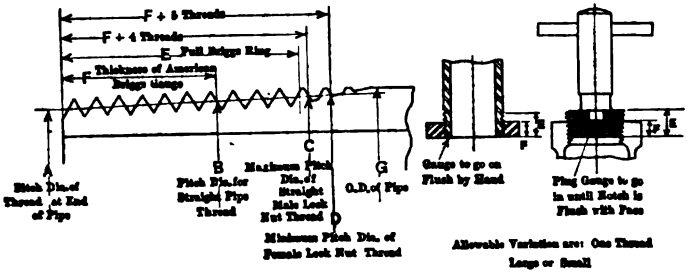
B equals $A + (F \times 0.0625)$

C equals $B + \left(\frac{4}{N} \times 0.0625\right)$

D equals $B + \left(\frac{5}{N} \times 0.0625\right)$

F equals American Briggs Standard.
N equals Number of threads per inch.
Total Taper ¾ inch per foot.

Depth of Thread $\frac{0.8}{N}$


F + 5 Threads
F + 4 Threads
E Full Briggs Ring
F Thickness of American Briggs Gauge
A Pitch Dia. of Thread at End of Pipe
B Pitch Dia. for Straight Pipe Thread
C Maximum Pitch Dia. of Straight Male Lock Nut Thread
D Minimum Pitch Dia. of Female Lock Nut Thread
G O.D. of Pipe
Gauge to go on Flush by Hand
Ring Gauge to go in until Notch is Flush with Face
Allowable Variation are: One Thread Large or Small

Size	A	B	C	D	E	F	G	Depth of thread	Threads per inch
⅛	0.363	0.375	0.384	0.386	0.264	0.180	0.405	0.030	27
¼	0.477	0.490	0.504	0.507	0.402	0.200	0.540	0.044	18
⅜	0.612	0.627	0.641	0.644	0.408	0.240	0.675	0.044	18
½	0.758	0.778	0.796	0.801	0.534	0.320	0.840	0.057	14
¾	0.968	0.989	1.007	1.011	0.546	0.339	1.050	0.057	14
1	1.214	1.239	1.260	1.266	0.683	0.400	1.315	0.070	11½
1¼	1.557	1.583	1.605	1.611	0.707	0.420	1.660	0.070	11½
1½	1.796	1.822	1.844	1.850	0.724	0.420	1.900	0.070	11½
2	2.269	2.296	2.318	2.323	0.757	0.436	2.375	0.070	11½
2½	2.720	2.762	2.793	2.801	1.138	0.682	2.875	0.100	8
3	3.341	3.388	3.420	3.428	1.200	0.766	3.500	0.100	8
3½	3.837	3.889	3.920	3.928	1.250	0.821	4.000	0.100	8
4	4.334	4.387	4.418	4.426	1.300	0.844	4.500	0.100	8
4½	4.831	4.886	4.917	4.925	1.350	0.875	5.000	0.100	8
5	5.391	5.449	5.481	5.488	1.406	0.937	5.563	0.100	8
6	6.446	6.506	6.537	6.545	1.513	0.958	6.625	0.100	8
7	7.440	7.502	7.534	7.541	1.613	1.00	7.625	0.100	8
8	8.434	8.500	8.531	8.539	1.713	1.06	8.625	0.100	8
9	9.427	9.498	9.529	9.537	1.813	1.13	9.625	0.100	8
10	10.54	10.62	10.65	10.66	1.925	1.21	10.75	0.100	8
11	11.54	11.62	11.65	11.66	2.025	1.28	11.75	0.100	8
12	12.53	12.62	12.65	12.66	2.125	1.36	12.75	0.100	8

pipes for salt water, Class B cast steel or Class B forged steel, galvanized; flanges for copper pipes, Brazing Metal, see p. 58, unless otherwise directed. Further the requirements are:

(1) The dimensions of all flanges will be in accordance with the Bureau of Steam Engineering tables of flanges. (See Tables 23 to 27 inc.)

(2) There will be no screw joints in steel steam piping. The pipes will be rolled into the flange and beaded over to fit a recess flush with the face of the flange.

Table 19. American Standard Pipe Flanges for Pressures up to 125 Lbs. per Sq. In. (Dimensions in Inches)

Pipe, diameter	Flanges			Bolt circle, diam.	Pitch chord	Bolts				Bolt hole, diam.
	Diam.	Thickness	Width of face			Number	Diam.	Effective area, sq. in.	Stress, lbs. per sq. in.	
1	4	7/16	1 1/16	3	2.12	4	7/16	0.093	264	9/16
1 1/4	4 1/2	1/2	1 1/16	3 1/16	2.38	4	7/16	0.093	412	9/16
1 1/2	5	9/16	1 1/16	3 1/16	2.73	4	1/2	0.126	438	5/8
2	6	5/8	2	4 1/16	3.35	4	5/8	0.202	486	3/4
2 1/2	7	11/16	2 1/16	5 1/16	3.88	4	5/8	0.202	750	3/4
3	7 1/2	3/4	2 1/16	6	4.23	4	5/8	0.202	1093	3/4
3 1/2	8 1/2	13/16	2 1/16	7	4.94	4	5/8	0.202	1488	3/4
4	9	15/16	2 1/16	7 1/16	2.87	8	5/8	0.202	972	3/4
4 1/2	9 1/16	15/16	2 3/16	7 3/16	2.96	8	3/4	0.302	823	7/8
5	10	15/16	2 1/16	8 1/16	3.25	8	3/4	0.302	1016	7/8
6	11	1	2 1/16	9 1/16	3.63	8	3/4	0.302	1463	7/8
7	12 1/16	1 1/16	2 3/16	10 3/16	4.11	8	3/4	0.302	1991	7/8
8	13 1/16	1 1/16	2 3/16	11 3/16	4.50	8	3/4	0.302	2600	7/8
9	15	1 1/16	3	13 1/16	3.43	12	3/4	0.302	2194	7/8
10	16	1 3/16	3	14 1/16	3.69	12	7/8	0.420	1948	1
12	19	1 1/16	3 1/16	17	4.40	12	7/8	0.420	2805	1
14	21	1 3/16	3 1/16	18 3/16	4.86	12	1	0.550	2915	1 1/16
15	22 1/16	1 3/16	3 3/16	20	3.90	16	1	0.550	2510	1 1/16
16	23 1/16	1 7/16	3 3/16	21 1/16	4.14	16	1	0.550	2856	1 1/16
18	25	1 9/16	3 1/16	22 3/16	4.44	16	1 1/16	0.694	2865	1 1/16
20	27 1/16	1 11/16	3 3/16	25	3.91	20	1 1/16	0.694	2829	1 1/16
22	29 1/16	1 13/16	3 3/16	27 1/16	4.26	20	1 1/16	0.893	2660	1 3/16
24	32	1 7/8	4	29 1/16	4.62	20	1 1/16	0.893	3166	1 3/16
26	34 1/16	2	4 1/16	31 3/16	4.14	24	1 1/16	0.893	3096	1 3/16
28	36 1/16	2 1/16	4 1/16	34	3.81	28	1 1/16	0.893	3078	1 3/16
30	38 3/16	2 1/16	4 3/16	36	4.03	28	1 3/16	1.057	2985	1 1/16
32	41 3/16	2 1/16	4 7/16	38 1/16	4.31	28	1 1/16	1.294	2775	1 3/16
34	43 3/16	2 5/16	4 7/16	40 1/16	3.97	32	1 1/16	1.294	2741	1 3/16
36	46	2 3/16	5	42 3/16	4.19	32	1 1/16	1.294	3073	1 3/16
38	48 3/16	2 3/16	5 3/16	45 1/16	4.43	32	1 3/16	1.515	2924	1 3/16
40	50 3/16	2 1/16	5 3/16	47 1/16	4.11	36	1 3/16	1.515	2880	1 3/16
42	53	2 3/16	5 1/16	49 1/16	4.31	36	1 3/16	1.515	3175	1 3/16
44	55 1/16	2 3/16	5 3/16	51 3/16	4.06	40	1 3/16	1.515	3136	1 3/16
46	57 1/16	2 13/16	5 3/16	53 3/16	4.22	40	1 3/16	1.515	3428	1 3/16
48	59 1/16	2 7/16	5 3/16	56	3.98	44	1 3/16	1.515	3393	1 3/16
50	61 3/16	2 3/16	5 7/16	58 1/16	4.14	44	1 3/16	1.746	3195	1 5/16
52	64	2 7/16	6	60 1/16	4.30	44	1 3/16	1.746	3456	1 5/16
54	66 1/16	3	6 1/16	62 3/16	4.45	44	1 3/16	1.746	3726	1 5/16
56	68 3/16	3	6 3/16	65	4.26	48	1 3/16	1.746	3674	1 5/16
58	71	3 1/16	6 1/16	67 1/16	4.40	48	1 3/16	1.746	3941	1 5/16
60	73	3 1/16	6 1/16	69 1/16	4.19	52	1 3/16	1.746	3892	1 5/16
62	75 3/16	3 1/16	6 7/16	71 3/16	4.34	52	1 7/16	2.051	3538	2
64	78	3 1/16	7	74	4.48	52	1 7/16	2.051	3770	2
66	80	3 3/16	7	76	4.60	52	1 7/16	2.051	4010	2
68	82 1/16	3 3/16	7 1/16	78 1/16	4.38	56	1 7/16	2.051	3952	2
70	84 1/16	3 1/16	7 1/16	80 1/16	4.51	56	1 7/16	2.051	4188	2
72	86 1/16	3 1/16	7 1/16	82 1/16	4.33	60	1 7/16	2.051	4136	2
74	88 1/16	3 3/16	7 1/16	84 1/16	4.44	60	1 7/16	2.051	4368	2
76	90 3/16	3 3/16	7 3/16	86 1/16	4.54	60	1 7/16	2.051	4608	2
78	93	3 3/16	7 1/16	88 3/16	4.66	60	2	2.302	4325	2 1/16
80	95 1/16	3 3/16	7 5/16	91	4.78	60	2	2.302	4549	2 1/16
82	97 1/16	3 7/16	7 3/16	93 1/16	4.90	60	2	2.302	4779	2 1/16
84	99 3/16	3 7/16	7 7/16	95 1/16	4.66	64	2	2.302	4702	2 1/16
86	102	4	8	97 3/16	4.79	64	2	2.302	4928	2 1/16
88	104 1/16	4	8 1/16	100	4.60	68	2	2.302	4857	2 1/16
90	106 1/16	4 1/16	8 1/16	102 1/16	4.71	68	2 1/16	2.648	4416	2 1/16
92	108 3/16	4 1/16	8 3/16	104 1/16	4.81	68	2 1/16	2.648	4615	2 1/16
94	111	4 1/16	8 1/16	106 1/16	4.89	68	2 1/16	2.648	4817	2 1/16
96	113 1/16	4 1/16	8 5/16	108 1/16	4.99	68	2 1/16	3.023	4401	2 3/16
98	115 1/16	4 3/16	8 3/16	110 3/16	5.09	68	2 1/16	3.023	4587	2 3/16
100	117 3/16	4 3/16	8 7/16	113	5.20	68	2 1/16	3.023	4776	2 3/16

Table 20. American Standard Extra Heavy Pipe Flanges For Pressures up to 250 Lbs. per Sq. In. (Dimensions in Inches)

Pipe diameter	Flanges: Diameter	Flanges: Thickness*	Flanges: Width of face	Bolt circle, diam.	Pitch chord	Bolts: Number	Bolts: Diameter, in.	Bolts: Effective area, sq. in.	Bolts: Stress, lbs. per sq. in.	Bolt hole, diam.
1	4½	1 1/16	1¾	3¼	2.29	4	½	0.126	389	5/8
1¼	5	¾	1⅞	3¼	2.65	4	½	0.126	609	5/8
1½	6	13/16	2	4½	3.17	4	5/8	0.202	547	¾
2	6½	⅞	2⅛	5	3.53	4	5/8	0.202	972	¾
2½	7½	1	2½	5⅞	4.15	4	¾	0.302	1016	⅞
3	8¼	1⅛	2⅝	6⅝	2.53	8	¾	0.302	731	⅞
3½	9	1 3/16	2¾	7¼	2.77	8	¾	0.302	995	⅞
4	10	1¼	3	7⅞	3.01	8	¾	0.302	1300	⅞
4½	10½	1 5/16	3	8½	3.25	8	¾	0.302	1646	⅞
5	11	1⅜	3	9¼	3.53	8	¾	0.302	2032	⅞
6	12½	1 7/16	3¼	10⅝	2.75	12	¾	0.302	1950	⅞
7	14	1½	3½	11⅞	3.07	12	⅞	0.420	1909	1
8	15	1 5/8	3½	13	3.36	12	⅞	0.420	2493	1
9	16¼	1¾	3⅝	14	3.62	12	1	0.550	2410	1⅛
10	17½	1⅞	3¾	15¼	2.97	16	1	0.550	2231	1⅛
12	20½	2	4¼	17¾	3.46	16	1⅛	0.694	2546	1¼
14	23	2⅛	4½	20¼	3.17	20	1⅛	0.694	2773	1¼
15	24½	2 3/16	4¾	21½	3.36	20	1¼	0.893	2473	1⅜
16	25½	2¼	4¾	22½	3.52	20	1¼	0.893	2814	1⅜
18	28	2⅜	5	24¾	3.23	24	1¼	0.893	2968	1⅜
20	30½	2½	5¼	27	3.52	24	1⅜	1.057	3096	1½
22	33	2⅝	5¼	29¼	3.81	24	1½	1.295	3058	1⅝
24	36	2¾	5¾	32	4.18	24	1⅝	1.515	3110	1¾
26	38¼	2 13/16	6⅛	34½	3.86	28	1⅝	1.515	3126	1¾
28	40¾	2 15/16	6⅜	37	4.14	28	1⅝	1.515	3629	1¾
30	43	3	6½	39¼	4.38	28	1¾	1.746	3615	1⅞
32	45¼	3⅛	6⅝	41½	4.64	28	1⅞	2.051	3501	2
34	47½	3¼	6¾	43½	4.87	28	1⅞	2.051	3952	2
36	50	3⅜	7	46	4.50	32	1⅞	2.051	3877	2
38	52¼	3 7/16	7⅛	48	4.70	32	1⅞	2.051	4320	2
40	54½	3 9/16	7¼	50¼	4.38	36	1⅞	2.051	4255	2
42	57	3 11/16	7½	52¾	4.59	36	1⅞	2.051	4691	2
44	59¼	3¾	7⅝	55	4.79	36	2	2.302	4587	2⅛
46	61½	3⅞	7¾	57¼	4.49	40	2	2.302	4512	2⅛
48	65	4	8½	60¾	4.76	40	2	2.302	4913	2⅛

* Thickness of flange given in table includes raised face.

(3) Flanges will not be riveted to pipes without authority from the Bureau.

(4) All flanges will be faced. No "male and female" flange joints will be fitted.

(5) All flanges will be finished on their faces and when flanges are not lagged the edges will be finished. The backs of all flanges which are conspicuously located will be finished; others may be spot faced.

(6) Bolts for flanges will have undersides of heads and nuts properly faced, and the bolt holes will not be more than 1/16 in. larger in diameter than the bolts.

(7) Copper pipes will have flanges or approved couplings brazed on, and

Table 21. Brass Flanges, "Heavy," 1914 Standard for Pressures up to 150 Lbs.

Pipe size, in.	Diameter of flange, in.	Thickness of flange, in.	Bolt circle, in.	Number of bolts	Size of bolts, in.	Length of bolts, in.
¼ and ⅜	2½	9/32	1 11/16	4	⅜	1
½	3	5/16	2⅛	4	⅜	1¼
¾	3½	11/32	2½	4	⅜	1¼
1	4	⅜	3	4	7/16	1¼
1¼	4½	13/32	3⅜	4	7/16	1½
1½	5	7/16	3⅞	4	½	1½
2	6	½	4¾	4	⅝	1¾
2½	7	9/16	5½	4	⅝	2
3	7½	⅝	6	4	⅝	2
3½	8½	11/16	7	4	⅝	2¼
4	9	11/16	7½	8	⅝	2¼
4½	9¼	25/32	7¾	8	¾	2½
5	10	¾	8½	8	¾	2½
6	11	13/16	9½	8	¾	2¾
7	12½	⅞	10¾	8	¾	2¾
8	13½	15/16	11¾	8	¾	3
9	15	15/16	13¼	12	¾	3
10	16	1	14¼	12	⅞	3¼
12	19	1 1/16	17	12	⅞	3¼

Table 22. Brass Flanges, "Extra Heavy," 1914 Standard for Pressures up to 250 Lbs.

Pipe size, in.	Diameter of flange, in.	Thickness of flange, in.	Bolt circle, in.	Number of bolts	Size of bolts, in.	Length of bolts, in.
¼ and ⅜	3	⅜	2	4	7/16	1¼
½	3½	13/32	2⅜	4	7/16	1½
¾	4	7/16	2⅞	4	½	1½
1	4½	½	3¼	4	½	1¾
1¼	5	17/32	3¾	4	½	1¾
1½	6	9/16	4½	4	⅝	2
2	6½	⅝	5	4	⅝	2
2½	7½	11/16	5⅞	4	¾	2¼
3	8¼	¾	6⅝	8	¾	2½
3½	9	13/16	7¼	8	¾	2¾
4	10	⅞	7¾	8	¾	2¾
4½	10½	⅞	8½	8	¾	2¾
5	11	15/16	9¼	8	¾	3
6	12½	1	10⅝	12	¾	3
7	14	1 1/16	11⅞	12	⅞	3¼
8	15	1⅛	13	12	⅞	3½
9	16¼	1⅛	14	12	1	3½
10	17½	1 3/16	15¼	16	1	3¾
12	20½	1¼	17¾	16	1⅛	4

Bolt holes drilled 1/16 in. larger than bolt diameter, 2 in. size and smaller and 1/8 in. larger for 2½ in. size and larger. "Heavy" and "Extra Heavy" flanges to have plain smooth faces. Full Faced Gaskets should be used. When bolted to Extra Heavy iron flanges, valves or fittings, the latter should be ordered with plain faces. Diameters given are finished dimensions.

the end of the pipe will be beaded into a recess in the face of the flange. (See Tables 24 to 26 inc.)

The *Newport News Shipbuilding and Dry Dock Co.*, describe their practice in pipe flanges as follows:

For Government vessels we use Bureau of Steam Engineering and Bureau of Construction & Repair Standards. For merchant vessels we have adopted the following standards.

For High Pressure, *i.e.*, over 125 lbs., flanged diameters and boltings agreeing with the "American" Standards.

For Low Pressure, *i.e.*, 125 lbs. and below, we use, usually, the "American" Standard High Pressure flange diameters in connection with the "American" Standard Low Pressure bolting. Our standard flanges are bored in accordance with our lists (which agree closely with Bureau Steam Engineering lists) for copper and steel pipes respectively, except that the chamfered face flange has not proven satisfactory and its use has therefore been abandoned.

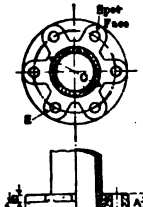
A comparison of the American Standard flanges and of Navy Standard flanges show two marked differences which are discussed by *Mr. A. G. Christie* in the *Jour. A. S. M. E.*, Feb., 1919, as follows:

"Shipbuilders generally have endeavored to use A.S.M.E. standards for flanges and drilling. However, some criticisms have been heard. For instance, many marine engineers do not favor the raised faces on extra heavy flanges. They prefer flat faces. The flat faces are also favored for bolting to bulkheads and to other ship plates."

"The bolting specified for certain sizes in the A.S.M.E. standards has not been satisfactory to many marine designers. For instance, in the 3-in. and 3½-in. standard-weight flanges the A.S.M.E. standard calls for four holes, while many marine designers use six holes. Also, in the 2-in. and 2½-in. extra heavy flanges the marine engineers frequently use six holes where the A.S.M.E. standards specify four. The A.S.M.E. standards increase the holes at one jump from four to eight. There seems considerable justification for the use of the six-hole flange."

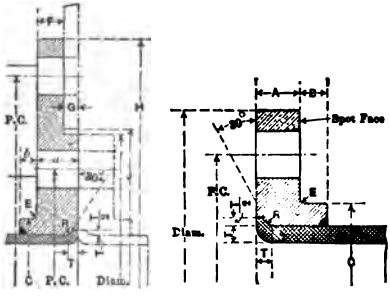
Table 23. Composition Flanges for Screwed Pipes, Iron Pipe Size Pressures to 100 Lbs. per Sq. In.

Bureau of Steam Engineering Standards



Nominal diameter of pipe	Size of flanges					Bolts			
	Diam.	A	B	C	E	No.	Diam.	P.C.	Pitch chord
¼	3	3/8	1/16	7/8	9/16	3	½	1 7/8	1.62
⅜	3 1/8	3/8	1/16	1	9/16	3	½	2	1.73
½	3 1/4	3/8	1/8	1 3/16	9/16	3	½	2 1/8	1.84
¾	3 9/16	3/8	1/8	1 1/2	9/16	3	½	2 5/16	2.11
1	3 13/16	7/16	1/8	1 3/4	9/16	4	½	2 11/16	1.90
1¼	4 1/4	7/16	3/16	2 1/8	9/16	4	½	3 1/8	2.21
1½	4 1/2	7/16	3/16	2 3/8	9/16	4	½	3 3/8	2.39
2	5 1/16	7/16	¼	3	9/16	6	½	3 15/16	1.97
2½	5 9/16	7/16	½	3 9/16	9/16	6	½	4 5/16	2.22
3	6 1/8	½	½	4 3/16	9/16	6	½	5	2.50
3½	6 5/8	½	9/16	4 3/4	9/16	8	½	5 1/2	2.10
4	7 3/16	½	5/8	5 5/16	9/16	8	½	6 1/16	2.32
4½	7 11/16	½	5/8	6	9/16	8	½	6 9/16	2.51
5	8 3/16	½	1 1/16	6 1/2	9/16	10	½	7 1/16	2.18
6	9 5/16	9/16	1 1/16	7 3/4	5/8	10	5/8	8 5/16	2.57

Holes for bolts to be drilled 1/16 in. larger than diameter of bolts. Flanges will not be finished on the edges, provided they are cast smooth and true.



Composition Flanges and Bulkhead Flanges for Copper Pipes Pressures from 1 to 500 Lbs. per Sq. In., See Tables 24, 25 and 26.

Table 24. Composition Flanges and Bulkhead Flanges for Copper Pipes
Pressures to 100 Lbs. per Sq. In. See Figure on p. 1277

Bureau of Steam Engineering Standards

Internal diam. of pipe	Size of flanges											Bolts, pipe flange				Bolts, blkhd. flange			
	Diam.	A	B	a	b	C	E	F	G	H	J	No.	Diam.	P.C.	Pitch chord	No.	Diam.	P.C.	Pitch chord
¼	3	3/8	1/4	7/8	3/16	3/4	1/16	3/8	1/2	5 13/16	3 3/16	3	1/2	1 7/8	1.62	4	1/2	4 5/8	3.27
½	3 1/4	3/8	1/4	7/8	3/16	1	1/16	3/8	1/2	6 1/16	3 3/16	3	1/2	2 1/8	1.84	4	1/2	4 7/8	3.45
¾	3 9/16	3/8	1/4	7/8	3/16	1 5/16	1/16	3/8	1/2	6 3/16	3 3/16	3	1/2	2 1/16	2.11	4	1/2	5 1/16	3.58
1	3 13/16	7/16	1/4	7/8	3/16	1 9/16	1/16	3/8	1/2	6 7/16	4	4	1/2	2 3/16	1.90	6	1/2	5 5/16	2.66
1¼	4 1/4	7/16	1/4	7/8	3/16	1 13/16	1/16	3/8	1/2	6 7/8	4 3/16	4	1/2	3 1/8	2.21	6	1/2	5 3/4	2.87
1½	4 1/2	7/16	9/16	7/8	3/16	2 1/16	1/16	3/8	1/2	7 1/16	4 5/16	4	1/2	3 3/8	2.39	6	1/2	6	3.00
2	5 1/16	7/16	9/16	1	3/16	2 5/8	1/16	1/2	1/2	7 13/16	5 1/4	6	1/2	3 15/16	1.97	6	1/2	6 9/16	3.28
2½	5 9/16	7/16	9/16	1	3/16	3 1/16	1/16	1/2	1/2	8 3/16	5 3/4	6	1/2	4 7/16	2.22	6	1/2	7 1/16	3.53
3	6 1/8	1/2	9/16	1	3/16	3 13/16	1/16	1/2	1/2	8 3/4	6 1/4	6	1/2	5	2.50	6	1/2	7 5/8	3.81
3½	6 5/8	1/2	9/16	1	3/16	4 5/16	1/16	1/2	1/2	9 1/4	6 3/4	8	1/2	5 1/2	2.10	8	1/2	8 1/4	3.11
4	7 3/16	1/2	5/8	1 3/16	3/16	4 3/4	1/16	1/2	1/2	9 13/16	7 3/16	8	1/2	6 1/16	2.32	8	1/2	8 13/16	3.32
4½	7 11/16	1/2	5/8	1 3/16	3/16	5 1/4	1/16	9/16	1/2	10 5/16	7 7/8	8	1/2	6 9/16	2.51	8	1/2	9 5/16	3.51
5	8 3/16	1/2	5/8	1 3/16	3/16	5 3/4	1/16	9/16	1/2	10 13/16	8 3/4	10	1/2	7 1/16	2.18	10	1/2	9 13/16	2.99
5½	9 1/16	9/16	5/8	1 3/16	3/16	6 3/16	1/16	9/16	1/2	12 3/8	9 1/4	10	5/8	7 13/16	2.41	10	5/8	10 1/16	3.38
6	9 9/16	9/16	3/8	1 3/16	3/16	6 13/16	1/16	9/16	1/2	12 7/8	9 3/4	10	5/8	8 5/16	2.57	10	5/8	11 1/16	3.53
6½	10 1/8	9/16	7/16	1 1/8	1/4	7 3/8	1/16	5/8	1/2	13 7/16	10 1/4	12	1/2	8 1/8	2.30	12	5/8	12	3.11
7	10 3/8	9/16	7/16	1 1/8	1/4	7 7/8	1/16	5/8	1/2	13 15/16	10 3/4	12	1/2	9 5/8	2.43	12	5/8	12 1/2	3.24
7½	11 5/16	9/16	7/16	1 1/8	1/4	8 3/16	3/16	5/8	1/2	14 1/16	11 1/4	12	1/2	10	2.59	12	5/8	13 5/16	3.41
8	11 7/8	5/8	7/16	1 1/8	1/4	8 13/16	3/16	5/8	1/2	15 3/16	12	12	1/2	10 9/16	2.73	12	5/8	13 3/4	3.56
8½	12 3/8	5/8	7/16	1 1/8	1/4	9 5/16	3/16	5/8	1/2	15 13/16	12 1/2	14	1/2	11 1/16	2.46	14	5/8	14 1/4	3.17
9	12 15/16	5/8	7/16	1 1/8	1/4	10	3/16	5/8	1/2	16 3/4	13 1/4	14	1/2	11 5/8	2.59	14	5/8	14 5/16	3.30
9½	13 5/16	5/8	1/2	1 1/8	1/4	10 1/2	3/16	5/8	1/2	18 5/16	14 1/16	14	3/4	12 3/8	2.75	14	3/4	16 9/16	3.69
10	14 1/2	5/8	1/2	1 1/8	1/4	11 1/16	3/16	5/8	1/2	18 7/8	14 7/16	14	3/4	12 15/16	2.88	14	3/4	17 1/16	3.81
10½	15	11/16	1/2	1 1/8	1/4	11 9/16	3/16	5/8	1/2	19 3/8	15 1/16	15	3/4	13 7/16	2.79	16	3/4	17 5/8	3.44
11	15 1/2	11/16	1/2	1 1/8	1/4	12 1/16	3/16	5/8	1/2	19 7/8	15 5/8	15	3/4	13 15/16	2.90	16	3/4	18 1/16	3.54
11½	16	11/16	9/16	1 1/8	1/4	12 9/16	3/16	5/8	1/2	20 5/8	16 1/16	16	3/4	14 7/16	2.82	16	3/4	18 5/8	3.63
12	16 9/16	11/16	9/16	1 1/8	1/4	13 1/16	3/16	5/8	1/2	21	16 1/4	16	3/4	15	2.93	16	3/4	19 1/4	3.75
12½	17 1/16	11/16	9/16	1 1/8	1/4	13 5/8	3/16	5/8	1/2	21 1/2	17 1/4	16	3/4	15 1/2	3.02	16	3/4	19 3/4	3.85
13	17 5/8	3/4	9/16	1 3/16	1/4	14 3/16	3/16	11/16	1/2	22	17 3/4	18	3/4	16 1/16	2.79	18	3/4	20 1/4	3.51
13½	18 1/8	3/4	9/16	1 3/16	1/4	14 11/16	3/16	11/16	1/2	22 1/2	18 1/4	18	3/4	16 9/16	2.90	18	3/4	20 3/4	3.62
14	19 1/8	3/4	9/16	1 3/16	1/4	15 1/4	3/16	11/16	1/2	23 1/2	19 1/4	19	7/8	17 3/8	2.86	18	3/4	21 3/8	3.77
14½	19 5/8	3/4	9/16	1 3/16	1/4	15 3/4	3/16	11/16	1/2	24	19 3/4	19	7/8	17 7/8	2.94	20	3/4	22 1/4	3.48
15	20 1/8	3/4	5/8	1 1/4	1/4	16 1/4	3/16	3/4	1/2	24 1/2	20 1/4	20	7/8	18 3/8	2.87	20	3/4	22 3/4	3.56
15½	20 3/4	3/4	5/8	1 1/4	1/4	16 3/4	3/16	3/4	1/2	25	20 1/4	20	7/8	18 7/8	2.95	20	3/4	23 1/4	3.64
16	21 3/16	3/4	5/8	1 1/4	1/4	17 5/16	3/16	3/4	1/2	25 3/4	21 3/16	20	7/8	19 7/16	3.04	20	3/4	23 7/8	3.73
16½	21 11/16	3/4	5/8	1 1/4	1/4	17 13/16	3/16	3/4	1/2	26 1/4	21 7/8	20	7/8	19 15/16	3.12	20	3/4	24 1/2	3.66
17	22 1/4	3/4	5/8	1 1/4	1/4	18 3/8	3/16	3/4	1/2	26 3/4	22 3/4	21	7/8						

18	23¼	⅞	5⅛	1¼	¼	19⅜	5/16	¾	½	27⅝	23¾	22	⅞	21⅛	9.06	22	¾	25⅞	3.68
18½	23¾	⅞	5⅛	1¼	¼	19⅝	5/16	¾	½	28⅛	23⅞	22	⅞	22	3.13	22	¾	26⅜	3.75
19	24¾	⅞	5⅛	1¼	¼	20 1/16	5/16	¾	½	29⅛	24⅞	23	1	22¾	3.10	24	¾	27⅜	3.57
19½	25¼	⅞	5⅛	1¼	¼	20 5/16	5/16	¾	½	29⅝	25¾	23	1	23¼	3.16	24	¾	27⅞	3.64
20	25 5/16	⅞	5⅛	1¼	¼	21½	5/16	¾	½	30¼	26	24	1	23 5/16	3.11	24	¾	28½	3.72
20½	26 9/16	⅞	1 1/16	1¼	¼	22	5/16	¾	½	30¾	26½	24	1	24 5/16	3.17	24	¾	29	3.78
21	26 13/16	⅞	1 1/16	1¼	5/16	22½	5/16	¾	½	31¼	27	25	1	24 5/16	3.11	26	¾	29½	3.55
21½	27 5/16	⅞	1 1/16	1¼	5/16	23	5/16	¾	½	31¾	27½	25	1	25 5/16	3.17	26	¾	30	3.62
22	27⅞	⅞	1 1/16	1¼	5/16	23 9/16	5/16	¾	½	32¼	28	26	1	25⅞	3.12	26	¾	30½	3.68
22½	28⅜	⅞	1 1/16	1¼	5/16	24 1/16	5/16	¾	½	32¾	28½	26	1	26⅜	3.18	26	¾	31	3.74
23	28 13/16	1	1 1/16	1¼	7/16	24⅝	5/16	¾	½	33⅜	29⅛	27	1	26 13/16	3.13	26	¾	31⅝	3.81
23½	29 5/16	1	1 1/16	1¼	7/16	25⅛	5/16	¾	½	33⅞	29⅝	27	1	27 1/16	3.18	26	¾	32⅛	3.87
24	30	1	1 1/16	1¼	7/16	25 1/16	5/16	¾	½	34⅜	30⅛	28	1	28	3.13	28	¾	32⅝	3.65
24½	30½	1	⅝	1¼	7/16	26 5/16	5/16	¾	½	34⅞	30⅝	28	1	28⅛	3.19	28	¾	33⅛	3.71
25	31½	1	⅝	1¼	½	26 1/16	5/16	¾	½	35⅜	31⅝	29	1⅛	29¼	3.16	28	¾	34⅛	3.82
25½	32	1	⅝	1¼	½	27 5/16	5/16	¾	½	36⅜	32⅛	29	1⅛	29⅜	3.22	28	¾	34⅝	3.88
26	32 9/16	1	⅝	1¼	½	27¾	5/16	¾	½	37	32¾	30	1⅛	30 5/16	3.17	30	¾	35¼	3.68
26½	33 1/16	1	⅝	1¼	½	28¼	5/16	¾	½	37½	33¼	30	1⅛	30 9/16	3.22	30	¾	35¾	3.74
27	33⅜	1	⅝	1¼	½	28 13/16	5/16	¾	½	38	33⅝	31	1⅛	31⅜	3.18	30	¾	36¼	3.79
27½	34⅛	1	⅝	1¼	½	29 5/16	5/16	¾	½	38½	34¼	31	1⅛	31⅞	3.23	30	¾	36¾	3.84
28	34 1/16	1	⅝	1¼	½	29⅞	5/16	¾	½	39⅛	34⅞	32	1⅛	32 7/16	3.18	32	¾	37⅜	3.66
28½	35 5/16	1	⅝	1¼	½	30⅜	5/16	¾	½	39⅝	35⅜	32	1⅛	32 15/16	3.23	32	¾	37⅞	3.71
29	35¾	1⅛	⅝	1¼	⅝	30 15/16	5/16	¾	½	40⅛	35⅞	33	1⅛	33½	3.19	32	¾	38⅜	3.76
29½	36¼	1⅛	⅝	1¼	⅝	31 7/16	5/16	¾	½	40⅝	36⅜	33	1⅛	34	3.23	32	¾	38⅞	3.81
30	36 13/16	1⅛	⅝	1¼	⅝	32	5/16	¾	½	41¼	37	35	1⅛	34 9/16	3.10	34	¾	39½	3.64
30½	37⅜	1⅛	⅝	1¼	⅝	32½	¼	¾	½	41¾	37⅝	35	1⅛	35⅛	3.15	34	¾	40	3.69
31	37⅞	1⅛	⅝	1¼	⅝	33	¼	¾	½	42¼	38⅛	35	1⅛	35⅝	3.19	34	¾	40½	3.74
31½	38⅜	1⅛	⅝	1¼	⅝	33½	¼	¾	½	42¾	38⅝	35	1⅛	36⅛	3.24	34	¾	41	3.78
32	39	1⅛	⅞	1⅜	⅝	34⅛	¼	⅞	½	43⅞	39¼	36	1⅛	36¾	3.20	36	⅞	41⅞	3.65
32½	39½	1⅛	⅞	1⅜	⅝	34⅝	¼	⅞	½	44⅛	39¾	36	1⅛	37¼	3.25	36	⅞	42⅜	3.69
33	40	1⅛	⅞	1⅜	⅝	35⅛	¼	⅞	½	44⅞	40¼	36	1⅛	37¾	3.29	36	⅞	42⅞	3.73
33½	40½	1⅛	⅞	1⅜	⅝	35⅝	¼	⅞	½	45⅜	40⅝	36	1⅛	38¼	3.33	36	⅞	43⅜	3.78
34	41	1⅛	⅞	1⅜	⅝	36⅛	¼	⅞	½	45⅞	41⅛	36	1⅛	38¾	3.38	36	⅞	43⅞	3.82
34½	42⅜	1⅛	⅞	1⅜	⅝	36¾	⅜	⅞	½	47¼	42⅛	36	1¼	39⅞	3.47	36	⅞	45¼	3.94
35	42⅞	1¼	⅞	1⅜	⅝	37¼	⅜	⅞	½	47¾	43¼	36	1¼	40⅜	3.52	36	⅞	45¾	3.99
35½	43⅜	1¼	⅞	1⅜	⅝	37¾	⅜	⅞	½	48¼	43⅝	36	1¼	40⅞	3.56	36	⅞	46¼	4.03
36	43⅞	1¼	⅞	1⅜	⅝	38¼	⅜	⅞	½	48¾	44⅛	36	1¼	41⅜	3.61	36	⅞	46¾	4.07
36½	44½	1¼	⅞	1⅜	⅝	38⅞	⅜	⅞	½	49⅜	44¾	36	1¼	42	3.66	36	⅞	47⅜	4.12
37	45	1¼	⅞	1⅜	⅝	39⅜	⅜	⅞	½	49⅞	45⅛	36	1¼	42½	3.70	36	⅞	47⅞	4.17
37½	45½	1¼	⅞	1⅜	¾	39⅞	⅜	⅞	½	50⅜	45¾	36	1¼	43	3.75	36	⅞	48⅜	4.21
38	46⅛	1¼	1	1⅜	⅞	40½	⅜	⅞	½	51	46⅜	36	1¼	43⅝	3.81	36	⅞	49	4.27
38½	46⅝	1¼	1	1⅜	⅞	41	⅜	⅞	½	51½	46⅞	36	1¼	44⅛	3.85	36	⅞	49½	4.31
39	47⅛	1¼	1	1⅜	⅞	41½	⅜	⅞	½	52	47⅜	36	1¼	44⅝	3.89	36	⅞	50	4.36
39½	47⅝	1¼	1	1⅜	⅞	42	⅜	⅞	½	52½	47⅞	36	1¼	45⅛	3.93	36	⅞	50½	4.40
40	48⅛	1¼	1	1⅜	⅞	42½	⅜	⅞	½	53	48⅜	36	1¼	45⅝	3.98	36	⅞	51	4.41

Table 25. Composition Flanges and Bulkhead Flanges for Copper Pipes
Pressures From 101 to 300 Lbs. per Sq. In. See Figure on p. 1277

Bureau of Steam Engineering Standards

Internal diam. of pipe	Size of flanges										J	Bolts, pipe flange				Bolts, blkhd. flange			
	Dia.	A	B	a	b	C	E	F	G	H		No.	Dia.	P.C.	Pitch chord	No.	Dia.	P.C.	Pitch coord
¼	3 7/16	9/16	3/16	7/16	3/16	1 5/16	3/16	3/8	1/2	6 1/4	3 5/8	3	1/2	2 5/16	2.00	6	1/2	5 1/4	2.62
½	3¾	5/8	3/16	7/16	3/16	1 3/16	3/16	3/8	1/2	6 3/4	3 7/8	3	1/2	2 5/8	2.27	6	1/2	5 1/2	2.75
¾	4	5/8	3/16	7/16	3/16	1 1/2	3/16	3/8	1/2	7	4 1/8	4	1/2	2 7/8	2.03	6	1/2	5 3/4	2.81
1	4 5/16	1 1/16	3/16	7/16	3/16	1 3/4	3/16	3/8	1/2	7 3/8	4 1/2	4	1/2	3 3/16	2.25	6	1/2	6 1/4	3.06
1¼	5 1/16	1 1/16	3/16	1	3/16	2 1/16	3/16	1/2	1/2	8 1/8	5 1/4	5	5/8	3 3/4	2.20	8	1/2	6 7/8	2.63
1½	5 3/8	1 1/16	3/16	1	3/16	2 5/16	3/16	1/2	1/2	8 3/8	5 1/2	5	5/8	4 1/16	2.39	8	1/2	7 1/8	2.73
2	5 15/16	3/4	3/16	1	3/16	2 7/8	1/4	1/2	1/2	9	6 1/8	6	5/8	4 5/8	2.31	8	1/2	7 3/4	2.96
2½	6 1/2	3/4	3/16	1 1/16	3/16	3 1/16	1/4	9/16	1/2	10 1/4	6 3/4	7	5/8	5 5/16	2.25	10	5/8	8 3/4	2.70
3	7 9/16	1 3/16	3/16	1 1/16	3/16	4	1/4	9/16	1/2	11 3/8	7 3/4	8	3/4	6	2.30	10	5/8	9 7/8	3.05
3½	8 1/8	1 3/16	1/4	1 1/16	3/16	4 9/16	1/4	9/16	1/2	11 7/8	8 1/4	8	3/4	6 9/16	2.51	10	5/8	10 5/8	3.21
4	8 15/16	7/8	1/4	1 1/8	3/16	5 1/8	1/4	5/8	1/2	12 1/2	8 7/8	9	3/4	7 1/8	2.44	12	5/8	11	3.95
4½	9 1/16	7/8	1/4	1 1/8	3/16	5 13/16	1/4	5/8	1/2	13	9 1/8	9	3/4	7 11/16	2.63	12	5/8	11 1/2	4.13
5	9 13/16	1 5/16	1/4	1 1/8	3/16	6 1/4	1/4	5/8	1/2	13 3/8	10	10	3/4	8 1/4	2.55	12	5/8	12 1/8	4.35
5½	10 5/8	1 5/16	5/16	1 1/8	3/16	6 13/16	1/4	5/8	1/2	14 1/8	10 1/8	11	3/4	8 15/16	2.48	12	5/8	12 3/8	4.46
6	11 3/8	1	5/16	1 1/8	3/16	7 3/8	1/4	5/8	1/2	15 3/8	11 1/2	11	7/8	9 5/8	2.71	12	3/4	14	5.02
6½	11 15/16	1	5/16	1 1/8	3/16	7 15/16	1/4	5/8	1/2	16 3/8	12 1/8	12	7/8	10 5/16	2.64	12	3/4	14 5/8	5.25
7	12 9/16	1 5/16	5/16	1 5/16	3/16	8 1/2	1/4	1 1/16	1/2	17	12 3/4	12	7/8	10 13/16	2.80	14	3/4	15 1/4	3.39
7½	13 1/8	1 5/16	3/8	1 5/16	1/4	9 1/16	1/4	1 1/16	1/2	17 1/2	13 1/4	12	7/8	11 3/8	2.94	14	3/4	15 3/4	3.50
8	14 3/16	1 1/16	3/8	1 5/16	3/16	9 5/8	5/16	1 1/16	1/2	18 5/8	14 1/8	12	1	12 3/16	3.15	14	3/4	16 7/8	3.75
8½	14 3/4	1 1/16	3/8	1 5/16	3/16	10 1/16	5/16	1 1/16	1/2	19 1/8	14 7/8	13	1	12 3/4	3.05	16	3/4	17 3/8	3.39
9	15 5/16	1 5/16	3/8	1 5/16	3/16	10 3/4	5/16	1 1/16	1/2	19 3/4	15 1/2	13	1	13 5/16	3.19	16	3/4	18	3.51
9½	15 7/8	1 5/16	7/16	1 5/16	7/16	11 1/16	5/16	1 1/16	1/2	20 1/4	16	14	1	13 7/8	3.09	16	3/4	18 1/2	3.61
10	16 7/16	1 1/16	7/16	1 1/16	7/16	11 7/8	5/16	1 1/16	5/16	20 7/8	16 5/8	15	1	14 5/16	3.00	16	3/4	19 1/4	3.73
10½	17	1 1/16	7/16	1 1/16	7/16	12 5/16	5/16	1 1/16	5/16	21 5/8	17 1/8	15	1	15	3.12	18	3/4	19 5/8	3.41
11	17 9/16	1 5/16	7/16	1 5/16	7/16	13	5/16	3/4	5/16	22	17 3/4	15	1	15 5/16	3.24	18	3/4	20 1/4	3.51
11½	18 5/16	1 5/16	1/2	1 5/16	1/2	13 9/16	5/16	3/4	5/16	23	18 3/4	15	1 1/8	16 3/8	3.40	18	3/4	21 1/4	3.69
12	19 1/16	1 3/16	1/2	1 3/16	1/2	14 1/16	5/16	3/4	5/16	23 1/2	19 1/4	16	1 1/8	16 15/16	3.30	18	3/4	21 3/4	3.78
12½	19 3/4	1 3/16	1/2	1 3/16	1/2	14 11/16	5/16	3/4	5/16	24 1/8	19 7/8	16	1 1/8	17 5/16	3.43	18	3/4	22 3/8	3.88
13	20 5/16	1 7/16	1/2	1 7/16	1/2	15 1/4	3/8	3/4	1 1/16	24 3/4	20 1/2	17	1 1/8	18 1/16	3.33	20	3/4	23	3.60
13½	20 7/8	1 7/16	9/16	1 7/16	9/16	15 7/8	3/8	3/4	1 1/16	25 1/4	21	18	1 1/8	18 11/16	3.24	20	3/4	23 1/8	3.67
14	21 7/16	1 1/2	9/16	1 1/2	9/16	16 3/8	3/8	3/4	3/4	25 7/8	21 5/8	18	1 1/8	19 1/4	3.34	20	3/4	24 1/8	3.77
14½	22 1/2	1 1/2	9/16	1 1/2	9/16	16 15/16	3/8	3/4	3/4	26 1/2	22 1/8	18	1 1/4	20 1/16	3.48	20	3/4	25 1/8	3.93
15	23	1 9/16	9/16	1 9/16	9/16	17 1/2	3/8	3/4	1 1/16	27 3/8	23 1/8	18	1 1/4	20 9/16	3.57	22	3/4	25 3/8	3.65
15½	23 5/8	1 9/16	5/8	1 9/16	5/8	18	3/8	3/4	1 1/16	28	23 3/4	18	1 1/4	21 1/8	3.67	22	3/4	26 1/4	3.75
16	24 1/4	1 9/16	5/8	1 9/16	5/8	18 1/2	3/8	3/4	7/8	28 5/8	24 3/8	18	1 1/4	21 5/8	3.78	22	3/4	26 7/8	3.86

Holes for bolts to be drilled 1/16 in. larger than diameter of bolts. T = thickness of pipe = $\frac{\text{diam. of pipe} \times \text{press. in. lbs. per sq. in.}}{8000}$

Pipes to be headed over at edge of flange a distance $= \frac{T}{2}$. Radius $R = \frac{T}{2}$.

Table 26. Composition Flanges and Bulkhead Flanges for Copper Pipes Pressures from 301 to 500 Lbs. per Sq. In. See Figure on p. 1277

Bureau of Steam Engineering Standards

Internal diam. of pipe	Size of flanges										J	Bolts, pipe flange				Bolts, blkhd. flange			
	Dia.	A	B	a	b	C	E	F	G	H		No.	Dia.	P.C.	Pitch chord	No.	Dia.	P.C.	Pitch chord
¼	3 9/16	9/16	3/16	7/16	3/16	1 5/16	3/16	9/16	1/2	6 5/16	3 3/16	3	1/2	2 7/16	2.11	6	1/2	5 5/16	2.69
½	3 13/16	1 1/16	3/16	7/16	3/16	1 1/4	3/16	9/16	1/2	6 7/16	4	4	1/2	2 13/16	1.90	6	1/2	5 5/16	2.84
¾	4 5/16	9/16	3/16	7/16	3/16	1 9/16	3/16	9/16	1/2	7 5/16	4 3/4	4	5/8	3 5/16	2.34	6	1/2	6 5/16	3.19
1	4 15/16	9/16	3/16	7/16	3/16	1 13/16	3/16	9/16	1/2	7 7/16	5	5	5/8	3 9/16	2.09	6	1/2	6 5/16	3.31
1¼	5 3/16	1 5/16	3/16	1	3/16	2 1/16	3/16	1/2	1/2	8 1/16	5 5/8	5	5/8	3 7/16	2.28	8	1/2	7	2.68
1½	5 1/2	1 5/16	1/4	1	3/16	2 7/16	3/16	1/2	1/2	8 1/2	5 5/8	6	5/8	4 5/16	2.09	8	1/2	7 1/4	2.77
2	6 5/8	7/8	1/4	1	3/16	3	3/16	1/2	1/2	9 5/8	6 1/4	6	3/4	5 1/16	2.53	8	1/2	8 3/16	3.20
2½	7 5/16	7/8	5/16	1 1/16	3/16	3 5/8	3/16	9/16	1/2	11	7 3/8	7	3/4	5 3/8	2.44	10	5/8	9 1/2	2.94
3	7 13/16	1 5/16	5/16	1 1/16	3/16	4 1/4	3/16	9/16	1/2	11 5/8	8	8	3/4	6 1/4	2.39	10	5/8	10 1/16	3.13
3½	8 7/16	1	5/16	1 1/16	1/4	4 13/16	3/16	9/16	1/2	12 1/4	8 5/8	9	3/4	6 7/8	2.35	10	5/8	10 5/8	3.32
4	9 7/16	1	3/8	1 1/8	1/4	5 7/16	3/16	5/8	1/2	13 1/4	9 5/8	9	7/8	7 11/16	2.63	12	5/8	11 3/4	3.04
4½	10 1/16	1 1/16	3/8	1 1/8	5/16	6	3/16	5/8	1/2	13 7/8	10 1/4	9	7/8	8 5/16	2.84	12	5/8	12 5/8	3.20
5	11 1/8	1 1/8	3/8	1 1/8	3/8	6 5/8	3/16	5/8	1/2	14 7/8	11 1/4	9	1	9 1/4	3.12	12	5/8	13 3/8	3.46
5½	11 3/4	1 1/8	7/16	1 1/8	7/16	7 1/4	3/16	5/8	1/2	15 5/8	11 7/8	10	1	9 3/4	3.01	14	5/8	14 1/8	3.14
6	12 3/4	1 3/16	7/16	1 3/16	7/16	7 13/16	3/16	1 1/16	1/2	17 1/8	12 7/8	10	1 1/8	10 9/16	3.26	14	5/8	15 3/8	3.42
6½	13 3/8	1 1/4	7/16	1 1/4	7/16	8 7/16	3/16	3/4	1/2	17 3/4	13 1/2	10	1 1/8	11 3/16	3.46	14	5/8	16	3.56
7	13 15/16	1 1/4	1/2	1 1/4	1/2	9 1/16	3/16	3/4	1/2	18 3/8	14 1/8	11	1 1/8	11 3/4	3.31	14	3/4	16 5/8	3.70
7½	14 9/16	1 5/16	1/2	1 5/16	1/2	9 5/8	3/16	3/4	5/16	19	14 3/4	11	1 1/8	12 3/8	3.49	16	3/4	17 1/4	3.36
8	15 5/8	1 3/8	1/2	1 3/8	1/2	10 1/4	3/16	3/4	5/8	20	15 3/4	11	1 1/4	13 5/16	3.72	16	3/4	18 1/4	3.56
8½	16 5/16	1 3/8	5/16	1 3/8	5/16	10 13/16	3/16	3/4	5/8	20 5/8	16 3/8	12	1 1/4	13 3/4	3.56	16	3/4	18 7/8	3.68
9	16 13/16	1 7/16	5/16	1 7/16	5/16	11 7/16	3/16	3/4	1 1/16	21 1/4	17	12	1 1/4	14 5/8	3.72	18	3/4	19 1/2	3.38
9½	17 3/4	1 1/2	5/16	1 1/2	5/16	12	3/16	3/4	3/4	21 3/4	17 1/2	13	1 1/4	14 15/16	3.58	18	3/4	20	3.47
10	18	1 1/2	3/8	1 1/2	3/8	12 5/8	3/16	3/4	3/4	22 3/8	18 1/8	13	1 1/4	15 9/16	3.73	18	3/4	20 1/8	3.58
10½	19 1/16	1 5/16	3/8	1 5/16	3/8	13 1/4	3/16	3/4	1 3/16	23 1/8	19 1/4	13	1 3/8	16 7/16	3.94	18	3/4	21 3/4	3.78
11	19 3/4	1 5/8	3/8	1 5/8	3/8	13 7/8	3/16	3/4	7/8	24	19 3/4	13	1 3/8	17	4.07	18	3/4	22 1/4	3.86
11½	20 1/4	1 5/8	1 1/16	1 5/8	1 1/16	14 7/16	3/16	3/4	7/8	24 3/8	20 1/8	14	1 3/8	17 5/8	3.92	20	3/4	22 5/8	3.58
12	20 13/16	1 11/16	1 1/16	1 11/16	1 1/16	15	3/16	3/4	1 5/16	25 1/4	21	14	1 3/8	18 3/16	4.04	20	3/4	23 1/2	3.67

Table 27. Steel Flanges and Bulk-Pressures From 101 to 300 Lbs. per

Bureau of Steam En-

Internal diam. of pipe	Size of flanges											J
	Dia.	A	B	a	b	C	D	E	F	G	H	
¼	3 7/16	½	7/16	⅞	5/16	1¼	....	1/16	3/8	½	6½	3 5/16
½	3 3/4	9/16	7/16	⅞	⅜	1 7/16	....	1/16	3/8	½	6¾	3 7/16
¾	4	9/16	7/16	⅞	⅜	1 11/16	....	1/16	3/8	½	7	4 1/16
1	4 5/16	⅝	7/16	⅞	5/16	2	....	1/16	3/8	½	7 3/16	4½
1¼	5 1/16	⅝	7/16	⅞	5/16	2¼	....	1/16	3/8	½	8¼	5¼
1½	5 3/8	⅝	½	1 5/16	5/16	2 9/16	....	1/16	7/16	½	8 3/8	5½
2	6 1/16	1 1/16	½	1	5/16	3¼	....	1/16	½	½	9	6¼
2½	6½	¾	½	1 1/16	5/16	3 1/16	....	1/16	9/16	½	10¼	6 5/16
3	7 5/16	1 1/16	½	1 1/16	¼	4¼	4 3/16	1/16	9/16	½	11 3/16	7 3/16
3½	8 1/16	1 1/16	9/16	1 1/16	5/16	4 1/16	4 1/16	1/16	9/16	½	11 7/16	8¼
4	8 11/16	⅞	9/16	1⅛	5/16	5 3/16	5½	1/16	9/16	½	12½	8 7/16
4½	9 1/16	1 5/16	9/16	1⅛	⅜	5 5/16	6 1/16	1/16	9/16	½	13	9 3/16
5	9 13/16	1	9/16	1⅛	7/16	6½	6 3/16	1/16	9/16	½	13 3/16	10
5½	10 1/16	1	⅝	1⅛	½	7 1/16	7¼	1/16	⅝	½	14 1/16	10 1/16
6	11 3/16	1 1/16	⅝	1⅛	9/16	7 5/16	7 7/16	1/16	9/16	½	15 1/16	11 1/16
6½	11 5/16	1 1/16	⅝	1⅛	⅝	8 5/16	8 7/16	1/16	9/16	½	16 1/16	12 1/16
7	12 1/16	1 1/16	⅝	1 1/16	⅝	8 13/16	9 1/16	1/16	11/16	½	17	12 13/16
7½	13 1/16	1 1/16	11/16	1 1/16	11/16	9 5/16	9 5/16	1/16	11/16	½	17 1/16	13 1/16
8	14 1/16	1 1/16	11/16	1 1/16	11/16	9 15/16	10 1/16	1/16	11/16	9/16	18 1/16	14 1/16
8½	14 1/16	1 5/16	11/16	1 5/16	11/16	10½	10 13/16	1/16	11/16	⅝	19 1/16	14 7/16
9	15 1/16	1 3/16	11/16	1 3/16	11/16	11 1/16	11 5/16	1/16	11/16	11/16	19 3/16	15 1/16
9½	15 7/16	1 3/16	¾	1 3/16	¾	11 5/16	11 7/16	1/16	11/16	11/16	20 1/16	16
10	16 7/16	1 7/16	¾	1 7/16	¾	12 1/16	12½	1/16	11/16	¾	20 7/16	16 7/16
10½	17	1½	¾	1½	¾	12 3/16	13 1/16	1/16	¾	¾	21 3/16	17 1/16
11	17 9/16	1½	13/16	1½	13/16	13 5/16	13 5/8	5/16	¾	¾	22	17 3/4
11½	18 9/16	1 9/16	13/16	1 9/16	13/16	13 7/8	14 3/16	5/16	¾	13/16	23	18 3/16
12	19 1/16	1 5/8	13/16	1 5/8	13/16	14 7/16	14 13/16	5/16	¾	⅞	23½	19 1/16
12½	19 3/16	1¾	13/16	1¾	13/16	15 1/16	15 3/16	5/16	¾	1	24 1/16	19 7/16
13	20 1/16	1¾	13/16	1¾	13/16	15 5/16	15 7/8	5/16	¾	1	24 5/16	20 1/2
13½	20 7/16	1 7/8	13/16	1 7/8	13/16	16 1/16	16 1/2	5/16	¾	1 1/16	25 1/16	21
14	21 1/16	1 7/8	1 5/16	1 7/8	13/16	16 3/4	17	5/16	¾	1 1/16	25 1/16	21 5/8
14½	22 1/2	2	1 5/16	2	13/16	17 3/16	17 5/8	5/16	¾	1 1/16	26 5/16	22 5/8
15	23	2	1 5/16	2	13/16	17 7/8	18 1/16	5/16	¾	1 1/16	27 1/16	23 1/16
15½	23 5/16	2	7/8	2	7/8	18 3/16	18 5/8	5/16	¾	1 1/16	28	23 3/4
16	24 1/4	2 1/16	7/8	2 1/16	7/8	19	19 1/16	5/16	¾	1 3/16	28 5/16	24 3/16
16½	24 3/4	2 1/16	7/8	....	....	19 1/2	19 3/4	5/16	....	....	....	....
17	25 3/16	2 1/4	7/8	....	....	20	20 1/4	5/16	....	....	....	....
17½	26	2 5/16	7/8	....	....	20 9/16	20 7/8	5/16	....	....	....	....
18	26 3/16	2 3/8	7/8	....	....	21 1/16	21 3/8	5/16	....	....	....	....
18½	27 3/16	2 7/16	7/8	....	....	21 3/4	22	5/16	....	....	....	....
19	27 7/16	2½	15/16	....	....	22 1/4	22 1/16	5/16	....	....	....	....
19½	29	2 9/16	15/16	....	....	22 7/8	23 1/16	5/16	....	....	....	....
20	29 1/2	2 5/8	15/16	....	....	23 3/8	23 5/8	5/16	....	....	....	....
20½	30 1/4	2 11/16	15/16	....	....	24	24 1/4	5/16	....	....	....	....
21	30 5/8	2¾	15/16	....	....	24 1/2	24 3/4	5/16	....	....	....	....
21½	31 3/8	2 13/16	15/16	....	....	25 1/16	25 1/16	5/16	....	....	....	....
22	31 7/8	2 7/8	15/16	....	....	25 5/8	26	5/16	....	....	....	....
22½	32 1/2	2 15/16	15/16	....	....	26 1/4	26 5/16	5/16	....	....	....	....
23	33 7/8	3	1 5/16	....	....	26 3/4	27 1/16	5/16	....	....	....	....
23½	34 1/2	3 1/16	1 5/16	....	....	27 3/8	27 3/4	5/16	....	....	....	....
24	35	3 1/16	1 5/16	....	....	27 7/8	28 1/16	5/16	....	....	....	....

Holes for bolts to be drilled 1/16 in. larger than diam. of bolts.

T = thickness of pipe = $\frac{\text{diam. of pipe} \times \text{press. in lbs. per sq. in.}}{10{,}000}$

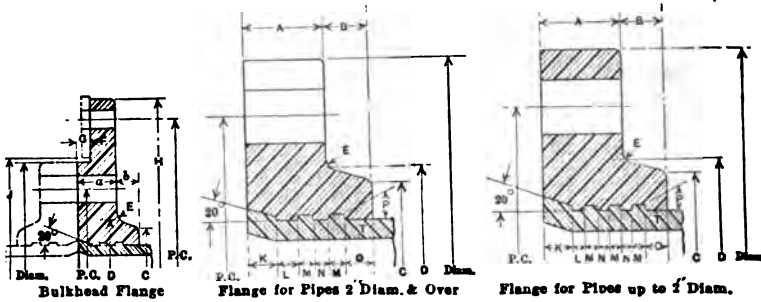
Pipes to be tightly rolled into flanges.

head Flanges for Steel Pipes
Sq. In. See Figure on p. 1284
gineering Standards

Bolts, pipe flange				Bolts, blkhd. flange				Grooves					
No.	Diam.	P.C.	Pitch chord	No.	Diam.	P.C.	Pitch chord	K	L	M	N	O	P
3	½	2 5/16	2.00	6	½	5¼	2.62	5/32	1/16	3/32	5/32	1/8	1/32
3	½	2⅜	2.27	6	½	5½	2.75	5/32	1/16	3/32	5/32	1/16	1/32
4	½	2⅞	2.03	6	½	5¾	2.87	5/32	1/16	3/32	5/32	1/16	1/32
4	½	3 3/16	2.25	6	½	6⅛	3.06	5/32	1/16	3/32	3/16	1/16	1/32
5	⅝	3¾	2.20	8	½	6⅞	2.63	5/32	1/16	3/32	3/16	1/16	1/32
5	⅝	4 1/16	2.39	8	½	7⅛	2.73	5/32	1/16	3/32	7/32	1/16	1/32
6	⅝	4⅝	2.31	8	½	7¾	2.96	¼	1/16	1/16	1/16	1/16	1/32
7	⅝	5 3/16	2.25	10	⅝	8¼	2.70	¼	1/16	1/16	1/16	¼	1/32
8	¾	6	2.30	10	⅝	9⅛	3.05	¼	¼	1/16	1/16	1/16	1/32
8	¾	6 9/16	2.51	10	⅝	10⅜	3.21	5/16	¼	1/16	3/16	¼	1/32
9	¾	7⅛	2.44	12	⅝	11	3.95	5/16	¼	7/32	3/16	¼	1/32
9	¾	7 11/16	2.63	12	⅝	11½	4.13	5/16	5/16	7/32	3/16	¼	3/64
10	¾	8¼	2.55	12	⅝	12⅛	4.35	5/16	5/16	7/32	7/32	9/32	3/64
11	¾	8 13/16	2.48	12	⅝	12⅝	4.46	5/16	5/16	7/32	7/32	11/32	3/64
11	⅞	9⅝	2.71	12	¾	14	5.02	3/8	5/16	7/32	7/32	11/32	3/64
12	⅞	10 1/16	2.64	12	¾	14⅜	5.25	3/8	5/16	¼	7/32	11/32	3/64
12	⅞	10 13/16	2.80	14	¾	15¼	3.39	3/8	5/16	¼	¼	3/8	3/64
12	⅞	11⅜	2.94	14	¾	15¾	3.50	7/16	5/16	¼	¼	5/16	3/64
12	1	12 1/16	3.15	14	¾	16⅞	3.75	7/16	3/8	¼	¼	5/16	3/64
13	1	12⅝	3.05	16	¾	17⅜	3.39	7/16	3/8	¼	¼	7/16	3/64
13	1	13 5/16	3.19	16	¾	18	3.51	½	3/8	¼	¼	7/16	1/16
14	1	13⅞	3.09	16	¾	18½	3.61	½	3/8	9/32	¼	7/16	1/16
15	1	14 7/16	3.00	16	¾	19⅛	3.73	½	3/8	9/32	¼	½	1/16
15	1	15	3.12	18	¾	19⅝	3.41	½	3/8	9/32	9/32	17/32	1/16
15	1	15 9/16	3.24	18	¾	20¼	3.51	½	3/8	9/32	9/32	19/32	1/16
15	1¼	16⅛	3.40	18	¾	21¼	3.69	9/16	3/8	9/32	9/32	19/32	1/16
16	1¼	16 13/16	3.30	18	¾	21¾	3.78	9/16	7/16	9/32	9/32	19/32	1/16
16	1¼	17 5/16	3.43	18	¾	22⅜	3.88	9/16	½	9/32	9/32	21/32	1/16
17	1¼	18¼	3.33	20	¾	23	3.60	9/16	½	9/32	9/32	21/32	1/16
18	1¼	18 11/16	3.24	20	¾	23½	3.67	5/8	½	9/32	9/32	23/32	1/16
18	1¼	19¼	3.34	20	¾	24⅛	3.77	5/8	½	9/32	9/32	23/32	1/16
18	1¼	20 1/16	3.48	20	¾	25⅛	3.93	5/8	½	5/16	5/16	¾	1/16
18	1¼	20 9/16	3.57	22	¾	25⅝	3.65	5/8	½	5/16	5/16	¾	1/16
18	1¼	21⅛	3.67	22	¾	26¼	3.75	5/8	½	5/16	5/16	13/16	1/16
18	1¼	21¾	3.78	22	¾	26⅞	3.86	5/8	½	¾	5/16	13/16	1/16
18	1¼	22¼	3.86	....	....	....	....	5/8	½	⅜	5/16	13/16	1/16
20	1¼	22⅞	3.58	....	....	....	....	11/16	½	⅜	⅜	13/16	1/16
20	1¼	23½	3.68	....	....	....	....	11/16	9/16	⅜	⅜	13/16	1/16
20	1⅜	24⅛	3.77	....	....	....	....	¾	9/16	⅜	⅜	13/16	1/16
20	1⅜	24¾	3.87	....	....	....	....	¾	9/16	7/16	⅜	13/16	1/16
20	1⅜	25⅛	3.95	....	....	....	....	¾	9/16	7/16	7/16	13/16	1/16
20	1½	26⅛	4.09	....	....	....	....	¾	⅝	7/16	7/16	13/16	1/16
20	1½	26⅝	4.16	....	....	....	....	13/16	⅝	7/16	7/16	13/16	1/16
20	1½	27¼	4.26	....	....	....	....	⅞	⅝	7/16	7/16	13/16	1/16
20	1½	27¾	4.34	....	....	....	....	15/16	⅝	7/16	7/16	13/16	1/16
20	1½	28⅛	4.46	....	....	....	....	15/16	11/16	7/16	7/16	13/16	1/16
22	1½	29	4.13	....	....	....	....	15/16	¾	7/16	7/16	13/16	1/16
22	1½	29⅝	4.22	....	....	....	....	1	¾	7/16	7/16	13/16	1/16
22	1⅝	30¼	4.38	....	....	....	....	1 1/16	¾	7/16	7/16	13/16	1/16
22	1⅝	31⅛	4.46	....	....	....	....	1 1/16	¾	7/16	7/16	13/16	1/16
22	1⅝	31⅞	4.54	....	....	....	....	1⅛	13/16	7/16	7/16	13/16	1/16

Table 27. Steel Flanges and Bulkhead Flanges for Steel Pipes Pressures From 101 to 300 Lbs. per Sq. In. (*Continued.*)

Bureau of Steam Engineering Standards



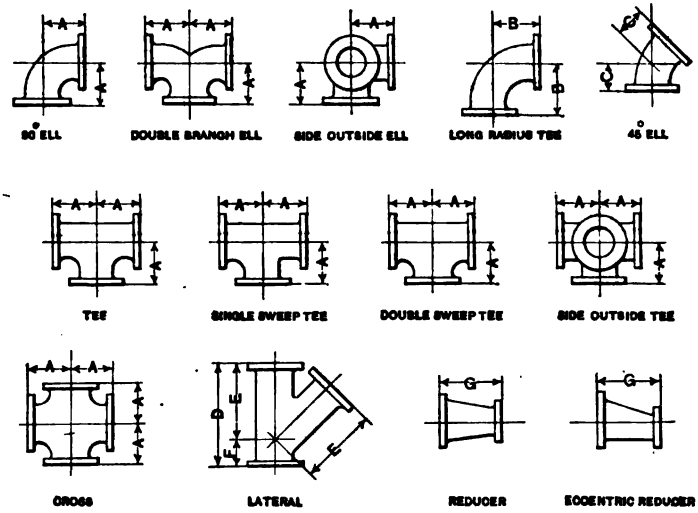
Pipe Fittings

Materials per Bureau of Steam Engineering, U. S. Navy, specifications: Fittings for steel steam pipes, Class B cast steel, or gun metal, see p. 1254; for copper pipes, gun metal unless otherwise directed; for wrought iron or steel pipes for salt water, cast iron or Class C cast steel, galvanized; pipe connections through inner bottom plating, cast iron or Class C cast steel, galvanized; all external fittings on boilers, gun metal unless otherwise directed. Flanges on fittings must conform to Bureau Standard Flange Lists, Tables 23 to 27 inc.

Classification of Fittings per Emergency Fleet Corporation follows the flange practice outlined on p. 1272. American Standard 125 lbs. working pressure fittings, Table 28, are allowed for exhaust steam 14 in. and larger only, and for fresh water, salt water, oil and exhaust steam, iron fittings with pressures not to exceed 125 lbs. and brass with pressures not to exceed 150 lbs. American Standard 250 lbs. working pressure fittings, Table 29, are allowed for fresh and salt water, oil, air and steam, iron fittings with pressures not to exceed 250 lbs. steam or air and 300 lbs. wateM or oil. Forpressuresabove 300 lbs., Hydraulic Standard Fittings, Table 30, are used.

Table 28. American Standard, 125-Lb. Working Pressure

	Size	1	1¼	1½	2	2½	3
A–A	Face to face	7	7½	8	9	10	11
A	Centre to face	3½	3¾	4	4½	5	5½
B	Centre to face of long radius ells	5	5½	6	6½	7	7¾
C	Centre to face of 45-deg. ells	1¾	2	2¼	2½	3	3
D	Face to face laterals	7½	8	9	10½	12	13
E	Centre to face	5¾	6¼	7	8	9½	10
F	Centre to face	1¾	1¾	2	2½	2½	3
G	Face to face reducer	....	....	....	....	....	6
	Diameter of flange	4	4½	5	6	7	7½
	Thickness of flange	7/16	½	9/16	⅝	11/16	¾
	Diameter of bolt circle	3	3⅜	3⅞	4¾	5½	6
	No. of bolts	4	4	4	4	4	4
	Diameter of bolts	7/16	7/16	½	⅝	⅝	⅝
	Minimum metal thickness of body	7/16	7/16	7/16	7/16	7/16	7/16



per Sq. In. Standard Flanged Fittings, Straight Sizes

3½	4	4½	5	6	7	8	9	10	12	14	15	16	18
12	13	14	15	16	17	18	20	22	24	28	29	30	33
6	6½	7	7½	8	8½	9	10	11	12	14	14½	15	16½
8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾	24	26½
3½	4	4	4½	5	5½	5½	6	6½	7½	7½	8	8	8½
14½	15	15½	17	18	20½	22	24	25½	30	33	34½	36½	39
11½	12	12½	13½	14½	16½	17½	19½	20½	24½	27	28½	30	32
3	3	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7
6½	7	7½	8	9	10	11	11½	12	14	16	17	18	19
8½	9	9¼	10	11	12½	13½	15	16	19	21	22¼	23½	25
1¾e	1¾e	1¾e	1¾e	1	1⅛e	1⅛	1⅛	1¾e	1⅛	1⅛	1⅛	1⅛e	1¾e
7	7½	7¾	8½	9½	10¾	11¾	13¼	14¼	17	18¾	20	21¼	22¾
4	8	8	8	8	8	8	12	12	12	12	16	16	16
⅝	⅝	¾	¾	¾	¾	¾	¾	⅞	⅞	1	1	1	1⅛
7⁄16	½	½	½	9⁄16	⅝	⅝	1⅛e	¾	1¾e	⅞	⅞	1	1¼e

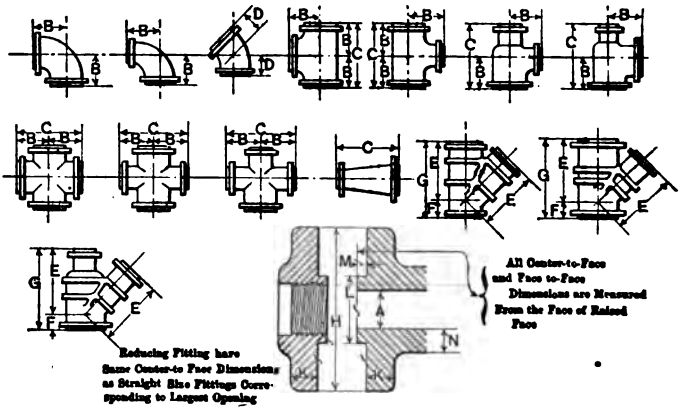
Table 29. American Standard, 250-Lb. Working Pressure

	Size	1	1¼	1½
A–A	Face to face	8	8½	9
A	Centre to face	4	4¼	4½
B	Centre to face of long radius ells	5	5½	6
C	Centre to face of 45-deg. ells	2	2½	2¾
D	Face to face, laterals	8½	9½	11
E	Centre to face, laterals	6½	7¼	8½
F	Centre to face, laterals	2	2¼	2½
G	Face to face, reducer	...	...	...
	Diameter of flange	4½	5	6
	Thickness of flange	1 1/16	3/4	1 3/16
	Diameter of bolt circle	3¼	3¾	4½
	No. of bolts	4	4	4
	Diameter of bolts	½	½	⅝
	Minimum metal thickness of body	½	½	½

per Sq. In., Extra Heavy Flanged Fittings, Straight Sizes

2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14
10	11	12	13	14	15	16	17	18	20	21	23	26	30
5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	13	15
6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½
3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½
11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½
9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31
2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½
.....	.....	6	6½	7	7½	8	9	10	11	11½	12	14	16
6½	7½	8¼	9	10	10½	11	12½	14	15	16¼	17½	20½	23
⅞	1	1⅛	1⅜	1¾	1⅝	1⅜	1⅞	1½	1⅝	1¾	1⅞	2	2⅛
5	5⅛	6⅝	7¼	7⅞	8½	9¼	10⅝	11⅞	13	14	15¼	17¾	20¼
4	4	8	8	8	8	8	12	12	12	12	16	16	20
⅝	¾	¾	¾	¾	¾	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛
½	9/16	9/16	9/16	⅝	⅝	1 1/16	¾	1 3/16	1 3/16	⅞	1 5/16	1	1⅛

Table 30. 800-Lb. Hydraulic American Standard Flanges and Flanged Fittings, 12 In. and Smaller, For Full-weight Wrought Pipe. Semi-steel and Cast Steel


All Center-to-Face and Face to-Face Dimensions are Measured From the Face of Raised Face
Reducing Fitting have Same Center-to Face Dimensions as Straight Size Fittings Corresponding to Largest Opening

800 Lbs. Cold Water Working Pressure—Hydrostatic (no shock).
500 Lbs. Cold Water Working Pressure—Shock.
800 Lbs. Air or Gas Working Pressure—Temperature Not Exceeding 100 Deg. F.

	Size	½	¾	1	1¼	1½	2	2½	3	3½
A	Inside diameter of port.	½	¾	1	1¼	1½	2	2½	3	3½
B	Centre to face, ell, tee, cross	3¼	3¾	4¼	4½	4¾	5¾	6½	7	7½
C	Face to face, tee, cross, reducer	6½	7½	8½	9	9½	11½	13	14	15
D	Centre to face, 45-deg. ell	2	2¼	2¼	2¾	3	4¼	4½	5	5½
E	Centre to face, lateral.	5½	6	6¾	7½	8¾	11	12½	13½	14½
F	Centre to face, lateral.	1¾	2	2¼	2½	2¾	3½	3½	4	4½
G	Face to face, lateral.	7¼	8	9	10	11½	14½	16	17½	19
H	Diameter of flange.	3½	4	4½	5	6	7	7½	8½	9½
K	Thickness of flange									
	Semi-steel	9/16	5/8	11/16	¾	13/16	1¼	1⅜	1½	1⅝
	Cast steel	..	..	..	..	..	1	1⅛	1¼	1⅜
L	Diameter of raised face	1⅜	1 11/16	2	2¼	2⅞	3⅝	4⅛	5	5½
M	Height of raised face.	¼	¼	¼	¼	¼	¼	¼	¼	¼
N	Minimum metal thickness									
	Semi-steel	7/16	7/16	½	½	½	9/16	5/8	5/8	¾
	Cast steel	..	..	..	..	..	½	9/16	9/16	5/8
	Diameter of bolt circle	2⅜	2⅞	3¼	3¾	4½	5¼	5⅞	6½	7½
	Number of bolts	4	4	4	4	4	4	8	8	8
	Diameter of bolts	7/16	½	½	½	5/8	¾	¾	¾	⅞
	Length of bolts									
	Semi-steel	2¼	2½	2½	2¾	3	4	4¼	4½	4¾
	Cast steel	..	..	..	..	..	3½	3¾	4	4¼

Notes for Table 30

Reducing fittings carry same dimensions centre to face as straight-size fittings corresponding to largest opening.

Flanges may be attached to the pipe by any of the following methods: Screw flanges; lap flanges; shrunk, peened or riveted flanges; flanges welded to pipe.

Flanges on fittings and valves, also all companion flanges except those for lap joint, should be furnished with ¼-in. raised face, unless otherwise specified.

Bolt holes are ⅛ in. larger in diameter than bolts. Bolt holes straddle centrelines. Unless otherwise specified, bolt holes in cast-steel fittings should be spot-faced.

Square-head bolts with hexagonal nuts are recommended. Hexagonal nuts on sizes 8 in. and smaller can be conveniently pulled up with open-end wrenches with minimum-design heads. Hexagonal nuts on sizes 9 in. and larger can be conveniently pulled up with box wrenches.

When flanges are screwed, shrunk, peened or riveted on the pipe, it is recommended that the end of the pipe and flange be refaced.

Gaskets extending from the inside of the pipe to the inside edge of the bolts are recommended. The ultimate compressive strength of the gasket must be sufficient to prevent its being crushed when bolts are pulled up.

Where long-radius elbows are desired, the use of pipe bends is recommended.

Table 30. *Continued*

	Size	4	4½	5	6	7	8	9	10	12
A	Inside diameter of port	4	4½	5	6	7	8	9	10	12
B	Centre to face, ell, tee, cross	8½	9	10	11	12	13	14½	15½	16½
C	Face to face, tee, cross, reducer	17	18	20	22	24	26	29	31	33
D	Centre to face, 45-deg. ell	6	6½	7	7½	8	8½	9	9½	10
E	Centre to face, lateral	16½	17½	19½	21	22½	24½	27½	29½	31½
F	Centre to face, lateral	4½	5½	6	6½	6½	7	7½	8	8½
G	Face to face, lateral	21	23	25½	27½	29	31½	35	37½	40
H	Diameter of flange	10¾	11½	13	14	15	16½	18½	20	22
K	Thickness of flange									
	Semi-steel	1⅞	2	2⅛	2¼	2⅜	2½	2⅝	2⅞	3
	Cast steel	1½	1⅝	1¾	1⅞	2	2⅛	2⅜	2½	2⅝
L	Diameter of raised face	6	6½	7¼	8⅛	9⅛	10⅝	11⅝	12¾	15¼
M	Height of raised face	¼	¼	¼	¼	¼	¼	¼	¼	¼
N	Minimum metal thickness									
	Semi-steel	⅞	1	1⅛	1⅛	1¼	1⅜	1½	1⅝	1¾
	Cast steel	1 1/16	¾	1 3/16	⅞	1	1 1/16	1⅛	1¼	1⅜
	Diameter of bolt circle	8½	9¼	10½	11½	12½	13⅝	15½	17	19¼
	Number of bolts	8	8	8	12	12	12	16	16	20
	Diameter of bolts	⅞	⅞	1	1	1	1⅛	1⅛	1¼	1¼
	Length of bolts									
	Semi-steel	5¼	5½	6	6¼	6½	7	7½	7¾	8
	Cast steel	4½	4¾	5¼	5½	5¾	6¼	6¾	7	7¼

Painting Pipes; Identification Colors. The body color of pipes should be the same as that of the adjacent bulkheads and decks. There is at present no recognised standard method of painting piping. To insure identification the U. S. Navy has adopted the standard method of painting flanges and

Colors for Pipes

Bureau of Steam Engineering Standards

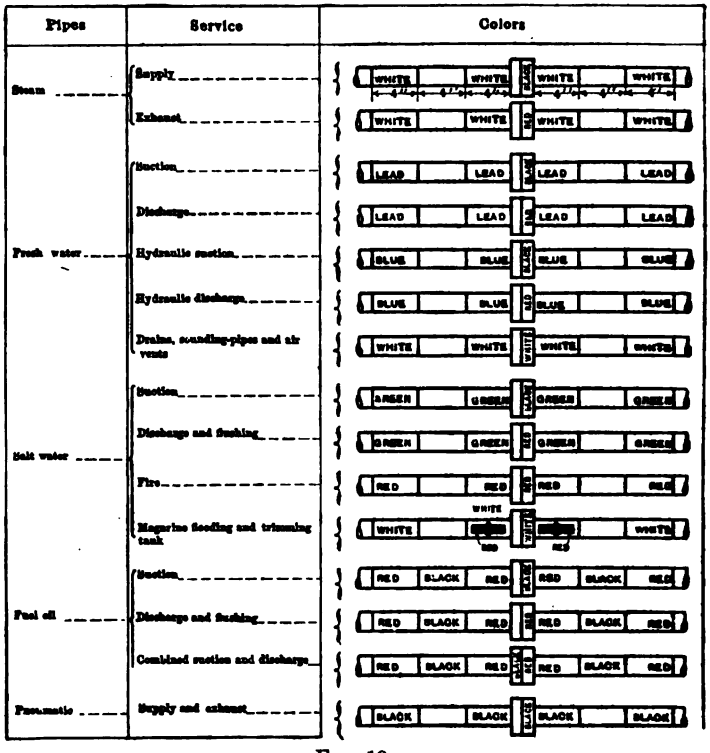


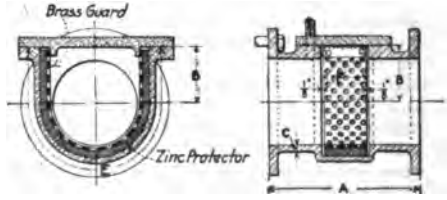
FIG. 10.

piping shown in Fig. 10. This method is recommended for general adoption in Marine Service. In addition to stripes on and near the flanges, pipes are striped at suitable intervals, so that their course may be easily followed. Wherever a pipe passes through a bulkhead, a band of its distinctive color is painted close to, and on each side of bulkhead.

Table 31. Zinc Boxes

(Proposed Standard)

Low pressures up to 100 lbs.						High pressures above 100 lbs.			
Diameter of pipe	Face to face	Center to face of cover	Thickness	Width of zinc	Length of zinc	Diameter of pipe	Face to face	Center to face of cover	Thickness
	A	B	C	F	E		A	B	C
1	7¼	1⅛	3/16	2	5	1	7½	1⅛	3/16
1¼	7½	1¼	3/16	2	5½	1¼	8¼	1¼	3/16
1½	7½	1⅜	3/16	2	6¼	1½	8¼	1⅜	3/16
2	7½	1⅝	3/16	2	7½	2	8¼	1⅝	3/16
2½	7¾	1⅞	3/16	2	8¾	2½	8½	1⅞	3/16
3	8½	2¼	3/16	2	10¼	3	9¼	2¼	3/16
3½	8½	2½	3/16	2	11½	3½	9½	2½	3/16
4	9½	2¾	3/16	3	13	4	10¼	2¾	3/16
4½	9¾	3	3/16	3	14¼	4½	10½	3	3/16
5	9¾	3¼	3/16	3	15⅝	5	10½	3⅜	3/16
5½	9¾	3½	¼	3	17	5½	10½	3⅝	¼
6	10	3¾	¼	3	18¼	6	11¼	4	¼
6½	10	4	¼	3	19½	6½	11¼	4¼	¼
7	10¾	4¼	¼	3	20¾	7	11½	4⅝	¼
7½	10¾	4⅜	¼	3	22⅛	7½	11½	4⅞	¼
8	11	5	¼	3	23⅞	8	11¾	5⅛	¼
8½	11	5¼	¼	3	25⅛	8½	11¾	5⅜	¼
9	11¼	5½	¼	3	26½	9	12	5⅝	¼
9½	11¼	5¾	¼	3	27¾	9½	12	5⅞	¼
10	11½	6	¼	3	29	10	12	6⅛	¼
10½	11½	6¼	¼	3	30¼				
11	11¾	6½	¼	3	31½				
11½	12¼	6¾	¼	3	32⅞				
12	12¼	7	¼	3	34⅛				
12½	12¼	7¼	¼	3	35½				
13	12¼	7½	¼	3	36¾				
13½	12½	7	¼	3	38				
14	12½	8	¼	3	39½				
14½	12¾	8¼	¼	3	40¾				
15	12¾	8¼	¼	3	41⅞				



NOTE

Protectors to be of rolled zinc ½ inch thick for all pipe sizes.
Guards to be of brass, 120 mils thick for pipes 4 inches diameter and larger, and 83 mils thick for pipes less than 4 inches diameter.
Guards to be punched with holes 5/16 inch diameter and not more than ½ inch pitch.
Flanges made to suit connecting pipes.
Studs in cover not to penetrate water side of box.
Diameter of cover studs = diameter of pipe flange bolts.

VALVES

Previous to 1914 it was the practice of most ship yards to design and make nearly all of the important valves used aboard ships. These valves were based upon the experience of the shipbuilder and the operating force on the ship. When the shipbuilding industry began to expand, it became necessary to conserve man power and machinery. The older yards naturally began to look for assistance in relation to various parts of ship equipment which could be purchased on the open market. New yards had no such equipment. As a result of war time conditions, it became necessary for the valve manu-

facturers to expand their lines of Marine material and prepare to furnish practically every kind of valve or fitting required on ship board. As in many other lines of effort the tendency is toward standardisation. Valves and fittings, 3½ in. and larger, must have flanged ends. Valves and fittings, 3 in. and smaller, except safety valves, may have screwed or flanged ends.

General Classification per Emergency Fleet Corporation Instructions. Low pressure valves and fittings may be used for exhaust steam only in sizes not less than 14 in. Standard valves and fittings may be used for fresh and salt water, oil, and exhaust steam, under max. press. of 125 lbs. for iron valves, and 150 lbs. for brass valves. Extra heavy valves and fittings may be used for fresh and salt water, oil, air and steam under max. press. of 250 lbs. for steam and air and 300 lbs. for water and oil. Above these pressures special high pressure valves and fittings must be used. When connected directly to the boiler shell or drum, all valves and fittings must have flanged ends and sizes 2 in. and smaller must be made of brass, hard metal or steel. For all other connections, valves and fittings, 3½ in. and larger, must have flanged ends. Valves and fittings, 3 in. and smaller, except safety valves, may have screwed or flanged ends.

Maximum Temperatures per Emergency Fleet Corporation Instructions. Valves and fittings of brass, cast iron, semi-steel or ferro-steel may be used on temperatures not to exceed 407 deg. F. Valves and fittings of hard metal may be used on temperatures not to exceed 550 deg. F. Valves and fittings of cast or forged steel, or of a composition with a tensile strength of at least 50,000 lbs. and with an elongation of at least 10 per cent in 2 in. may be used for temperatures not to exceed 800 deg. F.

Design and Manufacture per Emergency Fleet Corporation Instructions must be such as to withstand excessive pipe strains and vibrations. Valves and flanged fittings, all sizes, must have the trade mark or name of the manufacturer and the working pressure they are guaranteed to stand, cast or stamped on them. Screwed fittings do not require working pressure marking. All sizes of iron and steel body valves and all brass valves 2½ in. and larger must have bolted bonnets. When bolted glands are used on valves, they may be cast of brass in one piece or may consist of a brass gland screwed into a malleable iron or steel flange. Glands shall be held in place by a brass stud and brass or steel hexagon nut. Iron and steel body valves must have non-ferrous seat rings. Steel body valves for steam must have nickel composition seat rings. All valves should be so made that they can be repacked while open and under pressure. All valve stuffing boxes should be packed with high grade asbestos ring packing, by the manufacturer. All bonnet joints, Standard and Low pressure, will be packed with rubber sheet packing. All bonnet joints, Extra Heavy and High Pressure, will be packed with compressed sheet asbestos composition.

General Requirements per Bureau Steam Engineering, U. S. Navy. (1) Valves of approved pattern will be supplied wherever necessary to complete the various pipe systems, whether specified or not; (2) the size of a valve, when given in these specifications, refers to the diameter of the equivalent clear opening. This does not apply to regulating valves, in which the size of opening may be decreased below the nominal opening, as approved; (3) all valves will have their flanges finished on faces, and when flanges are not lagged the edges will be finished. Such valves as are not finished all over will have the backs of flanges spot faced; (4) all cocks and valves on boiler drums, unless fitted on pads, or in other approved manner, will have spigots, or nipples, passing through the boiler plates; (5) all valve stems will be so threaded as to turn right-handed to close; (6) the valve stems of all steam valve larger than ¾ in. and of all other valves larger than 1 in. will have outside squares or "Acme" threads and the stems fitted with packed

glands; (7) balanced valves must not rotate with stems and will open and close by direct pull and thrust; (8) the stems of valves should not point below the horisontal except where required for convenience in working from below. Where stems of valves in steam lines point below the horisontal, the stem will be made of rolled monel or other approved material; (9) all valves and cocks underneath the floor plates will have their wheels or handles easily accessible; (10) where cross-connection pipes have valves at bulkheads, the valves will be arranged to be operated from both sides of the bulkhead.

Materials per Emergency Fleet Corporation Instructions. All castings must be sound, clean, free from blow holes, porous places, cracks, or any other defects which will materially affect their strength or appearance. Cast iron must have a min. tensile strength of 22,000 lbs. and a min. transverse strength of 2500 lbs. Semi-steel or ferro-steel must have a min. tensile strength of 30,000 lbs. and a min. transverse strength of 3000 lbs. Malleable iron must have a min. tensile strength of 30,000 lbs. Cast steel must conform to the following requirements: Min. tensile strength, 50,000 lbs.; min. elastic limit, 22,500 lbs.; min. elongation in 2 in., 15 per cent. Brass must have a min. tensile strength of 30,000 lbs. and must contain not more than 2 per cent lead. Hard metal should be of the following composition: Copper, 87 to 89 per cent; tin, 9 to 11 per cent; zinc, the remainder. In the manufacture of valves, fittings and accessories for Marine Work, materials complying with foregoing specifications should be used. Where necessary, for extremely high temperature or for special purposes, other materials than described, but equal in quality may be substituted.

Materials per U. S. Naval Practice (the materials here given must conform to U. S. Naval Specifications; see Section 3): All valves, cocks, and their fittings, except as otherwise specified and approved will be composition or cast steel. Composition will be used for valves as follows: Gun metal, 4 in. diam. and above; Naval brass, below 4 in. diam. Valve stems for composition valves will be made of rolled Naval brass, rolled phosphor bronze or rolled manganese bronze. Cast steel valves will have bodies of class B, cast steel; valves and glands of gun metal or Naval bronze; valve seats of nickel; valve stems of class B forgings. Handwheels for all valves will be of cast Naval brass. All valves in bilge must have composition fittings.

General Practice. *A. G. Christie, in the Jour. A.S.M.E., Feb.*, 1919, says of **"Marine Practice in Valves and Fittings:"** "Steel valves are used for superheat. Some shipbuilders even specify steel valves for all boiler press. of 200 lbs. or greater. If superheat is not used the valves may be extra heavy, of cast iron or semi-steel. These materials are also used for auxiliary steam lines. Bronze valves are sometimes employed in the smaller sizes, but these are very much more expensive than the extra heavy cast iron valves and in sizes of 2½ in. and over have no advantage over them except in lightness of weight.

Valves 2½ in. and above must have bolted bonnets to comply with the U. S. Steamboat Inspection Rules. The marine or crosshead type of yoke is much superior to the standard cast yoke for ship purposes. It weighs less and is generally stronger and more readily repaired. It costs very little more than the cast yoke, but is not "standard" with many builders. The latter yoke can be safely used for all low-pressure services.

It is customary on board ship to use the screwed-union bonnet on 2 in. and smaller valves. This type of valve is generally more expensive than the ordinary bonnet type, but has many points of superiority such as its regrinding facilities, longer life and less liability to damage when dismantling. Valve manufacturers generally make the screwed-union bonnet valves in two weights, one to withstand 200 lbs. press., the other for 350 lbs. press. It is therefore necessary to use the 200-lb. valves for many low-pressure services. This has considerable advantages, for fewer types of valves need be bought and their interchangeability is increased.

Cast-iron or semi-steel valves for marine service are usually required to have a solid bronze disk with bronze seat. The stem may be of rolled phosphor bronze, rolled naval brass or manganese bronze. Gland studs and nuts should also be of bronze. Steel valves for superheat generally have monel-metal disks and seats with bronze or nickel-plated steel stems. The bronze generally used for valve bodies, disks, seats, glands, etc.,

Table 29. American Standard, 250-Lb. Working Pressure

	Size	1	1¼	1½
A–A	Face to face	8	8½	9
A	Centre to face	4	4¼	4½
B	Centre to face of long radius ells	5	5½	6
C	Centre to face of 45-deg. ells	2	2½	2¾
D	Face to face, laterals	8½	9½	11
E	Centre to face, laterals	6½	7¼	8½
F	Centre to face, laterals	2	2¼	2½
G	Face to face, reducer	...	...	...
	Diameter of flange	4½	5	6
	Thickness of flange	1 1/16	3/16	1 3/16
	Diameter of bolt circle	3¼	3¾	4½
	No. of bolts	4	4	4
	Diameter of bolts	½	½	⅝
	Minimum metal thickness of body	½	½	½

per Sq. In., Extra Heavy Flanged Fittings, Straight Sizes

2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14
10	11	12	13	14	15	16	17	18	20	21	23	26	30
5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	13	15
6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½
3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½
11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½
9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31
2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½
.....	.....	6	6½	7	7½	8	9	10	11	11½	12	14	16
6½	7½	8¼	9	10	10½	11	12½	14	15	16¼	17½	20½	23
⅞	1	1⅛	1 5/16	1¼	1 5/16	1⅜	1 7/16	1½	1⅝	1¾	1⅞	2	2⅛
5	5⅞	6⅝	7¼	7⅞	8½	9¼	10⅝	11⅞	13	14	15¼	17¾	20¼
4	4	8	8	8	8	8	12	12	12	12	16	16	20
⅝	¾	¾	¾	⅝	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛
½	9/16	9/16	9/16	⅝	⅝	1 1/16	¾	1 3/16	1 3/16	⅝	1 5/16	1	1⅛

without springs. When installed in any other position valves must be provided with a light spring to assist in holding the disk in place. Horizontal and angle check valves must have upper and lower disk guide and must conform to all applicable requirements for globe and angle valves. Swing

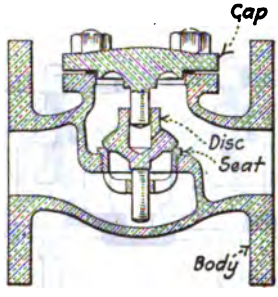


FIG. 19.

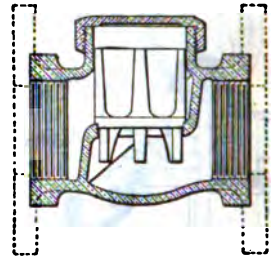


FIG. 20.

check valve disks must be one piece, non-ferrous casting. All swing check valves must have brass hinge, and rolled brass hinge pin, supported in the body. These checks may be used in horizontal or vertical up-flow lines. The common type of check valve used on ship board is called the "**Screw Down Check**" which is merely a stop valve with a sliding connection between

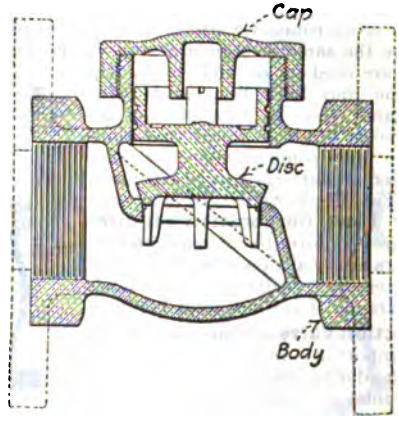


FIG. 21.

the stem and the disk, so that the lift may be controlled, Fig. 13. In some instances where this type is not desired, the regular **horizontal bolted bonnet check** such as shown in Fig. 19 is used. (See Table 44.) These valves are made similar to bolted bonnet stop valves except that a cap is used instead of the yoke and stem. **Horizontal screwed bonnet check**

valve is shown in Fig. 20 and Table 45. In the small size brass check valves where very severe service is encountered, the **union bonnet dash pot check,** such as shown in Fig. 21 and Table 46, gives very good results.

Standard iron body, Merchant Marine swing check valve is shown in Fig. 22 and Table 47; **medium brass swing check valve** is shown in Fig. 23 and Table 48; **Extra heavy brass swing check valve** is made to a similar pattern, Table 49.

Gate Valves

Gate Valves should be used only where service requires valve to be wide open or shut; they should not be used where throttling is necessary. Rising or Non-rising Stem Valves may be used. Sizes 8 in. and larger, when used for pressures exceeding 150 lbs. must be provided with by-pass, not smaller than 1½ in. diam. The stem threads in iron or steel body gate valves should work in brass. Where yoke or crosshead is not brass it should be bushed for

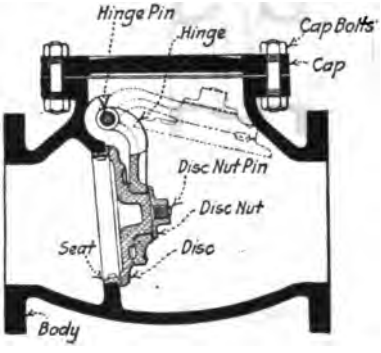


FIG. 22.

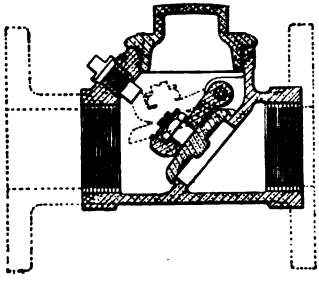


FIG. 23.

the stem. All sizes should have the disc cast in non-ferrous metal except exhaust steam valves, which may have brass seat rings rolled or screwed into iron disks. For exhaust steam, a low pressure gate valve is commonly used. The regular commercial low pressure valves are not satisfactory, as they will not withstand the vibration, and, therefore, a ribbed body is used or a **standard pattern** throughout. These valves are generally of the **rising stem type,** Fig. 24 and Table 50. These valves in the **non-rising stem pattern,** Fig. 25, Table 50, are sometimes used for main injection service.

Although gate valves are not as commonly used on ship board as on land, they are coming more into use daily. Standard pattern gate valves with solid brass disks, brass glands, gland studs and nuts, and brass stems, are used on exhaust steam lines and water and oil lines.

Gate valves are more compact than and offer less obstruction to flow than do globe valves of equal size; there is no change of direction of fluid in the valves. It is more difficult to keep gate valves tight.

Standard brass gate valves, Fig. 26, Table 51, are used on oil and water lines and are made with screwed or bolted bonnet, depending on the size. They are also made with male hose threads on the outlet and with cap and chain attached.

Brass, hose gate valves are shown in Fig. 27, Table 52.

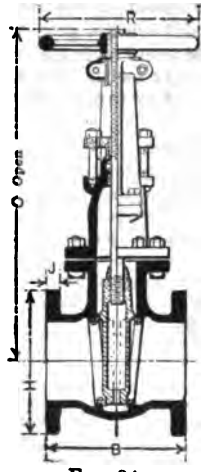


FIG. 24.

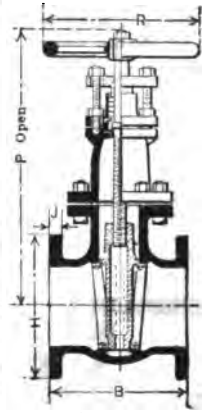


FIG. 25.

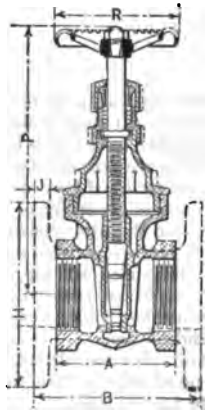


FIG. 26.

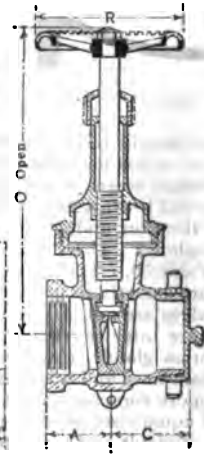


FIG. 27.

Bureau of Steam Engineering, U. S. Navy, Requirements for Gate Valves. (1) Gate valves will not be used on steam or exhaust pipes except where approved; (2) all valves in feed suction and discharge pipes, except boiler or other check valves, will be gate valves of approved design unless otherwise directed; (3) all gate valves for use with salt water, and fitted with non-rising valve stems, will be so made that the stems are not exposed to the water passing through the valves; (4) all gate valves 1½ in. diam. and above will have removable seats; (5) all gate valves above 5 in. in diam. subjected to a press. above 50 lbs. per sq. in. will have by-pass valves to relieve the valves when jammed on their seats, except where these valves are fitted in suction lines. For gate valves above 5 and to 8 in. in diam. the by-pass valves will be 1 in. in diam., from 8 to 12 in. 1½ in. in diam., and above this, 2 in. in diam.; (6) all gate valves will be fitted with indices to show whether they are open or closed.

Cocks and Blow-off Valves

When used for **boiler blow-off** service or for pressures above 150 lbs., cocks should have the plug held into body with a guard or gland. Boiler blow-off valves should be of special design to meet the severe service to which they are subjected; see p. 449. **Pet cocks** for press. of 150 lbs. and up should be extra heavy, with ¼-in. male pipe thread inlet. When subjected to temperatures above 100 deg. F., iron body cocks must have iron **plugs.** When used for cold water, iron body cocks must have brass plugs. When used for oil, iron body cocks may have brass or iron plugs. When used for water all bearing surfaces must be brass to brass or brass to iron. A good standard iron stop cock is shown in Fig. 28 and Table 53.

FIG. 28.

Brass blow-off valves are generally used for **surface blow** and often for **bottom blow.** A valve with a self-cleaning, non-wire drawing seat will give the best service. It is very desirable that the seat and disk be made of an extremely hard metal. Valves of this design can be successfully used for throttling service, boiler feed, etc.

For **boiler blow-down valves** see p. 449.

For **boiler try-cocks** see p. 447.

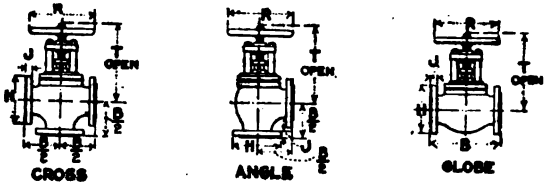
U. S. Naval Specifications allow use of cocks to a limited extent only and only in places where valves cannot be satisfactorily fitted; all cocks communicating with vacuum spaces must have bottoms of shell cast in and packed plugs. All cocks over 1 in. in diam. must have packed plugs. Plug cocks will not be used on pipes over 2 in. in diam.

Handwheels and Wrenches per U. S. Naval Specifications. (1) Valves and cocks will have, where directed, removable box or socket wrenches in lieu of handwheels or permanent handles. These wrenches will be so fitted that they can be removed only when the valves are closed, and they will be marked and stowed in convenient racks; (2) all handwheels conspicuously located will be finished all over; other handwheels, unless otherwise directed, will have rims and faces of bosses finished; (3) all handwheels will be at least 1½ times as great in diameter as their valves.

For dimensions of handwheels, see Table 54, p. 1323.

Table 32. Dimensions of Cast Steel Merchant Marine Globe, Angle and Cross Valves, Crosshead Type

(See Fig. 11. Crane Co.)

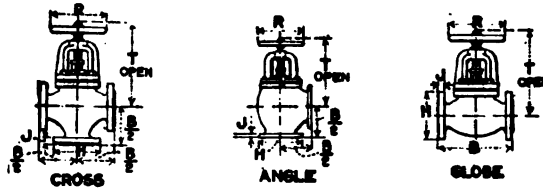


Cast steel body and crosshead; cold rolled monel or rolled steel stem; monel seat; hard metal or monel disk.

Size.....	1½	2	2½	3	3½	4	4½	5	6	7	8
B	9½	10½	11½	12½	13¼	14	15	15¾	17½	19¼	21
$\frac{B}{2}$	4¾	5¼	5¾	6¼	6⅝	7	7½	7⅞	8¾	9⅝	10½
H	6	6½	7½	8¼	9	10	10½	11	12½	14	15
J	1 3/16	⅞	1	1⅛	1 5/16	1¼	1 5/16	1⅜	1 7/16	1½	1⅝
R	7½	7½	9	10	10	12	12	14	16	18	20
T	13	13¾	14½	17½	17½	19½	19½	21½	25	26¼	29½

Table 33. Dimensions of Extra Heavy, Ferro-steel Merchant Marine Globe, Angle and Cross Valves

(See Fig. 12. Crane Co.)

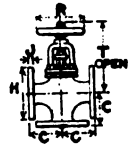


Ferro-steel body and yoke; manganese bronze or steel stem; hard metal disk and seat.

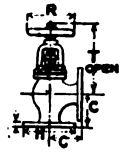
Size.....	1½	2	2½	3	3½	4	4½	5	6	7	8
B	9½	10½	11½	12½	13¼	14	15	15¾	17½	19¼	21
$\frac{B}{2}$	4¾	5¼	5¾	6¼	6⅝	7	7½	7⅞	8¾	9⅝	10½
H	6	6½	7½	8¼	9	10	10½	11	12½	14	15
J	1 3/16	⅞	1	1⅛	1 5/16	1¼	1 5/16	1⅜	1 7/16	1½	1⅝
R	7½	7½	9	10	10	12	12	14	16	18	20
T	13	13¾	14½	17½	17½	19½	19½	21½	25	26¼	29½

Table 34. Dimensions of Medium, Brass, Merchant Marine Stop and Screw Down Check Valves, Flanged and Screwed Types

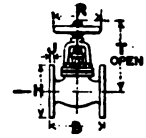
(See Fig. 13. Crane Co.)



CROSS



ANGLE

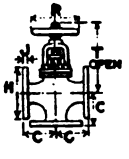


GLOBE

Size	1½	2	2½	3
B	5⅞	6⅞	8	9⅛
C	3⅛	3⅜	4¼	4⅝
H	5	6	7	7½
J	7/16	½	9/16	⅝
R	5	5½	6½	7½
T	9	9½	10¾	12
Centre to End Screwed	2 1/16	3	3⅜	4⅛

Table 35. Dimensions of Extra Heavy, Brass, Merchant Marine Stop and Screw Down Check Valves, Flanged and Screw Types

(See Fig. 13. Crane Co.)



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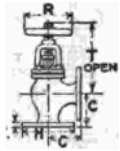
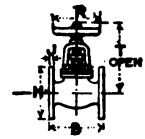


ANGLE

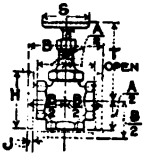


GLOBE

Brass body and yoke; hard metal disk and seat.

Size	1½	2	2½	3
B	6⅜	7⅜	8¾	10
C	3¾	4⅛	4⅝	5¼
H	6	6½	7½	8¼
J	9/16	⅝	1 1/16	¾
R	5½	6½	7½	9
T	9	10⅛	11½	12⅞
Centre to End Screwed	2¾	3⅛	3¾	4¼

Table 36. Dimensions of Extra Heavy, Brass, Globe, Angle and Cross Valves, Flanged and Screwed Type
(See Fig. 14. Crane Co.)



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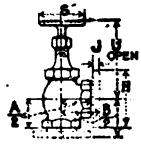
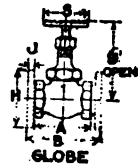


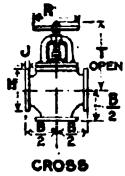
ANGLE



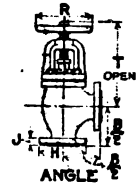
GLOBE

Size	¼	⅜	½	¾	1	1¼	1½	2
A	1¾	2	2 15/32	2 15/16	3½	4 1/16	4 5/16	5¾
A/2	⅞	1	1 15/64	1 15/32	1¾	2 1/32	2 5/16	2⅞
B	....	....	....	3¾	4⅜	4 13/16	5½	6½
B/2	....	....	....	2 5/16	2⅞	3⅛	3¾	4 1/16
H	....	....	....	4	4½	5	6	6½
J	....	....	....	7/16	½	17/32	9/16	⅝
S	1¾	2⅛	2 5/16	2¾	3⅛	3⅝	4⅛	4¾
U	3½	4	4½	5 1/16	6 5/16	7 1/16	8	8¾

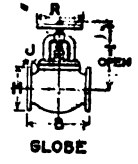
Table 37. Dimensions of Standard, Iron Body, Marshall Marine, Stop and Screw Down Check Valves, Flanged and Screw Types
(See Fig. 15. Crane Co.)



CROSS



ANGLE

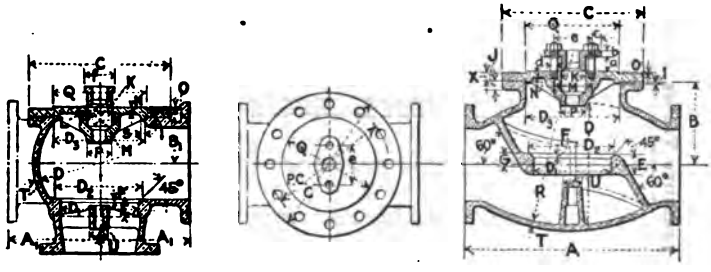


GLOBE

Iron body and yoke; brass disk, gland, studs and nuts.

Size	2	2½	3	3½	4	4½	5	6	7	8	10	12
B	8	8½	9½	10½	11½	12	13	14	16	17	20	24
B/2	4	4¼	4¾	5¼	5¾	6	6½	7	8	8½	10	12
H	6	7	7½	8½	9	9¼	10	11	12½	13½	16	19
J	⅝	1 1/16	¾	1 3/16	1 5/16	1 5/16	1 5/16	1	1 1/16	1 1/8	1 3/16	1 1/4
R	6½	6½	7½	7½	9	9	10	12	14	16	18	20
T	11⅛	11¾	13⅛	13⅞	15⅜	15½	17¼	19	21¾	23¾	27¾	32½
Centre to End Screwed	3¼	3½	4									

Table 38. Bureau of Steam Engineering Standard Low-pressure Valves



Limit of working pressure, 100 pounds.
All material to be composition G or M, except where otherwise specified.
Studs for cover Class B bolt material.
Studs for gland, nuts for gland studs, and nuts for cover studs, composition P-r, N-r, or M-r.
Dividing wall conical for valves 5½ inches and above, 5 inches and below straight.
Valves will be cast without removable seats.
Flange shown in dot and dash on cross valve only.
Valves 1½ inches diameter and below have cover and yoke one piece.
Flanges to conform to Bureau of Steam Engineering standard flange list.

D_2 = diameter of false seat.
G = depth of false seat.

P.C. = diameter of pitch circle for cover.

g = number of studs in cover.
h = diameter of studs in cover.

i = number of ribs in cover.

j = thickness of ribs in cover.

k = number of ribs under valve seat.

m = thickness of ribs under valve seat.

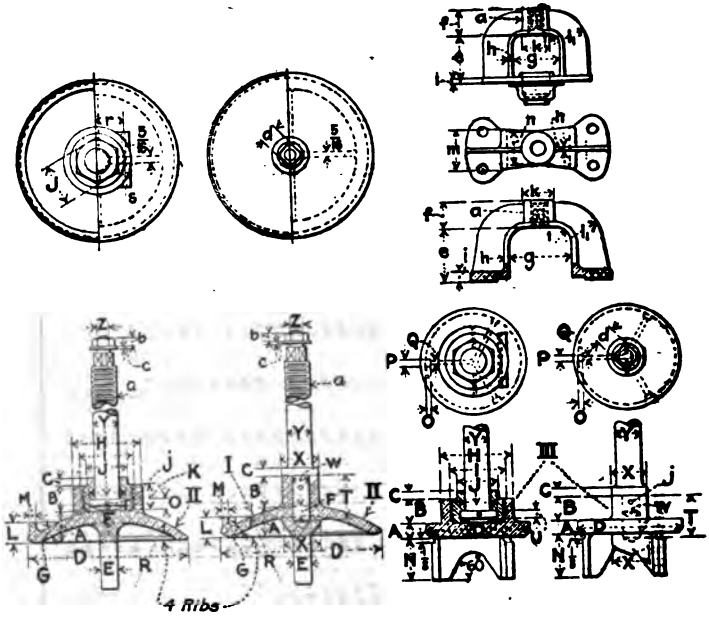
Table 38. Bureau of Steam Engineering Standard Low-pressure Valves.—(*Continued*)

Size	A	B	A_1	B_1	C	D	D_1	D_2	D_3	E	F	G	I	J	K	L	M	N	O
½	6¼	1¼	3⅛	1 3/16	3⅝	2	¾	....	1⅝	⅜	1/16	....	⅛	½	⅜	9/16	...	....	...
¾	7	1½	3½	1 5/16	4	2¼	1	....	1¾	⅜	1/16	....	⅛	½	⅝	⅝	...	....	...
1	7⅛	1¾	3¾	1	4⅛	2¾	1⅜	....	2⅛	½	1/16	....	⅛	½	1	⅞	1⅜	....	...
1¼	8¼	2⅛	4¼	1 3/16	4½	3	1⅝	....	2¼	⅝	1/16	....	⅛	½	1	⅞	1⅜	....	...
1½	8¾	2½	4⅝	1⅜	4¾	3½	1¾	....	2⅜	¾	1/16	....	⅛	½	1⅛	1	1⅜	....	...
2	10¼	2⅞	5⅛	1⅜	5½	4	2¼	....	2¾	⅝	1/16	....	⅛	½	1⅜	1⅛	1⅞	5/16	1/16
2½	11½	3⅜	5¾	1⅞	6⅛	4¾	2⅞	....	3⅜	⅞	1/16	....	⅛	9/16	1½	1¼	2	5/16	1/16
3	12½	3¾	6¼	2⅛	7⅛	5½	3⅜	....	4	⅞	⅛	....	5/16	9/16	1½	1¼	2	5/16	1/16
3½	13	4¼	6½	2 7/16	7¾	6¾	3⅝	....	4½	1	⅛	....	5/16	⅝	1⅝	1⅜	2¼	5/16	1/16
4	13¾	4⅝	6⅞	2 11/16	8⅜	7½	4½	....	5⅜	1	⅛	....	5/16	⅝	1⅝	1⅜	2¼	⅜	1/16
4½	15	5¼	7½	3	9¼	8	5	5¾	6	1⅛	⅛	¾	5/16	11/16	1⅝	1½	2⅝	⅜	1/16
5	16	5¾	8	3¼	9⅝	8¾	5½	6¼	6⅝	1⅛	⅛	⅞	5/16	¾	1¾	1½	2⅝	⅜	1/16
5½	16¾	6¼	8⅜	3½	10¼	9½	6	6⅞	7	1¼	⅛	1 1/16	5/16	¾	1⅞	1½	2½	⅜	1/16
6	18	6¾	9	3¾	11	10¾	6½	7⅝	7½	1¼	⅛	1	¼	¾	1⅞	1⅝	2½	7/16	1/16
6½	18¾	7⅛	9⅜	4⅛	11¾	11	7⅛	8	8¼	1⅜	⅛	1 1/16	¼	⅜	2	1⅝	2⅝	7/16	1/16
7	20	8	9⅝	4⅜	13¼	11⅞	7¾	8⅝	9	1½	⅛	1⅛	¼	⅞	2	1⅝	2⅝	7/16	1/16
7½	21½	8⅝	10	4⅝	13½	12⅝	8¼	9⅛	9¼	1½	⅛	1¼	¼	⅞	2½	1⅝	3⅛	7/16	1/16
8	22¾	9⅛	10⅝	5	14¼	13½	9	9⅞	10	1⅝	⅛	1 5/16	¼	⅞	2½	1¾	3⅛	½	1/16
8½	24	9⅝	10¾	5¼	14⅞	14¼	9½	10½	10⅝	1¾	⅛	1 5/16	¼	⅞	2⅝	1¾	3⅝	½	1/16
9	25½	10⅛	11¼	5½	15⅝	15	10	11	11⅛	1⅞	⅛	1⅜	¼	1 1/16	2¼	1⅞	3⅝	½	1/16
9½	26¾	10⅝	11⅝	5⅞	16	15¾	10½	11½	11¾	1⅞	⅛	1 7/16	¼	1 1/16	2¾	2	3¾	½	1/16
10	28	11⅛	12	6⅛	16⅝	16½	11	12	12⅛	2	⅛	1½	¼	1 1/16	2⅞	2	3¾	⅝	1/16
10½	29¼	11¾	12½	6⅜	17⅛	17¼	11½	12½	12⅞	2	⅛	1½	¼	1 1/16	2⅞	2	3⅞	⅝	1/16
11	30½	12⅛	13	6⅝	17¾	18	12	13	13¼	2	⅛	1⅝	¼	1	2⅞	2⅛	3⅞	⅝	1/16
11½	31¾	12⅝	13¼	6⅞	19	18¾	12½	13⅝	13¾	2¼	⅛	1⅝	¼	1	3	2¼	4	⅝	1/16
12	33	13	13⅝	7⅛	19⅝	19½	13½	14¼	14⅜	2¼	⅛	1⅝	¼	1	3	2½	4	11/16	1/16
12½	34½	13⅝	14⅛	7½	20¼	20¼	13⅝	14¾	14⅞	2⅛	⅛	1⅝	¼	1	3 1/16	2½	4⅛	11/16	1/16
13	36	13¾	14⅜	7¾	20⅝	21	14⅛	15¼	15⅜	2⅛	⅛	1¾	¼	1	3⅛	2¾	4⅛	11/16	1/16
13½	37½	14⅛	15¼	8	21⅛	21¾	14⅝	15¾	15⅞	2⅛	⅛	1¾	¼	1	3⅜	2¾	4⅜	¾	1/16
14	39	14½	15⅝	8⅛	21⅝	22¼	15¼	16⅜	16⅝	2¼	⅛	1¾	¼	1	3⅜	3	4⅜	¾	1/16
14½	40½	15	16¼	8⅜	22¾	23¼	15⅝	16⅞	17⅛	2¼	⅛	1¾	¼	1	3½	3	4¾	¾	1/16
15	42	15½	16¾	9	22¾	24	16¼	17⅜	17½	2¼	⅛	1¾	¼	1	3½	3	4¾	¾	1/16

Table 38. Bureau of Steam Engineering Standard Low-pressure Valves.—(*Continued*)

Size	P	Q	R	S	T	U	X	P.C.	a	b	c	d	e	f	g	h	i	j	k	m
½	⅝	....	2 9/16	...	5/32	...	⅜	2 9/16	5/16	⅜	13/32	⅜	1 5/16	1 5/16	4	⅛				
¾	¾	....	3	...	5/16	...	⅜	2 7/8	⅜	⅜	13/32	⅜	1½	1⅛	4	½				
1	1	....	3 9/16	...	5/16	...	⅜	3¼	⅝	⅜	13/32	⅜	1¾	1¼	4	½				
1¼	1	....	4	...	5/16	...	⅜	3 7/16	⅝	⅜	13/32	⅜	1¾	1¼	4	½				
1½	1¼	....	4 9/16	...	5/16	...	⅜	3½	⅝	⅜	13/32	⅜	1¾	1⅝	4	½				
2	1¼	3⅛	6	5/16	5/16	...	7/16	4⅜	⅞	5/16	½	7/16	2⅛	1⅝	6	½				
2½	1⅝	3⅝	6 7/8	5/16	5/16	...	7/16	4⅞	⅞	5/16	9/16	½	2⅛	1¾	6	½				
3	1⅝	4⅛	7⅝	5/16	¼	...	7/16	5⅜	1	7/16	9/16	½	2 7/16	1¾	6	⅝	4	5/16		
3½	1½	4⅝	8⅝	5/16	¼	...	½	6¼	1	½	9/16	½	2⅝	1⅞	6	⅝	4	5/16		
4	1½	5⅛	9⅝	5/16	¼	1⅜	½	6⅞	1	½	9/16	½	2⅝	1⅞	6	⅝	4	5/16	3	⅛
4½	1⅝	5⅜	10¼	5/16	¼	1⅜	½	7¾	1⅛	½	9/16	½	2¾	2	8	⅝	4	5/16	3	⅛
5	1⅝	6¼	11⅛	5/16	5/16	1½	½	8	1⅛	½	9/16	½	2⅝	2	8	⅝	4	⅜	3	⅜
5½	1¾	6½	12¼	5/16	5/16	1½	9/16	8¾	1⅛	⅝	¾	⅝	2⅞	2⅛	10	⅝	4	⅜	3	⅜
6	1¾	7⅛	13¼	½	5/16	1½	9/16	9½	1¼	⅝	¾	⅝	2⅞	2⅛	10	⅝	4	⅜	3	⅜
6½	1⅞	7½	14⅛	½	5/16	1¾	⅝	10¼	1¼	⅝	¾	⅝	3⅛	2⅜	12	⅝	4	⅜	3	⅜
7	1⅞	8⅛	15½	½	⅜	1¾	⅝	11½	1¼	⅝	¾	⅝	3⅛	2¾	12	¾	4	½	3	½
7½	2	8½	16⅝	½	⅜	1¾	1 1/16	11¾	1⅜	⅝	¾	⅝	3¼	2⅞	12	¾	4	½	3	½
8	2	9	17⅝	½	⅜	2⅛	¾	12½	1⅜	⅝	¾	⅝	3¼	2⅞	14	¾	4	½	3	½
8½	2⅛	9½	18¾	½	⅜	2⅛	¾	13⅛	1⅜	⅝	¾	⅝	3⅞	3	14	¾	4	½	3	½
9	2⅛	10	20	½	7/16	2⅛	1 5/16	13⅝	1⅜	⅝	¾	⅝	3⅞	3	14	¾	4	½	3	½
9½	2¼	10½	21	½	7/16	2⅜	1 1/16	14⅛	1½	¾	⅞	¾	4⅛	3⅛	16	¾	4	½	3	½
10	2¼	11	22	½	7/16	2⅜	⅞	14⅞	1½	¾	⅞	¾	4⅛	3⅛	16	¾	4	½	3	⅝
10½	2⅜	11½	23	½	7/16	2⅜	⅞	15¾	1½	¾	⅞	¾	4⅜	3⅛	18	¾	4	½	3	⅝
11	2⅜	12	24	½	½	2⅜	⅞	16	1½	¾	⅞	¾	4⅜	3¼	18	¾	4	½	3	⅝
11½	3	12½	25	½	½	2⅝	⅞	17	1⅝	¾	⅞	¾	4½	3⅜	18	⅞	4	½	3	⅝
12	3	13	26½	½	½	2⅝	⅞	17⅝	1⅝	¾	⅞	¾	4½	3⅜	18	⅞	4	⅝	4	⅝
12½	3⅛	13½	28	½	½	2⅝	⅞	18½	1⅝	¾	⅞	¾	4⅝	3½	20	⅞	4	⅝	4	⅝
13	3¼	14	28½	½	9/16	2⅝	⅞	18⅝	1⅝	⅞	1	⅞	5	3⅝	20	⅞	4	⅝	4	⅝
13½	3⅜	14½	29	½	9/16	2¾	⅞	19⅛	1⅝	⅞	1	⅞	5⅛	3¾	20	⅞	4	¾	4	⅝
14	3⅜	15	31	½	⅝	2¾	⅞	19⅞	1⅝	⅞	1	⅞	5⅛	3¾	20	⅞	4	¾	4	⅝
14½	3½	15½	32	½	⅝	2¾	⅞	20¾	1⅝	⅞	1	⅞	5¼	3⅞	20	⅞	4	¾	4	⅝
15	3½	16	33	½	⅝	2¾	⅞	20¾	1⅝	⅞	1	⅞	5¼	3⅞	22	⅞	4	¾	4	⅝

Table 88a. Bureau of Steam Engineering Standard Low-pressure Valves



Limit of working pressure, 100 pounds.
All material to be composition G or M except where otherwise specified.
Valve stems and nuts composition P-r, N-r, or Mn-r.
K = threads per inch on lock nut for stop valve.
a = R. H. Acme threads per inch on valve stems and yoke.
s = split pin in lock nut, drill for, after lock nut is in place.
j = pin for use in regrinding valve, same diameter as "s."
I = valves 6 in. diameter and above made thus.
II = valves 4 in., 4½ in., 5 in., and 5½ in. diameter made thus.
III = valves 3½ in. diameter and below made thus.
Valves 1½ in. diameter and below have cover and yoke one piece.

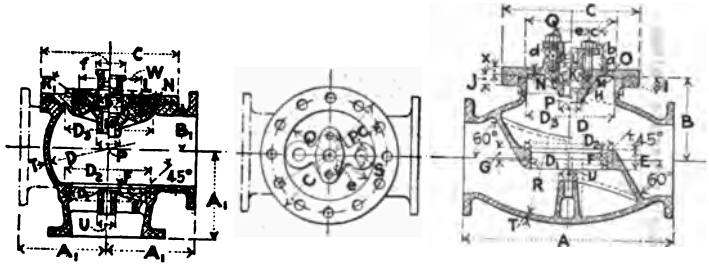
Table 38a. Bureau of Steam Engineering Standard Low-pressure Valves.—(*Continued*)

Size	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	T	
½	3/16	5/16	¼	1	...	....	....	⅞	¾	9/16	20	...	...	½	....	...	⅛	....	½	5/32
¾	3/16	⅜	¼	1¼	...	....	....	1⅛	15/16	⅝	20	...	...	½	....	...	⅛	....	½	7/32
1	3/16	7/16	¼	1⅜	...	....	....	1⅜	1⅛	⅞	18	...	...	⅝	⅛	5/16	⅛	....	⅝	3/16
1¼	¼	7/16	¼	1⅝	...	....	....	1½	1 5/16	15/16	18	...	...	¾	⅛	5/16	⅛	....	¾	3/16
1½	¼	½	5/16	2	...	....	....	1⅝	1⅜	1⅛	16	...	...	¾	⅛	5/16	⅛	....	1 3/16	3/16
2	5/16	½	5/16	2½	...	....	....	2⅛	1⅝	1¼	14	...	...	1	3/16	¼	3/16	....	1 5/16	5/16
2½	⅜	⅝	¼	3⅛	...	....	....	2⅝	1¾	1 7/16	14	...	...	1⅛	3/16	¼	3/16	....	1⅛	3/16
3	7/16	⅜	¼	3⅝	...	....	....	2½	1⅞	1⅝	14	...	...	1⅜	¼	¼	3/16	....	1¼	3/16
3½	½	⅜	¼	4⅛	...	....	....	2½	1⅞	1⅝	14	...	...	1½	¼	¼	3/16	....	1⅜	3/16
4	1⅛	⅞	¼	4⅝	⅜	⅜	⅜	2¾	2	1⅝	14	...	...	...	...	...	...	4	1½	⅜
4½	1¼	⅞	¼	5¼	⅜	⅜	⅜	3	2⅛	2	12	...	...	...	...	...	...	4½	1⅝	⅜
5	1 5/16	1	⅜	5⅞	⅞	⅜	⅜	3	2¼	2	12	...	...	...	...	...	...	5	1 13/16	⅜
5½	1⅜	1⅛	⅜	6¾	⅞	7/16	⅜	3⅜	2½	2 5/16	12	...	...	...	...	...	...	5½	1⅞	7/16
6	1½	1⅜	⅜	6¾	⅞	½	⅜	3⅜	2½	2 7/16	12	1	¾	...	...	...	...	6	2	7/16
6½	1½	1¼	⅜	7½	1	½	7/16	3¾	2⅞	2 9/16	12	1	½	...	...	...	...	6½	2 1/16	½
7	1⅝	1⅜	½	8¼	1	⅝	7/16	3¾	2⅞	2 9/16	12	1⅛	⅝	...	...	...	...	7	2 5/16	½
7½	1 11/16	1½	½	8¾	1	⅝	7/16	4	3⅛	2¾	12	1⅛	7/16	...	...	...	...	7½	2½	9/16
8	1¾	1½	½	9½	1⅛	⅝	7/16	4	3⅛	2¾	12	1¼	7/16	...	...	...	...	8	2⅝	9/16
8½	1⅞	1⅝	½	10	1⅛	⅝	7/16	4½	3½	2 15/16	12	1¼	7/16	...	...	...	...	8½	2¾	⅝
9	1 15/16	1¾	½	10½	1⅛	11/16	⅝	4½	3½	2 15/16	12	1⅜	7/16	...	...	...	...	9	2⅞	11/16
9½	2	1¾	½	11	1¼	¾	⅝	4⅝	3⅝	3⅛	12	1½	½	...	...	...	...	9½	3	11/16
10	2⅛	1⅞	½	11½	1¼	¾	½	4¾	3¾	3⅛	12	1½	½	...	...	...	...	10	3	¾
10½	2¼	1⅞	½	12	1¼	13/16	½	4⅞	3⅞	3⅛	12	1⅝	½	...	...	...	...	10½	3¼	¾
11	2¼	2	½	12½	1¼	13/16	9/16	4⅞	3⅞	3⅛	12	1⅝	9/16	...	...	...	...	11	3⅜	¾
11½	2⅜	2	½	13	1⅜	13/16	9/16	5	4	3⅛	10	1¾	9/16	...	...	...	...	11½	3½	13/16
12	2½	2	½	13½	1⅜	⅞	⅝	5	4	3⅛	10	1⅞	⅝	...	...	...	...	12	3⅝	13/16
12½	2⅝	2	½	14½	1⅜	⅞	⅝	5⅛	4⅛	3½	10	1⅞	11/16	...	...	...	...	12½	3¾	13/16
13	2¾	2¼	½	14⅝	1⅜	⅞	⅝	5¼	4¼	3½	10	2	¾	...	...	...	...	13	3⅞	⅞
13½	2⅞	2⅛	½	15⅛	1⅜	⅞	11/16	5⅜	4⅜	3½	10	2	¾	...	...	...	...	13½	4⅛	⅞
14	3	2⅛	½	15¾	1½	⅞	11/16	5⅜	4⅜	3½	10	2⅛	¾	...	...	...	...	14	4¼	⅞
14½	3⅛	2¼	½	16¼	1½	⅞	11/16	5½	4½	3 7/16	10	2⅛	¾	...	...	...	...	14½	4⅜	15/16
15	3⅛	2¼	½	16¾	1½	⅞	11/16	5½	4½	3⅞	10	2¼	¾	...	...	...	...	15	4½	15/16

Table 38a. Bureau of Steam Engineering Standard Low-pressure Valves.—(*Continued*)

Size	V	W	X	Y	Z	a	b	c	d	e	f	g	h	i	k	l	l_1	m	n	r	s
½	½	¼	9/16	⅜	¼	9	¼	....	9/32	2⅛	⅝	1 3/16	5/32	⅜	⅝	3/16	⅜	1 3/16	⅝	1 1/32	⅜ 2
¾	⅝	¼	⅝	½	¼	8	¼	....	9/32	2 3/16	¾	1⅜	9/16	⅜	1	¼	1 5/16	1	1	7/16	9/32
1	1 3/16	⅜	1 3/16	⅝	5/16	7	¼	....	⅜	2 7/16	1	1¾	¼	⅜	1⅜	⅜	1¼	1¼	1¼	1 7/32	⅛
1¼	1 3/16	⅜	⅞	⅝	5/16	7	¼	....	⅜	2½	1	1¾	¼	⅜	1⅜	⅜	1¼	1⅜	1¼	9/16	⅛
1½	1	⅜	1	¾	⅜	6	5/16	....	7/16	2½	1⅛	2	¼	⅜	1½	⅜	1⅜	1½	1¼	2 1/32	⅛
2	1⅜	½	1 1/16	⅞	½	6	5/16	....	7/16	2⅜	1⅛	2⅜	¼	7/16	1½	⅜	1⅜	1½	1 1/16	2 5/32	⅛
2½	1¼	½	1⅛	⅞	½	6	⅜	....	9/16	2½	1⅜	2⅞	¼	7/16	1¾	½	1¾	1⅝	1¼	2 7/32	⅛
3	1¼	⅝	1¼	⅞	½	6	⅜	....	9/16	2¾	1⅜	3¼	¼	7/16	1¾	⅝	1⅞	1¾	1⅜	2 9/32	⅛
3½	1½	⅝	1¼	1	½	5	⅜	....	9/16	3	1⅜	3 ⅝	¼	½	1⅞	¾	2 1/16	1⅞	1½	2 9/32	⅛
4	1½	¾	1⅜	1	½	5	⅜	....	9/16	3⅛	1⅜	4¼	¼	½	1⅞	⅞	2⅜	1⅞	1¼	2 1/32	⅛
4½	1⅝	¾	1⅜	1⅛	⅝	5	½	....	11/16	3⅜	1⅞	4⅝	¼	½	2	1	2⅜	2	1⅜	1 1/16	⅛
5	1⅝	⅞	1⅝	1¼	⅝	5	½	....	11/16	3 7/16	1⅞	5¼	5/16	9/16	2	1⅛	2⅞	2⅛	1⅝	1 1/16	⅛
5½	1⅞	⅞	1⅝	1¼	⅝	4	½	....	¾	3⅜	2	5½	5/16	9/16	2⅛	1¼	3	2⅛	1¾	1 1/16	⅛
6	1⅞	⅞	1⅝	1¼	⅝	4	½	....	¾	4	2	6	⅜	9/16	2⅛	1⅜	3⅛	2⅛	1¾	1 1/16	⅛
6½	2	1	1¾	1⅜	¾	4	½	....	13/16	4 1/16	2⅛	6½	⅜	⅝	2⅜	1½	3⅜	2¼	2	1⅜	⅛
7	2	1	1¾	1⅝	¾	4	½	....	13/16	4⅜	2⅜	6⅞	⅜	⅝	2⅜	1¾	3⅞	2⅜	2	1⅜	5/16
7½	2¼	1	1¾	1½	¾	4	½	....	15/16	4½	2½	7⅜	⅜	⅝	2½	1⅞	4⅛	2⅜	2¼	1½	5/16
8	2¼	1⅛	2	1½	¾	4	½	....	15/16	4¾	2½	7¾	½	⅝	2½	2	4¼	2½	2¼	1½	5/16
8½	2⅜	1⅛	2	1⅝	⅞	4	9/16	5/32	1	5	2⅜	8¼	½	⅝	2¾	2⅛	4⅜	2⅛	2⅛	1 1/16	5/16
9	2⅜	1⅛	2	1⅝	⅞	4	9/16	5/32	1	5⅛	2⅜	8⅝	½	⅝	2¾	2¼	4⅝	2⅛	2⅛	1 1/16	5/16
9½	2⅝	1¼	2¼	1⅝	⅞	4	9/16	5/32	1	5⅜	3	9⅛	½	⅝	3	2⅜	5⅛	2⅜	2¼	1¾	5/16
10	2⅝	1¼	2¼	1⅝	⅞	4	9/16	5/32	1	5½	3	9¾	⅝	¾	3	2½	5¼	2⅜	2¼	1 1/16	5/16
10½	2¾	1⅜	2½	1⅞	1	4	⅝	5/16	13/16	5⅝	3⅛	10	⅝	¾	3¼	2⅝	5⅜	2⅝	2¾	1⅞	5/16
11	2¾	1⅜	2½	1⅞	1	4	⅝	5/16	13/16	5⅞	3¼	10½	⅝	¾	3¼	2¾	5⅝	3	3	1⅞	5/16
11¼	2⅞	1½	2¾	2	1	4	⅝	5/16	13/16	5⅞	3⅜	10⅞	⅝	¾	3½	3	6⅛	3⅛	3⅛	1 1/16	⅛
12	2⅝	1½	2¾	2	1	4	⅝	5/16	13/16	6	3⅜	11⅜	⅝	¾	3½	3⅛	6¼	3⅛	3⅛	1 1/16	⅛
12½	3⅛	1½	2¾	2⅛	1¼	4	¾	5/16	13/16	6 1/16	3⅜	11¾	⅝	¾	3¾	3⅛	6⅜	3⅛	3⅛	2	⅛
13	3¼	1½	2¾	2¼	1¼	4	¾	5/16	13/16	6⅛	3⅜	12¼	⅝	¾	4	3⅛	6⅝	3⅜	3¼	2 1/16	⅛
13½	3⅜	1½	2¾	2⅜	1¼	4	13/16	5/16	1½	6¼	3¾	12⅝	⅝	¾	4¼	3⅜	7⅛	4	4	2⅛	⅛
14	3⅜	1⅝	3	2⅜	1¼	4	13/16	5/16	1½	6¼	3¾	13⅛	⅝	¾	4¼	4	7⅛	4⅛	4⅛	2⅛	⅛
14½	3½	1⅝	3	2½	1¼	4	13/16	5/16	1½	6⅜	4	13½	⅝	¾	4½	4	7¾	4⅛	4⅛	2 1/16	⅛
15	3½	1⅝	3	2½	1¼	4	13/16	5/16	1½	6⅜	4	14	⅝	¾	4½	4	7¾	4⅛	4⅛	2 1/16	⅛

Table 39. Bureau of Steam Engineering Standard High-pressure Valves



Limit of working pressure 300 lbs.
All material to be composition G or M except where otherwise specified.
Studs for cover class B bolt material.
Studs for gland, nuts for gland studs and nuts for cover studs, composition P-r, N-r or Mn-r.
Dividing wall conical for all valves.
Valves will be cast without removable seats.
Flange shown in dot and dash on cross valve only.
Flanges to conform to Bureau of Steam Engineering Standard Flange List.

D_2 = diameter of false seat.
G = depth of false seat.
P.C. = diameter of pitch circle for cover.
g = number of studs in cover.
h = diameter of studs in cover.
i = number of ribs in cover.
j = thickness of ribs in cover.
k = number of ribs under valve seat.
m = thickness of ribs under valve seat.

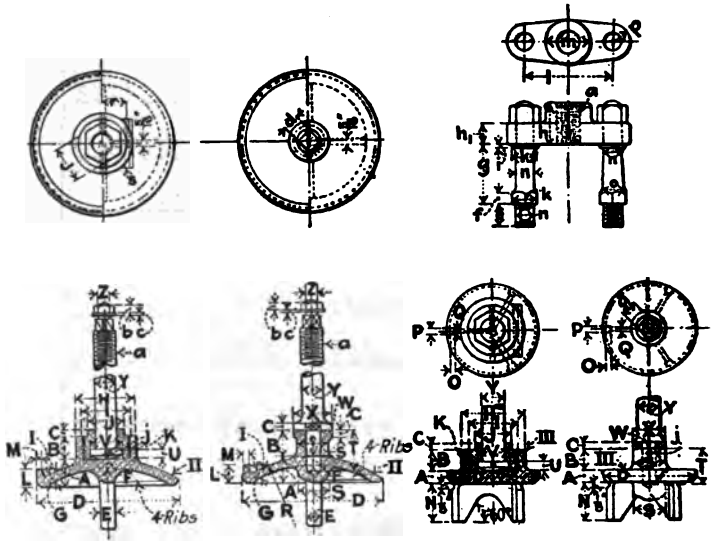
Table 39. Bureau of Steam Engineering Standard High-pressure Valves.—(*Continued*)

Size	A	B	A_1	B_1	C	D	D_1	D_2	D_3	E	F	G	H	I	J	K	L	M	N	O	P
½	6¼	1¼	3⅛	1	3 15/16	2	¾	....	1¼	⅜	1/16	....	2⅛	⅛	9/16	⅞	¾	..	....	..	¾
¾	7	1½	3½	1⅛	4⅜	2¼	1	....	1 1/16	½	1/16	....	2⅞	⅛	9/16	1	⅞	1½	½	1/16	1
1	7½	1¾	3¾	1⅜	4⅞	2¾	1⅜	....	2⅛	½	1/16	....	3 1/16	⅛	⅝	1⅜	1 1/16	1⅛	½	1/16	1½
1¼	8½	2⅛	4¼	1½	5¼	3	1⅝	....	2⅛	⅝	1/16	....	3⅜	⅛	⅝	1⅜	1 1/16	1⅞	½	1/16	1¼
1½	8¾	2½	4⅜	1⅝	5⅝	3½	1¾	....	2⅜	¾	1/16	....	3⅞	⅛	⅝	1½	1¼	2	½	1/16	1⅝
2	10¼	2⅞	5⅛	1⅞	6¾	4	2¼	....	2¾	¾	1/16	....	4½	⅛	¾	1½	1¾	2½	9/16	1/16	1⅝
2½	11½	3⅜	5¾	2¼	7¾	4¾	2⅞	....	3⅜	⅞	1/16	....	5⅛	⅛	¾	1⅝	1⅝	2¾	⅝	1/16	1½
3	12½	3¾	6¼	2½	8⅜	5½	3⅜	....	4	⅞	⅛	....	5¾	9/16	⅞	1¾	1½	2½	⅝	1/16	1¾
3½	13	4¼	6½	2⅞	8⅞	6⅜	3¾	....	4½	1	⅛	....	6¼	9/16	⅞	1⅞	1⅝	2⅝	⅝	1/16	1⅞
4	13¾	4¾	6⅞	3⅛	9½	7⅛	4½	....	5⅛	1	⅛	....	6⅞	9/16	⅞	2	1¾	2¾	1 1/16	1/16	2
4½	15	5⅛	7½	3½	10½	8	5	5¾	6	1⅛	⅛	¾	7⅞	9/16	⅞	2⅛	1⅞	3⅛	1 1/16	1/16	2¼
5	16	5¾	8	3¾	11⅛	8¾	5½	6¼	6¾	1⅛	⅛	⅞	8¼	9/16	1⅛	2½	2	3⅜	¾	1/16	2¼
5½	16¾	6¼	8⅜	4⅛	12	9½	6	6⅞	7	1¼	⅛	1 9/16	8⅞	9/16	1⅛	2⅜	2⅛	3⅝	¾	1/16	2⅜
6	18	6¾	9	4½	12½	10¼	6½	7⅜	7½	1¼	⅛	1	9⅜	9/16	1⅛	2¾	2¼	3⅞	¾	1/16	2⅝
6½	18¾	7⅜	9⅜	4¾	13¼	11	7⅛	8	8¼	1⅜	⅛	1 1/16	10⅛	9/16	1⅛	2⅞	2½	3⅞	1 1/16	1/16	2¾
7	20	8	9⅝	5	14	11⅞	7⅝	8⅝	9	1½	⅛	1⅛	10⅞	9/16	1⅛	3	2½	4	1 1/16	1/16	2⅞
7½	21½	8⅝	10	5¼	14½	12⅝	8¼	9⅛	9¼	1½	⅛	1¼	11⅜	9/16	1¼	3 1/16	2¾	4⅛	⅞	1/16	3⅛
8	22¾	9⅛	10⅜	5½	15¼	13½	9	9⅞	10	1⅝	⅛	1 9/16	11⅞	9/16	1¼	3¼	3	4⅜	⅞	1/8	3¼
8½	24	9⅝	10¾	5¾	16	14¼	9½	10½	10⅝	1¾	⅛	1 9/16	12½	9/16	1¼	3⅜	3	4½	⅞	1/8	3⅜
9	25½	10⅛	11¼	6	16½	15	10	11	11⅛	1⅞	⅛	1⅜	13	9/16	1¼	3½	3	4¾	1 9/16	1/8	3½
9½	26¾	10¾	11⅝	6¼	17½	15¾	10½	11½	11¾	1⅞	⅛	1 11/16	13⅜	9/16	1⅜	4	3¼	5⅛	1 9/16	1/8	3⅝
10	28	11¼	12	6½	18	16½	11	12	12⅛	2	1/16	1¼	14⅛	9/16	1⅜	4	3⅝	5¼	1	1/8	3⅞
10½	29¼	11¾	12½	6⅝	18¾	17¼	11½	12½	12⅞	2	1/16	1½	14⅞	9/16	1⅜	4 1/16	3½	5½	1	1/8	3¾
11	30½	12⅛	13	7	19½	18	12	13	13¼	2	1/16	1½	15½	9/16	1⅜	4¼	3½	5⅝	1 1/16	1/8	4
11½	31¾	12⅜	13¼	7¼	20	18¾	12½	13⅝	13¾	2⅛	1/16	1⅝	16	9/16	1⅜	4⅜	3¾	5⅞	1 1/16	1/8	4¼
12	33	13	13⅝	7½	20 1/16	19½	13⅛	14¼	14⅜	2⅛	1/16	1⅝	16⅜	9/16	1⅜	4½	3¾	6	1 1/16	1/8	4⅜
12½	34½	13⅜	14¼	7¾	21½	20¼	13⅝	14¾	14⅞	2⅛	¼	1⅝	17⅛	9/16	1½	4⅝	3¾	6⅛	1⅛	1/8	4½
13	36	13¾	14¾	8	22¼	21	14⅛	15¼	15⅜	2¼	¼	1¾	17⅞	9/16	1½	4⅝	4	6¼	1⅛	1/8	4⅝
13½	37½	14⅛	15¼	8¼	23	21⅜	14⅝	15⅝	15⅞	2¼	¼	1¾	18⅝	9/16	1½	4¾	4	6⅜	1 3/16	1/8	4¾
14	39	14½	15⅝	8⅝	22⅜	22½	15¼	16⅜	16⅝	2¼	¼	1¾	19⅜	9/16	1½	4⅞	4	6½	1 3/16	1/8	4⅞
14½	40½	15	16¼	8⅞	24¼	23¼	15¾	16⅞	17⅛	2¼	¼	1¾	19⅞	9/16	1½	5	4¼	6⅝	1 3/16	1/8	5¼
15	42	15½	16¾	9¼	24¾	24	16¼	17⅜	17½	2¼	¼	1¾	20⅝	9/16	1½	5 1/16	4¼	6⅞	1¼	1/8	5¼

Table 39. Bureau of Steam Engineering Standard High-pressure Valves.—(*Continued*)

Size	Q	R	R_1	S	T	U	W	X	P.C.	a	b	c	d	e	f	g	h	i	j	k	m
½	....	2 9/16	5/16	1 3/16	5/16	....	2 1/8	½	2 7/8	½	3/8	1 3/32	3/8	1 5/8	1	4	½				
¾	2 3/4	3	3/8	⅞	¼	....	2 1/4	½	3 1/2	⅝	3/8	1 3/32	3/8	1 3/4	1 1/4	4	½				
1	2 3/4	3 5/16	5/16	⅞	¼	....	2 1/2	9/16	3 1 3/16	⅝	3/8	1 3/32	3/8	2 1/4	1 9/16	4	½				
1¼	2 3/4	4	3/16	⅞	¼	....	2 1/2	9/16	4 1/8	⅝	3/8	1 3/32	3/8	2 1/4	1 9/16	6	½				
1½	3 3/8	4 3/16	7/16	1 1/8	¼	....	2 13/16	9/16	4 3/16	⅞	7/16	½	7/16	2 5/16	1¾	6	½				
2	4	6	½	1 1/4	3/16	....	3	⅝	5 3/8	⅞	7/16	½	7/16	2 7/16	1 7/8	6	⅝				
2½	4 5/8	6 7/8	½	1 3/8	⅜	....	3 3/8	⅝	6	1	½	9/16	½	2 5/8	1 7/8	8	⅝				
3	5 1/8	7 3/4	½	1 3/8	⅜	....	3 3/4	¾	6 3/4	1 1/8	½	9/16	½	2 3/4	2	8	¾	4	⅜		
3½	5 5/8	8 5/8	½	1 3/8	7/16	....	3 7/8	¾	7 1/4	1 1/8	⅝	⅝	⅝	2 7/8	2 1/8	8	¾	4	⅜		
4	6 1/4	9 3/8	1 3/16	1 5/8	7/16	1 3/8	4 1/4	¾	7 7/8	1 1/8	⅝	⅝	⅝	3 3/8	2 3/4	10	¾	⅝	⅜	3	¼
4½	7 1/4	10 1/4	1	1 7/8	½	1 3/8	4 3/4	¾	8 3/8	1 1/4	⅝	¾	⅝	3 3/4	2 7/8	10	¾	4	⅜	3	¼
5	7 3/8	11 1/8	1	2 1/8	½	1 1/2	5 1/4	⅞	9 3/8	1 1/4	⅝	¾	⅝	3 3/4	2 7/8	10	⅞	4	⅜	3	⅜
5½	8	12 1/4	1	2 1/8	9/16	1 1/2	5 1/4	⅞	10	1 1/4	⅝	¾	⅝	4	2 7/8	10	⅞	4	⅜	3	⅜
6	8 1/2	13 3/8	½	2 3/8	9/16	1 1/2	5 1/2	⅞	10 1/2	1 3/8	¾	⅞	¾	4 1/4	3	10	⅞	4	½	3	⅜
6½	9 1/4	14 3/8	½	2 3/8	⅝	1 3/4	5 5/8	⅞	11 1/4	1 3/8	¾	⅞	¾	4 3/8	3 1/4	12	⅞	4	½	3	½
7	10	15 1/4	¼	2 3/8	⅝	1 3/4	6 1/4	⅞	12	1 3/8	¾	⅞	¾	4 1/2	3 3/8	12	⅞	4	⅝	3	½
7½	10 1/8	16 1/8	¼	2 7/8	1 1/16	1 3/4	6 3/4	1	12 7/8	1 7/8	⅝	¾	⅝	4 1/4	3 1/2	12	1	4	½	3	½
8	10 7/8	17 3/8	¼	2 7/8	1 1/16	2 1/4	7 1/4	1	13 1/8	1 1/2	⅞	1	⅞	5	3 3/8	12	1	4	⅝	3	⅝
8½	11 1/2	18 1/4	¼	3 1/8	⅝	2 1/4	7 1/2	1	13 3/4	1 1/2	⅞	1	⅞	5 1/8	3 3/4	14	1	4	⅝	3	⅝
9	12	20	¼	3 1/4	⅝	2 1/4	7 3/4	1	14 1/4	1 1/2	⅞	1	⅞	5 1/4	3 7/8	14	1	4	⅝	3	⅝
9½	12 1/4	21	¼	3 1/4	⅝	2 3/8	8 1/4	1 1/8	15	1 3/8	⅞	1	⅞	5 7/8	4 3/8	14	1 1/8	4	⅝	3	⅝
10	12 3/4	22	....	3 1/2	1 3/16	2 3/8	9	1 1/8	15 1/2	1 3/8	¾	1	⅞	5 7/8	4 3/8	14	1 1/8	4	⅝	3	¾
10½	13 1/2	23	....	3 3/4	1 3/16	2 3/8	9	1 1/8	16 1/4	1 3/8		1 1/8	1	6	4 1/2	16	1 1/8	6	¾	3	¾
11	14 1/4	24	....	3 3/4	⅞	2 3/8	9 1/4	1 1/8	17	1 3/4	1	1 1/8	1	6 1/8	4 3/8	16	1 1/8	4	¾	3	¾
11½	14 3/4	25	....	4 1/4	⅞	2 3/8	10	1 5/16	17 1/2	1 3/4	1	1 1/8	1	6 1/4	4 3/8	16	1 1/8	4	¾	3	¾
12	15 3/8	26 1/2	....	4 1/4	1 5/16	2 3/8	10	1 3/16	18 1/8	1 3/4	1	1 1/8	1	6 3/8	4 7/8	16	1 1/8	4	¾	4	¾
12½	15 3/4	28	....	4 1/4	1	2 3/8	10 1/2	1 3/16	18 3/4	1 3/4	1	1 1/8	1	6 3/4	5	18	1 1/4	4	¾	4	¾
13	16 1/2	28 1/2	....	4 3/4	1	2 3/8	11	1 1/4	19 1/2	1 7/8	1	1 1/8	1	6 7/8	5	18	1 1/4	4	¾	4	¾
13½	17 1/4	29	....	4 3/4	1 1/16	2 3/8	11 1/4	1 1/4	20 1/4	1 7/8	1	1 1/8	1	6 3/4	5 1/4	18	1 1/4	4	¾	4	¾
14	18	31	....	4 3/4	1 1/16	2 3/4	11 1/2	1 3/8	21	1 7/8	1	1 1/8	1	6 3/4	5 1/4	18	1 1/4	4	¾	4	¾
14½	18 1/2	32	....	5	1 1/8	2 3/4	12 1/4	1 3/8	21 1/2	2	1	1 1/8	1	7	5 3/8	18	1 1/4	4	¾	4	¾
15	19	33	....	5	1 1/8	2 3/4	13	1 1/2	22	2	1	1 1/8	1	7 1/8	5 3/8	18	1 1/4	4	¾	4	¾

Table 39a. Bureau of Steam Engineering Standard High-pressure Valves



Limit of working pressure, 300 lbs.
All material to be composition, G or M, except where otherwise specified.
Valve stems and nuts, composition P-r, N-r, or Mn-r.
Yoke and standards, wrought steel, Class B forgings.
Nuts for standards, wrought iron.
K = threads per inch on lock nut for stop valve.
a = R.H. Acme threads per inch on valve stems and yoke.
s = split pin in lock nut, drill for after lock nut is in place.
j = pin for use in regrinding valve, same diameter as "s."
I = valves 6 in. diameter and above, made thus.
II = valves 4 in., 4½ in., 5 in., and 5 in. diameter, made thus
III = valves 3½ in. diameter and below, made thus.

Table 39*a*. Bureau of Steam Engineering Standard High-pressure Valves.—(*Continued*)

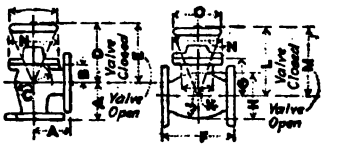
Size	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V
½	3/16	5/8	3/16	1 1/16	..	....	....	1 1/16	7/8	5/8	20	..	..	1/2	..	..	1/8	....	5/8	7/16	5/32	5/8
¾	1/4	3/8	3/16	1 3/16	..	....	....	1 3/16	1 1/16	13/16	18	..	..	5/8	1/8	3/16	1/8	....	3/4	5/8	5/32	1 3/16
1	1/4	1/2	1/4	1 5/8	..	....	....	1 1/2	1 1/4	1 3/16	18	..	..	5/8	1/8	3/16	1/8	....	7/8	1 1/16	3/16	1
1¼	5/16	1/2	1/4	1 7/8	..	....	....	1 3/4	1 1/2	1 1/4	16	..	..	3/4	1/8	3/16	1/8	....	7/8	3/4	3/16	1 5/16
1½	5/16	1/2	1/4	2	..	....	....	1 3/4	1 1/2	1 1/4	16	..	..	3/4	1/8	3/16	1/8	....	1	1 3/16	1/4	1 1/4
2	3/8	9/16	1/4	2 1/2	..	....	....	2 1/8	1 3/4	1 7/16	14	..	..	1	3/16	1/4	3/16	....	1 1/4	1 3/16	5/16	1 1/4
2½	3/16	5/8	5/16	3 1/8	..	....	....	2 3/8	2	1 5/8	14	..	..	1 1/8	3/16	1/4	3/16	....	1 3/8	1 1/8	3/8	1 1/2
3	1/2	3/4	5/16	3 3/4	..	....	....	2 5/8	2 1/4	1 13/16	12	..	..	1 3/8	1/4	1/4	3/16	....	1 3/8	1 1/4	3/8	1 5/8
3½	9/16	1 3/16	3/8	4 1/8	..	....	....	2 7/8	2 1/2	2	12	....	..	1 1/2	1/4	1/4	3/16	....	1 1/2	1 3/8	7/16	1 7/8
4	1 1/8	7/8	3/8	4 7/8	3/4	1/2	7/16	3 1/8	2 3/8	2	12	..	..	..	..	..	..	4	1 3/8	1 1/2	7/16	2
4½	1 1/4	7/8	3/8	5 3/8	3/4	5/8	7/16	3 3/4	2 7/8	2 3/8	12	..	..	..	..	..	..	4 1/2	1 3/8	1 5/8	1/2	2 1/4
5	1 3/8	1 & 1 1/8	1/2	5 7/8	7/8	5/8	7/16	3 3/4	3 1/8	2 3/8	12	..	..	..	..	..	..	5	1 3/4	1 13/16	1/2	2 1/4
5½	1 1/2	1 1/8	3/8	6 3/8	7/8	1 1/16	7/16	4	3 1/4	2 3/4	12	....	..	..	..	..	..	5 1/2	1 3/4	1 7/8	9/16	2 3/8
6	1 5/8	1 1/8	3/8	6 7/8	7/8	1 1/16	1/2	4 1/4	3 1/2	2 3/4	12	1 1/4	3/8	..	..	..	..	6	1 3/4	2	5/8	2 3/8
6½	1 5/8	1 1/4	3/8	7 1/2	1	3/4	1/2	4 1/2	3 3/4	2 13/16	12	1 1/2	3/8	..	..	..	..	6 1/2	1 3/4	2 1/16	5/8	2 3/4
7	1 3/4	1 3/8	1/2	8 1/4	1	3/4	9/16	4 3/4	4	3 1/8	10	1 5/8	3/8	..	..	..	..	7	1 3/4	2 5/16	1 1/16	2 7/8
7½	1 7/8	1 1/2	1/2	8 3/4	1	1 3/16	9/16	4 7/8	4 1/4	3 1/4	10	1 3/4	7/16	..	..	..	..	7 1/2	1 3/4	2 1/2	3/4	3 1/8
8	2	1 1/2	1/2	9 1/2	1 1/4	1 3/16	5/8	5 1/4	4 1/4	3 1/2	10	1 3/4	7/16	..	..	..	..	8	2	2 5/8	1 3/16	3 1/4
8½	2 1/8	1 5/8	1/2	10	1 1/4	7/8	5/8	5 1/2	4 1/2	3 1/2	10	1 7/8	7/16	..	..	..	..	8 1/2	2	2 3/4	7/8	3 3/8
9	2 1/4	1 3/4	1/2	10 1/2	1 1/4	7/8	1 1/16	5 3/4	4 3/4	3 7/8	10	1 7/8	7/16	..	..	..	..	9	2	2 7/8	7/8	3 1/2
9½	2 1/4	1 3/4	1/2	11	1 1/4	1 3/16	1 1/16	5 7/8	4 7/8	3 7/8	10	2	1/2	..	..	..	..	9 1/2	2 1/4	3	1 3/16	3 7/8
10	2 3/8	1 7/8	1/2	11 1/2	1 1/4	1	3/4	6 1/4	5 1/4	4 1/4	10	2	1/2	..	..	..	..	10	2 3/8	3	1	3 7/8
10½	2 1/2	1 7/8	1/2	12	1 1/4	1 1/16	3/4	6 3/8	5 1/4	4 1/4	10	2	1/2	..	..	..	..	10 1/2	2 1/2	3 1/4	1	4
11	2 5/8	2	1/2	12 1/2	1 1/4	1 1/16	1 3/16	6 1/2	5 1/4	4 1/4	10	2 1/2	9/16	..	..	..	..	11	2 1/2	3 3/4	1 1/16	4 1/2
11½	2 5/8	2	1/2	13	1 3/4	1 1/8	1 3/16	6 7/8	5 3/8	4 3/8	10	2 1/2	9/16	..	..	..	..	11 1/2	2 3/4	3 1/2	1 1/16	4 3/8
12	2 3/4	2	1/2	13 3/8	1 3/4	1 1/8	7/8	7	5 1/8	5	10	2 1/2	5/8	..	..	..	..	12	2 3/4	3 3/8	1 1/16	4 1/2
12½	2 7/8	2 1/8	1/2	14 3/8	1 3/4	1 1/4	7/8	7 1/4	6	5	10	2 1/2	1 1/16	..	..	..	..	12 1/2	2 3/4	3 3/4	1 1/8	4 5/8
13	3	2 1/8	1/2	14 7/8	1 3/4	1 1/4	1 1/16	7 1/2	6 1/4	5	10	2 3/4	3/4	..	..	..	..	13	2 3/4	4	1 1/8	4 5/8
13½	3 1/8	2 1/8	1/2	15 3/8	1 3/4	1 1/4	1 1/16	7 3/4	6 1/2	5 3/8	10	2 3/4	3/4	..	..	..	..	13 1/2	2 3/4	4 1/8	1 1/8	4 3/4
14	3 1/8	2 1/4	5/8	16	1 1/2	1 3/8	1 1/16	8	6 3/8	5 3/4	10	2 3/4	3/4	..	..	..	..	14	3	4 1/4	1 3/16	4 7/8
14½	3 1/4	2 1/4	5/8	16 1/2	1 1/2	1 3/8	1	8 1/4	6 3/4	5 3/4	10	2 1/2	3/4	..	..	..	..	14 1/2	3	4 3/8	1 5/16	5
15	3 3/8	2 1/4	5/8	17	1 1/2	1 1/2	1	8 1/2	7	5 3/4	10	2 1/2	3/4	..	..	..	..	15	3	4 1/2	1 1/4	5 1/4

Table 39a. Bureau of Steam Engineering Standard High-pressure Valves.—*(Continued)*

Size	W	X	Y	Z	a	b	c	d	e	f	g	h	h_1	i	k	l	m	n	o	p	r	s
½	¼	⅝	½	¼	8	¼	...	9/32	½	¼	15/16	⅞	½	3/16	⅝	2¼	1	7/16	...	½	13/32	3/32
¾	⅜	¾	⅝	5/16	7	5/16	...	⅜	½	¼	1⅛	⅞	½	3/16	⅝	2¼	1⅛	7/16	...	½	17/32	3/32
1	⅜	⅞	¾	⅜	6	5/16	...	7/16	9/16	5/16	1 15/16	1	9/16	¼	3 11/16	2½	1¼	½	...	9/16	1 9/32	⅛
1¼	⅜	⅞	¾	⅜	6	5/16	...	7/16	9/16	5/16	1 15/16	1	9/16	¼	1 11/16	2½	1¼	½	...	9/16	2 9/32	⅛
1½	⅜	1	⅞	½	6	⅜	...	9/16	¾	⅜	2⅜	1⅛	¾	¼	⅞	2 13/16	1½	⅝	...	1 1/16	8 3/32	⅛
2	⅝	1⅛	⅞	½	6	⅜	...	9/16	¾	⅜	2⅜	1⅛	¾	¼	⅞	3	1⅛	⅝	...	1 1/16	2 7/32	⅛
2½	⅝	1⅜	1	½	5	⅜	...	9/16	¾	⅜	2 11/16	1¼	¾	¼	⅞	3⅜	1⅜	⅝	...	1 1/16	3 1/32	⅛
3	⅝	1⅜	1⅛	⅝	5	½	...	1 1/16	15/16	½	2 15/16	1⅜	⅞	5/16	1 1/16	3¾	2⅛	¾	...	1 5/16	1 1/16	⅛
3½	⅝	1½	1¼	⅝	4	½	...	¾	1 5/16	½	3⅛	1⅜	⅞	5/16	1 1/16	3⅞	2¼	¾	...	1 5/16	1 5/16	⅛
4	¾	1⅝	1⅜	¾	4	½	...	1 5/16	1⅛	9/16	3¼	1⅞	1	5/16	1¼	4¼	2⅜	⅞	...	1 5/16	1¼	⅛
4½	¾	1¾	1½	¾	4	½	...	1 5/16	1¼	⅝	3½	2⅛	1⅛	⅜	1 7/16	4¾	2⅝	1	...	1	1⅜	5/16
5	⅞	1¾	1½	¾	4	½	...	1 5/16	1⅜	⅝	3¾	2⅜	1¼	⅜	1⅝	5¼	2⅝	1⅛	...	1¼	1½	5/16
5½	⅞	1⅞	1⅝	⅞	4	9/16	5/32	1	1⅜	⅝	3 15/16	2⅝	1¼	⅜	1⅝	5¼	2¾	1⅜	...	1¼	1 9/16	5/16
6	⅞	2	1¾	⅞	4	9/16	5/32	1	1 9/16	¾	4 3/16	2¾	1⅜	⅜	1¾	5½	3	1⅛	...	1⅜	1 11/16	5/16
6½	1	2⅛	1⅞	1	4	⅝	5/16	1 13/16	1 9/16	¾	4 5/16	2⅞	1⅝	⅜	1¾	5⅝	3⅛	1¼	...	1⅜	1 13/16	5/16
7	1	2¼	2	1	4	⅝	5/16	1¼	1⅜	⅞	4 7/16	3	1½	7/16	2	6¼	3½	1⅝	...	1⅝	1 15/16	5/16
7½	1	2⅜	2⅛	1⅛	4	¾	5/16	1⅜	1 15/16	1	4¾	3¼	1⅝	7/16	2 9/16	6¾	3⅜	1½	...	1 7/16	2	5/16
8	1⅛	2½	2¼	1⅛	4	¾	5/16	1⅜	1 15/16	1	5	3½	1⅝	7/16	2 9/16	7⅛	4	1½	...	1 7/16	2 1/16	5/16
8½	1⅛	2⅝	2⅜	1¼	4	1 3/16	5/16	1½	2 1/16	1 1/16	5⅛	3⅜	1¾	7/16	2⅝	7½	4⅛	1⅝	...	1½	2 9/16	5/16
9	1⅛	2¾	2½	1¼	4	1 3/16	5/16	1½	2 1/16	1⅛	5¼	3⅞	1⅞	½	2 9/16	7¾	4¼	1¾	...	1 1/16	2 9/16	5/16
9½	1¼	2⅞	2⅝	1½	4	⅞	7/32	1⅜	2 3/16	1⅛	5⅜	4	1⅞	½	2 9/16	8¼	4⅝	1⅝	...	1 1/16	2⅝	5/16
10	1¼	3	2⅝	1½	4	⅞	7/32	1⅜	2¼	1¼	5⅝	4	2	½	2¾	9	4⅝	1⅞	2¼	1⅞	2½	¼
10½	1⅜	3	2¾	1½	4	⅞	7/32	1⅜	2½	1⅜	6	4¼	2¼	⅝	2 13/16	9	5	2	2⅜	1 15/16	2 9/16	¼
11	1⅜	3⅛	2⅞	1⅝	4	1 3/16	7/32	1⅞	2½	1⅜	6¼	4½	2¼	⅝	2 13/16	9¼	5⅛	2	2⅜	1 5/16	2 1/16	¼
11½	1⅜	3¼	3	1¾	4	1	7/32	2	2 1/16	1⅜	6½	4⅝	2½	¾	3⅛	10	5¼	2¼	2⅝	2⅛	2 1/16	¼
12	1½	3⅜	3⅛	1¾	4	1	7/32	2	2 1/16	1⅝	6¾	4¾	2½	⅜	3⅛	10	5⅜	2¼	2⅝	2⅛	2⅞	¼
12½	1½	3⅜	3¼	1¾	4	1	7/32	2	2 1/16	1⅝	6⅞	4⅞	2½	¾	3⅛	10½	6	2¼	2⅝	2⅛	2 1/16	¼
13	1¼	3½	3¼	1¾	4	1	7/32	2	3⅛	1¾	7⅛	5	2¾	⅞	3½	11	6	2½	3	2⅝	3 1/16	¼
13½	1¼	3⅝	3⅜	2	4	1¼	7/32	2¼	3⅛	1¾	7¼	5 1/16	2¾	⅞	3½	11¼	6⅛	2½	3	2⅝	3 3/16	¼
14	1⅜	3⅝	3½	2	4	1¼	7/32	2¼	3⅛	1¾	7½	5 1/16	2¾	⅞	3½	11½	6⅛	2½	3	2⅝	3¼	¼
14½	1⅜	3⅞	3⅝	2	4	1¼	7/32	2¼	3¼	2	7⅞	5⅜	3	1	3½	12¼	6⅜	2¾	3¼	2⅝	3 5/16	5/16
15	1⅜	4¼	3¾	2	4	1¼	7/32	2¼	3¼	2	8	5 1/16	3	1	3⅝	13	6¾	2¾	3¼	2⅝	3 1/16	5/16

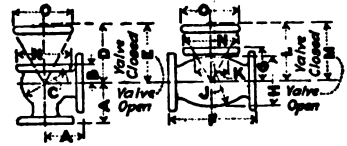
Table 40. Dimensions of Inside Screw Cast Iron Valves

(Newport News Shipbuilding & Dry Dock Co.)
2½ in. to 4 in. pressures 200 pounds.
4½ in. to 10 in. pressures 100 pounds.



Size of valve	Angle and cross valves										Globe valves												N	O
	A	B	C	Stop valve		Check valve		Stop lift check valve			F	G	H	J	K	Stop valve		Check valve		Stop lift check valve			
				D	E	D	E	D	E							L	M	L	M	L	M		
2½	5⅛	2⅛	4⅞	7⅛	7¾	7⅛	7¾	7 31/16	9 1/16		10	3⅞	5¾	4 3/16	4¼	8⅞	9½	8⅞	9½	9 9/16	10 13/16	6½	6
3	6	2⅝	5¾	8	8¾	8	8¾	9 9/16	10 9/16		10¾	4⅝	6½	5⅛	5	10¼	11	10¼	11	11 5/16	12 13/16	6¾	7
3½	6¼	2⅝	6½	8 15/32	9⅜	8 17/32	9 13/32	9½	11 5/16		11½	5⅛	7¼	6¼	5½	10 15/16	11 13/16	11	11⅞	11 13/32	12¾	7½	7
4	6¾	2⅞	7⅛	9	10	9	10	10 3/16	12 3/16		12½	5⅝	7⅝	7¼	6¼	11¾	12¾	11¾	12⅝	12 15/16	14 15/16	8¼	8
4½	6¾	3⅛	8	8¼	9⅜	8 5/16	9 7/16	10	12¼		14½	5⅛	8½	8	6¾	11½	12 9/16	11½	12⅝	13 3/16	15 7/16	8⅞	8
5	6¾	3⅜	8¾	8 11/16	9 15/16	8 11/16	9 15/16	10 7/16	12 15/16		15	5½	9½	8	7¼	12 5/16	13 3/16	12 5/16	13 1/16	13 15/16	16 7/16	9¾	9
6	7½	3⅞	10⅛	9⅛	10 5/16	9¼	10¾	11 9/16	14 3/16		16	6¼	11	8½	8⅛	13 9/16	14⅞	13½	15	15 7/16	18 7/16	10½	9
7	8¼	4¾	11¾	11 9/16	13 5/16	11 5/16	12 13/16	12 15/16	16 7/16		16½	7¼	12	10½	9½	16¼	18	15¾	17½	17 9/16	21 1/16	12¼	10
8	9	4⅞	13⅜	10½	12½	10¼	12¼	12 15/16	16 15/16		19½	7⅞	15⅛	12 9/16	10⅝	16 15/16	18 1/16	15 13/16	17 13/16	18½	22½	13	10
9	9⅝	5⅜	14⅝	11⅜	13⅝	10¾	13	13 15/16	18 7/16		20	8½	16⅛	12 9/16	11⅞	17 9/16	19 1/16	16 9/16	18 13/16	19¼	24¼	14¼	12
10	10¼	5⅞	16⅛	12 5/16	14 13/16	11⅜	13⅞	14¾	19⅝		22½	9	17⅛	14 9/16	12⅞	18 11/16	21 5/16	17¾	20¼	20	25	15¼	12
11	11	6½	18⅜	13 1/16	15 13/16	13 1/16	15 13/16	15⅝	18⅝		24	11	19⅜	15 11/16	14⅛	19 1/16	21 13/16	19 1/16	21 13/16	21⅝	27½	16¼	14

Table 41. Dimensions of Outside Screw Cast Iron Valves Light Pressure, 100 lbs. and Below, Merchant Standard



(Newport News Shipbuilding & Dry Dock Co.)

Size of valve	Angle and cross valves					Globe valves							Both	
	A	B	C	D	E	F	G	H	J	K	L	M	N	O
2	4 7/8	2 1/4	5 1/2	8 1/4	8 7/8	9 3/4	3 3/8	5 1/2	5	4 1/4	9 7/16	10 1/16	6 1/2	6
2 1/2	5 1/8	2 3/8	6 1/4	8 1/2	9 1/4	10 1/2	3 13/16	6 1/4	6	4 3/4	9 5/8	10 5/8	7	6
3	6	2 1/2	7 1/8	9 1/2	10 3/8	12	4 7/32	7 1/8	6 5/16	5 3/8	11 7/32	12 3/32	7 1/2	7
3 1/2	6 1/4	2 3/4	8	10	11	13	4 23/32	8	8	7 1/8	11 5/8	12 7/8	8 1/2	8
4	6 3/4	3	8 3/4	11 1/4	12 3/8	14 1/2	5 1/4	8 3/4	10 3/8	7	13 7/16	14 9/16	9 1/2	8
4 1/2	7 1/4	3 1/4	9 3/8	11 15/16	13 3/16	15	5 3/4	9 3/8	9 11/16	7 3/8	14 7/16	15 11/16	9 3/4	9
5	7 1/4	3 1/2	10 1/8	12 3/16	13 9/16	16	6 1/8	10 1/8	10 1/16	8	14 13/16	16 3/16	10 1/2	9
6	8	4 1/8	11 3/8	13 3/4	15 1/4	18	7 5/16	11 3/8	12 9/16	9 3/8	16 15/16	18 7/16	11 3/4	10
7	8 3/4	4 3/4	13 1/4	14 7/16	16 5/16	20 1/2	8 3/8	13 1/4	15 1/8	10 3/4	18 1/16	19 15/16	13 1/4	10
8	9 1/2	5 1/8	14 3/8	15 5/16	17 3/8	22 1/2	9 1/4	14 3/4	17 1/8	12	19 7/16	21 5/8	14 1/2	10
9	10	5 3/4	16 3/4	17 1/8	19 5/8	24 1/2	10 3/4	16 3/4	19 3/16	13 1/4	22 1/4	24 5/8	15 1/2	12
10	11	6 1/4	17 7/8	17 15/16	20 7/16	26 1/2	11 5/8	17 7/8	21 3/16	14 3/8	23 5/16	25 13/16	17	12
12	12 1/2	7 1/2	21 1/4	20 5/16	23 7/16	30 1/2	11	21 1/4	25 3/4	17 1/8	23 13/16	26 15/16	20 3/4	14
16	15	9 3/4	27 1/4	25	29	40	16	27 1/4	34 3/8	20 3/4	31 3/4	35 7/8	24 1/4	18

Heavy Pressure, 101–200 lbs., Merchant Standard

Sizes 2 in. to 4 1/2 in. inc. same dimensions as light pressure valve given above

Size of valve	Angle and cross valves					Globe valves							Both	
	A	B	C	D	E	F	G	H	J	K	L	M	N	O
5	7 1/2	3 1/2	10 3/4	12 5/8	14	16	6 1/4	10 9/16	10 3/16	8 1/16	15 3/16	16 3/4	10 1/2	10
8	10	5 1/8	15 1/4	17 9/16	19 3/4	23	9 1/4	15 1/16	17 5/16	12 3/16	21 5/16	23 1 5/16	14 1/2	14

Extra Heavy Pressure, up to 275 lbs., Merchant Standard

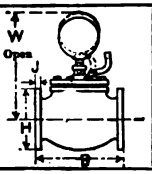
Size of valve	Angle and cross valves					Globe valves							Both	
	A	B	C	D	E	F	G	H	J	K	L	M	N	O
2 1/2	5 1/8	2 1/2	6 1/4	9 21/32	10 13/32	10 1/2	3 15/16	6 1/4	6	4 3/4	11 3/32	11 27/32	7	7
3	6	2 3/4	7 1/8	10 29/32	11 25/32	11 3/4	4 15/32	7 1/8	6 5/16	5 3/8	12 9/16	13 7/16	7 1/2	8
3 1/2	6 1/4	3	8	11 3/4	12 11/16	12 3/4	5 1/32	8	8	6 1/8	13 3/8	14 1 3/16	8 3/8	9
4	6 3/4	3 1/8	8 3/4	12 9/16	13 1 1/16	14 1/2	5 3/8	8 3/4	10 3/8	7	14 1 3/16	15 1 5/16	9 1/4	10
4 1/2	7 1/4	3 7/16	9 3/8	13 3/8	14 7/8	15	5 15/16	9 3/8	9 11/16	7 3/8	16 1/8	17 3/8	10	10
5	7 1/2	3 7/8	10 3/8	14 7/8	16 1/4	16	6 1/2	10 3/8	10 3/16	8 1/4	17 7/16	18 1 3/16	10 7/8	12
6	8 1/4	4 3/8	12 1/4	16 3/8	17 7/8	18 1/2	7 5/16	11 3/4	12 1 1/16	9 3/8	19 5/16	20 1 3/16	12	14
7	9 1/2	5 1/8	14	17 15/16	19 1 1/16	20 3/4	8 1 3/16	13 1/2	15 1/4	11	21 3/8	23 3/8	13 1/2	16
8	10	5 1/2	15 1/4	19 27/32	22	22 1/2	9 5/8	15 1/8	17 5/16	12 3/16	23 31/32	26 1/8	14 1/2	18

Composition Valves, Merchant Standard, Pressures up to 200 lbs.

Size of valve	Angle and cross valves					Globe valves							Both	
	A	B	C	D	E	F	G	H	J	K	L	M	N	O
1 1/2	3 7/8	1 9/16	3 3/4	6 1 1/16	7 3/16	7 1/2	2 3/4	3 3/4	3 1/8	3 1/8	7 7/8	8 3/8	5	4 1/2
2	4 1/2	1 5/8	4 3/4	7 1 1/16	8 3/16	9	3 1/4	4 3/4	4 1 3/16	3 7/8	9 1/16	9 1 1/16	6 1/4	6
2 1/2	4 5/8	1 3/4	5 1/8	7 5/8	8 3/8	9 1/4	3 9/16	5 1/8	5 9/16	4 1/8	9 7/16	10 9/16	6 5/8	6
3	4 3/4	2 1/4	6 1/4	8 1 5/16	9 1 3/16	10	4 1/2	6 1/4	6 9/16	5	11 5/16	12 1 1/16	7	7

Table 42. Dimensions of Merchant Marine Overboard Discharge Valves

(See Fig. 16. Crane Co.)



Size	2½	3	3½	4	4½	5	6	8	10	12	14	15
B	8½	9½	10½	11½	12	13	14	17	20	24	28	30
H	7	7½	8½	9	9¼	10	11	13½	16	19	21	22¼
J	1 1/16	¾	13/16	15/16	15/16	15/16	1	1⅛	1 3/16	1¼	1⅜	1⅜
W	....	11¾	....	14⅝	....	15⅞	..	..	23¾	27⅞	..	32¾

Table 43. Dimensions of Merchant Marine, Main Injection Valve

(See Fig. 18. Crane Co.)

Size			10	12	14	16
D			10¾	12¾	14¾	16⅞
E			11	12½	14	16
F			16	19	21	23½
G			1 3/16	1¼	1⅜	1 7/16
H			17½	20½	23	25½
J			1⅞	2	2⅛	2¼
R			18	20	24	27
T			28	34	38½	41½

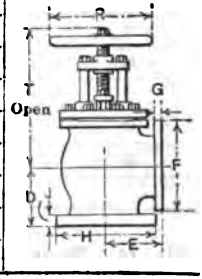
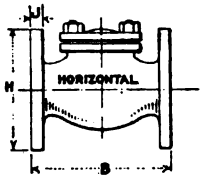
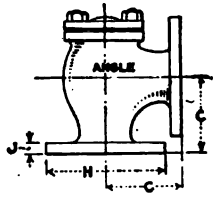


Table 44. Dimensions of Extra Heavy, Brass, Merchant Marine, Check Valves

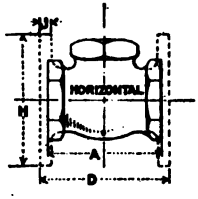


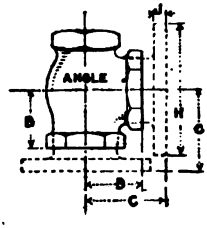


(See Fig. 19. Crane Co.)

Size.....	1½	2	2½	3
B	6⅜	7⅜	8¾	10
C	3¾	4⅛	4⅝	5⅛
H	6	6½	7½	8¼
J	9/16	⅝	1 1/16	¾
Centre to End Screwed.....	2¾	3⅛		

Table 45. Dimensions of Medium, Brass, Regrinding, Merchant Marine, Check Valves, Screwed and Flanged Types



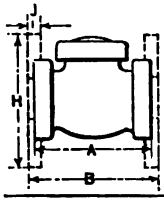


(See Fig. 20.)

Size	¼	⅜	½	¾	1	1¼	1½	2
A	1⅝	1 15/16	2 7/32	2¾	3 5/16	3⅞	4½	5¼
B	25/32	1	1 5/32	1⅜	1 21/32	1 15/16	2¼	2⅝
C	.....	.....	.....	2¼	2½	2⅞	3 5/16	3¾
D	.....	.....	.....	3 11/16	4 5/16	4 5/16	5 5/16	6
H	.....	.....	.....	3½	4	4½	5	6
J	.....	.....	.....	1 1/32	⅝	1 3/32	1 5/16	½

Table 46. Dimensions of Extra Heavy, Brass, Merchant Marine, Cushioned Check Valves, Screwed and Flanged Types

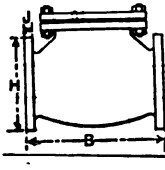
(See Fig. 21. Crane Co.)



Size	¼	⅜	½	¾	1	1¼	1½	2
A	1¾	2	2 15/32	2 15/16	3½	4 1/16	4⅝	5¾
B	.....	....	.....	3¾	4⅝	4 13/16	5½	6½
H	.....	.....	.....	4	4½	5	6	6½
J	.....	.....	.....	7/16	½	17/32	9/16	⅝

Table 47. Dimensions of Standard, Iron Body, Merchant Marine Swing, Check Valves

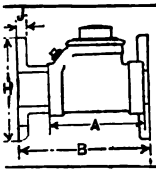
(See Fig. 22. Crane Co.)



Size	2½	3	4	5	6	8	10	12
B	10	11	13	15	16	18	22	26
H	7	7½	9	10	11	13½	16	19
J	1 1/16	¾	1 5/16	1 5/16	1	1⅛	1 3/16	1⅛

Table 48. Dimensions of Medium, Brass, Merchant Marine, Swing Check Valves

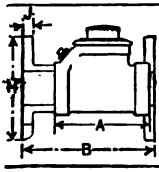
(See Fig. 23. Crane Co.)



Size	⅜	½	¾	1	1¼	1½	2
A	2 5/16	2 15/16	2¾	3 7/32	3 13/16	4¼	5 5/16
B	.....	.....	4⅜	5	5⅝	6⅛	7⅜
H	.....	.....	3½	4	4½	5	6
J	.....	.....	1 1/32	⅜	1 3/32	7/16	½

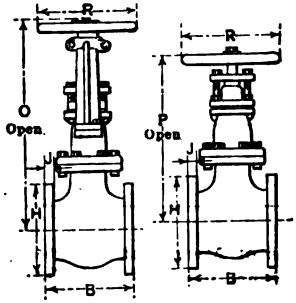
Table 49. Dimensions of Extra Heavy, Brass, Merchant Marine, Swing Check Valves, Screwed and Flanged

(See Fig. 23. Crane Co.)



Size	½	¾	1	1¼	1½	2
A	2 3/32	3 1/16	3 5/16	4⅛	4 13/16	5 13/16
B	.....	.....	5⅛	6¼	7 5/16	8⅜
H	.....	.....	4½	5	6	6½
J	.....	.....	½	1 7/32	9/16	⅝

Table 50. Dimensions of Standard Iron Body, Merchant Marine, Gate Valves, Rising and Non-rising Stems

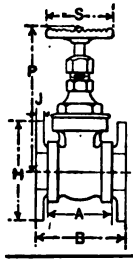


(See Figs. 24 and 25. Crane Co.)

Size.....	2	2½	3	3½	4	4½	5	6	7	8	9	10	12
B	7	7½	8	8½	9	9½	10	10½	11	11½	12	13	14
H	6	7	7½	8½	9	9¼	10	11	12½	13½	15	16	19
J	5/8	1 1/16	¾	1 3/16	1 5/16	1 5/16	1 5/16	1	1 1/16	1⅛	1⅛	1 3/16	1¼
O	14½	16	19	21¼	24	25½	28½	31¾	37¼	41	44¾	50	57¼
P	11¾	12¾	14¼	15¼	16¼	17⅝	19	20¾	23	26	28	30¼	35¼
R	6½	6½	7½	7½	9	9	10	12	12	14	14	16	18
End to End Screwed....	5 7/16	5⅞	6⅛										

Table 51. Dimensions of Standard, Brass, Merchant Marine, Gate Valves, Non-rising Stem, Screwed and Flanged Types

(See Fig. 26. Crane Co.)

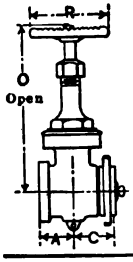


Size	¼	⅜	½	¾	1	1¼	1½	2
A	1 25/32	1 25/32	2 1/16	2 9/32	2 5/8	2 15/16	3 1/4	3 29/32
B	.....	.....	....	.....	3 3/8	3 7/8	4 3/8	5 1/2
H	.....	.....	....	.....	4	4 1/2	5	6
J	.....	.....	....	.....	5/8	13/32	7/16	½
P	3 9/16	3 9/16	4 1/16	4 1/16	5 1/16	6 1/16	7 1/4	8 3/4
S	1 3/4	1 3/4	2 1/16	2 9/16	2 3/4	3 1/16	3 5/8	4 1/16

Table 52. Dimensions of Merchant Marine, Standard, Brass, Hose Gate Valves

(See Fig. 27. Crane Co.)

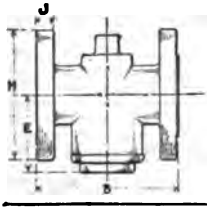
Valves furnished without cap and chain unless otherwise ordered.
Hose Threads made to standard specified on order.



Size	1	1¼	1½	2	2½
A	1 3/8	1 1/2	1 11/16	2	2 1/4
C	1 1/2	1 13/16	2 1/8	2 1/4	2 13/16
O	7 1/8	8 1/2	9 5/8	11 5/8	13 3/8
R	3 1/8	3 5/8	4 1/8	4 3/4	5 3/8

Table 53. Dimensions of Merchant Marine Extra Heavy, All Iron Stop Cocks, Screwed and Flanged

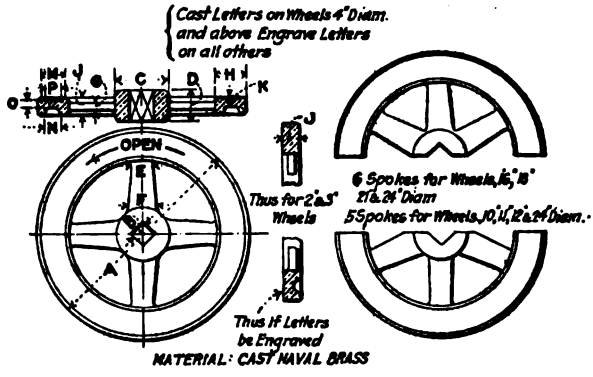
(See Fig. 28)



Size	1	1¼	1½	2	2½	3
A	3 1/16	3 15/16	4 1/4	5 1/4	6 5/8	8 1/8
B	.....	.....	6 1/2	7 1/2	8 1/2	10
E	2 15/16	3 1/4	3 9/16	4 3/8	5 1/8	6 3/16
H	.....	.....	6	6 1/2	7 1/2	8 1/4
J	.....	.....	13/16	7/8	1	1 1/8

Sizes 1 inch to 2 inch have screwed cap as shown
Sizes 2½ inch and 3 inch have bolted cap.

Table 54. Dimensions of Bureau of Steam Engineering Standard Handwheels


Cast Letters on Wheels 4" Diam. and above Engrave Letters on all others
OPEN
Thus for 2" & 5" Wheels
Thus If Letters be Engraved
6 Spokes for Wheels, 16, 18" 21 & 24" Diam
5 Spokes for Wheels, 10, 11, 12 & 24" Diam.
MATERIAL: CAST NAVAL BRASS

Light pressure

Diameter	Square	Hub		Spokes				Rim						
A	B	C	D	E	F	G	No.	H	J	K	M	N	O	P
2	9/32	5/8	3/8	1/4	5/16	1/4	4	1 1/32	1/4	..	..	....	....	1/4
3	9/32,	3/8	3/4	5/16	7/16	1/8	4	7/16	1/4	..	..	....	....	1/4
4	3/8,	7/16	1	3/8	9/16	5/16	4	5/8	3/8	1/32	7/16	5/16	5/32	1/4
5		7/16	1 1/8	5/8	7/16	5/8	4	5/8	3/8	1/32	7/16	5/16	5/32	1/4
6		9/16	1 5/16	1 1/16	1/2	1 3/16	1/4	5/8	1 3/32	1/32	7/16	5/16	5/32	1/4
7		9/16	1 5/16	3/4	9/16	3/4	1/4	1 1/16	1/2	1/32	7/16	3/8	5/32	1/4
8	9/16,	1 1/16	1 1/2	7/8	5/8	1 3/16	5/16	1 1/16	1 7/32	1/16	7/16	3/8	3/16	1/4
9		1 1/16	1 5/8	1 5/16	5/8	7/8	5/16	3/4	9/16	1/16	7/16	1 3/32	3/16	1/4
10		3/4	1 3/4	1	1 1/16	1 5/16	3/8	1 3/16	1 9/32	1/16	9/16	1 3/32	3/16	3/8
11		1 3/16	2	1 1/16	3/4	1	3/8	7/8	2 1/32	1/16	9/16	1/2	3/16	3/8
12	1 3/16,	1 5/16	2 1/8	1 1/8	3/4	1 1/16	3/8	1 5/16	1 1/16	1/16	5/8	1/2	3/16	3/8
14	1 5/16,	1	2 1/8	1 5/16	7/8	1 3/16	7/16	1 1/8	2 7/32	3/32	5/8	5/8	1/4	3/8
16	1,	1 1/16	2 3/8	1 7/16	1 1/16	1 5/16	1/2	1 1/4	1 5/16	3/32	3/4	1 1/16	1/4	3/8
18		1 3/16	2 1/2	1 5/8	1 1/16	1 7/16	9/16	1 3/8	1 1/16	3/32	3/4	3/4	5/16	1/2
21	1 3/8,	1 1/2	2 5/8	1 7/8	1 3/16	1 5/8	5/8	1 5/8	1 3/16	1/8	1	1 1/16	5/16	1/2
24		1 1/2	3	2 1/16	1 5/16	1 3/4	1 1/16	1 13/16	1 5/16	1/8	1	1 1/16	5/16	1/2

Heavy pressure

Diameter	Square	Hub		Spokes				Rim						
A	B	C	D	E	F	G	No.	H	J	K	M	N	O	P
3	9/32	3/4	7/16	5/16	7/16	1/8	4	7/16	1/4	..	..	....	..	1/4
4	3/8	1	1/2	3/8	9/16	5/16	4	5/8	3/4	1/32	7/16	5/16	5/32	1/4
5		1 1/8	5/8	7/16	5/8	5/16	4	5/8	3/8	1/32	7/16	5/16	5/32	1/4
6	7/16,	1 5/16	1 1/16	1/2	1 1/16	1/4	4	5/8	1 3/32	1/32	7/16	5/16	5/32	1/4
7	9/16	1 5/16	3/4	9/16	3/4	1/4	4	1 1/16	1/2	1/32	7/16	3/8	5/32	1/4
8	1 1/16	1 1/2	7/8	5/8	1 3/16	5/16	4	1 1/16	1 7/32	1/16	7/16	3/8	3/16	1/4
9	3/4	1 5/8	1 5/16	5/8	7/8	5/16	4	3/4	9/16	1/16	7/16	1 3/32	3/16	1/4
10	1 3/16,	1 5/16	1 3/4	1	1 1/16	1 5/16	3/8	1 3/16	1 9/32	1/16	9/16	1 3/32	3/16	3/8
11	1 5/16	2	1 1/16	3/4	1	3/8	5	7/8	2 1/32	1/16	9/16	1/2	3/16	3/8
12	1	2 1/8	1 1/8	3/4	1 1/16	3/8	5	1 5/16	1 1/16	1/16	5/8	1/2	3/16	3/8
14	1 1/4,	1 3/8	2 3/4	1 5/16	7/8	1 3/16	7/16	1 1/8	2 7/32	3/32	5/8	5/8	1/4	3/8
16	1 3/8,	1 1/2	2 7/8	1 7/16	1 5/16	1 5/16	1/2	1 1/4	1 5/16	3/32	3/4	1 1/16	1/4	3/8
18		1 3/4	3 1/8	1 5/8	1 1/16	1 7/16	9/16	1 3/8	1 1/16	3/32	3/4	3/4	5/16	1/2
21	1 7/8,	2	3 3/8	1 7/8	1 3/16	1 5/8	5/8	1 5/8	1 3/16	1/8	1	1 5/16	5/16	1/2
24		2 1/4	3 7/8	2 1/16	1 5/16	1 3/4	1 1/16	1 13/16	1 5/16	1/8	1	1 1/16	5/16	1/2

SAFETY AND RELIEF VALVES

Marine Use. Safety and relief valves are employed to prevent pressure from rising above the designed working pressure in the various units and pipe lines. **Safety valves** are used for vapors and gases, such as steam and air. **Relief valves** are used for fluids such as water and oil.

Huddling or Pop Chambers. A safety valve should have an adjustable pop or huddling chamber located between the valve seat and the atmosphere. Its **function** is to give the valve a popping action; that is, when the valve relieves, the escaping steam is temporarily confined in this chamber before escaping to the atmosphere. This temporary confinement increases the upward load against the spring, thereby causing the disk to lift higher; and after a predetermined drop in pressure in the boiler, the valve closes with a slight snap.

The adjustable huddling chamber should be **concentric with the disk** in all positions. The adjustable mechanism should be located preferably so as to be free from corrosion and scale.

Relief valves do not require a huddling chamber of the same type as a safety valve, the energy of the moving liquid being utilized to assist the valve in lifting, and so help in reducing the drop in pressure before closure.

Many types of water relief valves have no huddling chamber, and, owing to their simplicity, are preferred to the more complex types. They lose considerable pressure before closing.

Snifting and cylinder relief valves, for a mixture of steam and water, require a different form of huddling chamber than do valves for steam and cold liquids. When called upon to act, they must give instant relief to a considerable quantity of steam and water.

Springs. For H.P. marine work **outside springs** are used; **inside springs,** where used with steam, should be encased; for compressed air pops, either inside or outside springs are used; for water relief valves inside springs should be used wherever the relieving and closing pressures are important and must be maintained. Cylinder relief valves should have enclosed springs. **Ammonia relief valves,** and valves for similar work, should be of the enclosed type to prevent the escape of fumes, and the valves should be all iron or steel.

The **length and number of free coils** should be such that there is no danger of the spring being compressed solid while the valve is discharging under over-pressure conditions. Springs for steam and hot water service should be capable of being compressed solid and should return to within 0.03 in. of original length. The **pitch diam. of safety valve springs = 5 times diam. of stock.**

The valve being set, the spring must have sufficient **clearance between coils** to allow further compression equal to 10 per cent of valve seat diameter. Steel springs for steam should be drawn at a temperature of about 600 deg. F. (500 deg. minimum). **The ends** should be ground absolutely square with the centre line of spring.

Stems. Single piece give the best service. Pop valves for steam, air and high-pressure water should have a single stem extending from the disk to the outside of valve bonnet. To prevent bending, crystallization, and wear at the point, safety valve stems should be of steel.

The design of huddling chamber governs the **location of stem end in the disk.** Locating the stem end a distance equal to ¼ diam. of valve below the bearing on the seat gives good results, although some types of flat seat safety valves with the stem end above the seat bearing are giving good results.

For **water relief valves**, the lower the stem the longer the valve keeps tight in service.

Disk Guides are of integral type that assists the disk to find its seat. The multiple wing or **feather guide** is the best type, suffering less from rapid wear than the single round or centre guide. Where scale forming solids are present in oil, exhaust steam or water, centre **pin guides** are preferred.

Clearance between guides and bore of valve bushing of steam valves should be approximately 0.01 in. Water and oil valves may have more clearance. The valve bushing bore and the wing guides must have an absolutely smooth finish to prevent rapid wear, chattering and leakage.

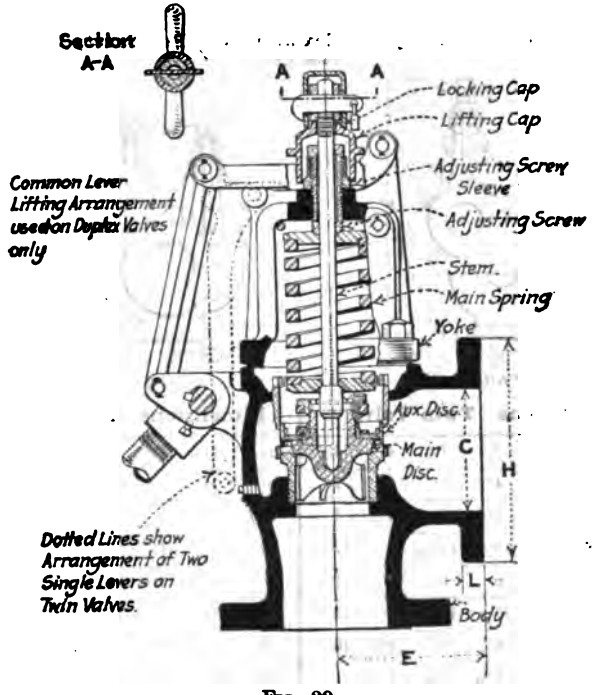


FIG. 29.

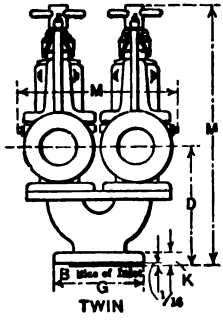
The Relieving Capacity of conventional types of water relief valves do not vary much. On the contrary, the relieving capacity of pop safety valves of the same size, but of different designs, vary widely. (Tests made by P. G. Darling, member A.S.M.E., also by M. W. Link, Valve World, May, 1914.)

The general results of both these tests were the same, viz., a wide variation in relieving capacities for different designs was demonstrated; two 4-inch valves differed by as much as 600 per cent. **The relieving capacity is directly proportional to the lift.**

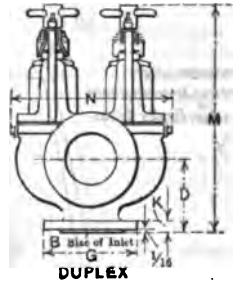
This feature was recognised by the Boiler Code Committee of the A.S.M.E. in the formation of their National Boiler Code. The Board of Supervising Inspectors recognise no fixed relieving capacities for safety valves.

Boiler Safety Valves must be designed to meet with the requirements of the Board of Supervising Inspectors, the U. S. Shipping Board, and, generally, to comply with the requirements of Lloyd's register of Shipping and the American Bureau of Shipping. A high lift, large capacity pop valve is always desirable. A valve which complies with all of the above requirements is shown in Fig. 29. A 3-in. duplex valve of this design has a discharging capacity of 26,500 lbs. of steam per hour at 200 lbs. boiler press. and 3 per cent overpressure rating.

Table 55. Iron Body, Duplex and Twin Pop Safety Valves
(Crane Co.)



TWIN



DUPLEX

Dimensions—Twin Valve

Size...	2	2½	3	3½	4	4½
B	3	3½	4½	5	6	7
C	2½	3	3½	4	4½	5
D	11⅝	12 9/16	13 15/16	15 1/16	16½	17 7/16
E	4¾	5½	6	6½	7	7½
G	8¼	9	10½	11	12½	14
H	7	7½	8½	9	9¼	10
K	1⅛	1 3/16	1 5/16	1⅜	1 7/16	1½
L	1 1/16	¾	1 3/16	1 5/16	1 5/16	1 5/16
M	22½	26¾	30	32	34¾	37½
N	14	16	17¼	19	20¾	22½

Dimensions—Duplex Valve

Size...	2	2½	3	3½	4	4½
B	3	3½	4½	5	6	7
C	3½	4	5	6	7	8
D	6½	7¼	7⅞	8⅝	9	10
E	4½	5½	6½	7½	7¾	8¾
G	8¼	9	10½	11	12½	14
H	8½	9	10	11	12½	13½
K	1⅛	1 3/16	1 5/16	1⅜	1 7/16	1½
L	1 3/16	1 5/16	1 5/16	1	1 1/16	1⅛
M	18	22	24½	26½	28½	34½
N	13	16	17¼	19	20¾	22½

Twin Valves, that is, two single safety valves mounted on a twin base, are the safest and best type. **Duplex valves**, that is, two sets of trimmings in one body, are popular for marine use because but one joint is required between the valve and boiler. Being more liable to warp than twin valves, they are

more difficult to keep tight. Triplex and quadruplex safeties are not used in marine work.

The spring and disk of all steam safeties should have **outside lifting gear** so that the disk may be raised from its seat by this outside lever an amount equal to ¼ valve diam. when working under 75 per cent popping pressure. Lloyd's Rules require valve arrangement so that disk may be ground by turning stem from the outside of valve, and a device that prevents spring from being compressed beyond the original set pressure, when the valve is locked. This last should be standard for all H.P. safeties.

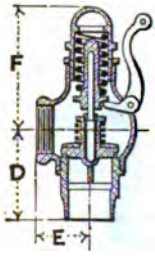


FIG. 30.

Table 56. Dimensions for Fig. 30

Size	1	1¼	1½	2
D	2 1⁵⁄₁₆	3⅛	3¼	4
E	1⅝	1 1⁵⁄₁₆	2	2½
F	3 1⁵⁄₁₆	4⅞	5¼	6

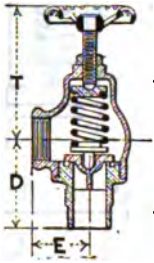


FIG. 31.

Table 57. Dimensions for Fig. 31

Size	¾	1	1¼
D	2⅝	2¾	3¼
E	1¾	1¾	1¾
T	3½	4	4½

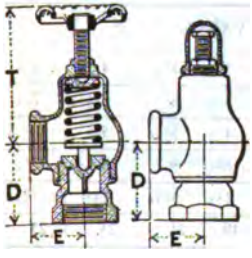


FIG. 32. FIG. 33.

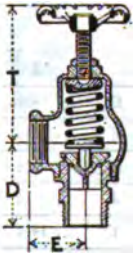


FIG. 34.

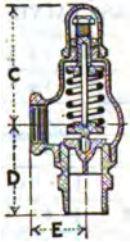


FIG. 35.

Table 58. Dimensions for Figs. 32 and 33

Size	¾	1	1¼	1½	2
D	2⅝	2⅝	3¼	3½	4
E	1⅜	1⅝	1¾	2	2½
T	3½	4¼	4½	5¼	6½

Table 59. Dimensions for Fig. 34

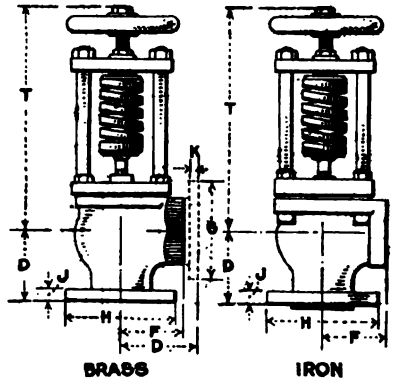
Size	¾	1	1¼	1½	2
D	2⅛	2½	2 1³⁄₁₆	3 1⁄₁₆	3½
E	1¼	1⅝	1 1⁵⁄₁₆	1 1⁵⁄₁₆	2⅛
T	3½	4	4½	5	6½

Table 60. Dimensions for Fig. 35

Size	¾	1	1¼	1½	2
C	3 3⁄₁₆	3½	4 5⁄₁₆	5	6
D	2⅝	2⅝	3¼	3½	4
E	1⅜	1⅝	1¾	2	2½

Table 61. Relief Valve for H.P. Service Outside Spring and Yoke, Brass and Iron Body

(See Fig. 36)



DIMENSIONS

BRASS VALVE			IRON VALVE						
Size.....	1½	2	Size.....	2	2½	3	3½	4	4½
D	3⅜	4	D	4 5/16	4¾	5⅜	6	6 11/16	7⅛
F	2¾	3	F	3⅜	4½	4⅞	5 7/16	5¾	6 5/16
G	5	6	H	6½	7½	8¼	9	10	10½
K	7/16	½	J	⅞	1	1⅛	1 3/16	1¼	1 5/16
H	6	6½	T	14¾	16¾	19	21	23	25
J	9/16	⅝							
T	11½	14½							

Some features essential to **good safety valve design,** see Fig. 29, are: A locking cap (19) to prevent set pressure being changed by unauthorised person; spring washer (9) should attack in such manner as to assure proper distribution of load; spring (10) and lifting gear should permit disc (3) to be raised $\frac{3}{8}$ of valve diam. drop forged steel stem (13) bearing $\frac{1}{4}$ of valve diam. below seat bearing; spring (10) placed where air cooled and spring chamber open to atmosphere to prevent back pressure on valve disc; steam outlet expanded to permit easy outflow of expanded steam; main disc (3) guided aboue and below valve seat to prevent cromping and sticking; auxiliary device (5, 6, 8, 9) permits high lift with small blow-back and easy seating.

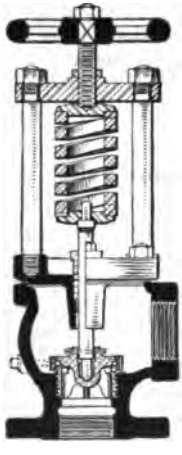


FIG. 36.

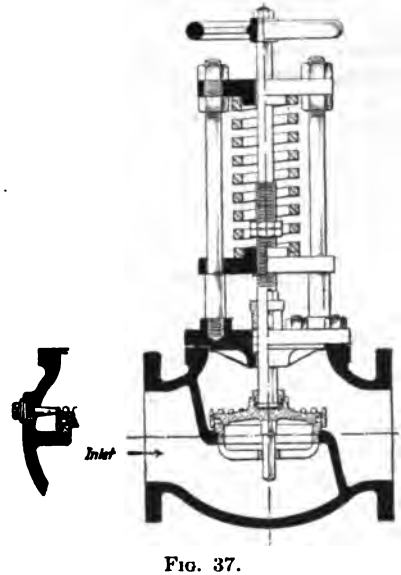


FIG. 37.

Brass Safeties for saturated steam, not for boilers, are generally made with a male thread on the base. (See Fig. 30.) Flanged and female-threaded bases should be avoided as the former spring easily when bolted into place, and the latter leak after short service. The general design of brass pop should follow that of the iron valve. Brass pops should not be used for high pressure work in sizes over 2 inches.

Relief Valves for Hot Water. For high pressure the design shown in Fig. 31, with adjustable blow-down is the best. For medium pressures and small sizes, the design shown in Figs. 31 to 34 inc. is recommended. The blowdown is about 20 per cent of set pressure. **Snifting Valves** must be extremely simple. Fig. 35 shows a design that has been in successful use for over 25 years.

Water Relief valves of the **outside spring type** are desirable, as the spring is not exposed to the action of salt water. They are made of brass or iron, as required. The stuffing box type, as shown in **Fig. 36**, is preferable. Because of their simplicity the **enclosed spring type** relief valves are more common in the smaller sizes. These valves are designed with different types of seats for different service, such as, steam, air, or water. Valves of this type cannot be used as pop valves on boilers, as they do not comply with the regulations for that service.

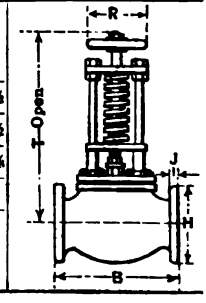
Back pressure valves for Marine Service are of the outside spring type. All working surfaces of these valves should be bronze so as to prevent sticking. A type is shown in Fig. 37. For this service a valve should contain the following features for successful operation: (1) Outside spring; (2) handwheel control; (3) stuffing box on vacuum side; (4) adjustable back pressure; (5) huddling chamber to give popping action.

Testing Valves. Precautions: (1) Be sure the test gage is accurate; (2) see that the valve is clean—absolutely clean; (3) test water valves with water and steam valves with steam or air (not water); (4) thoroughly understand the valve being tested; (5) when testing have enough steam or water to meet approximately the discharging capacity of valve under test; (6) do not lift disk of a steam safety from its seat when cold; (7) always install safety and relief valves with the stem in vertical position.

Table 62. Dimensions of Back Pressure Valve

(See Fig. 37)

Size.....	2	2½	3	3½	4	4½	5	6	7	8
B	..	9½	10	11	12	13	13¾	16	..	18½
H	..	7	7½	8½	9	9¼	10	11	..	13½
J	..	1 5/16	¾	1 3/16	1 5/16	1 5/16	1 5/16	1	..	1¼
R	..	5	6½	6½	7½	9	10	10	..	12
T	..	14½	19	21¾	24¾	27	28	30	..	36



Materials of Valve Parts

Kind of valve	Body and bonnet	Stem	Disk	Valve seat	Spring
Duplex safety superheated steam	Cast steel	Forged steel	Monel metal	Monel metal	Crucible spring steel
Duplex safety saturated steam	Cast iron	Forged steel	Hard bronze	Monel metal	Crucible spring steel
L.P. brass safety	Comp.	Rolled bronze	Comp.	Comp.	Spring steel
Hot-water relief valve	Comp.	Rolled bronze	Comp.	Comp.	Spring steel
Cold-water relief valve	Comp.	Rolled bronze	Comp.	Comp.	Spring steel
Snifting and cylinder relief valve	Comp.	Rolled bronze	Comp.	Comp.	Spring steel
Air-relief valve	Comp.	Rolled bronze	Comp.	Comp.	Spring steel
Back-pressure steam valve	Cast iron	Rolled bronze	Comp.	Comp.	Spring steel

Hard bronze to be equal to U. S. Navy gun metal.
Composition to be equal to U. S. Navy valve bronze.
Rolled bronze to be equal to rolled Tobin Bronze.
Crucible steel spring to be drawn at 600 deg. F. or more.
Spring-steel spring to be drawn at 400 deg. F. or more.

REDUCING VALVES AND PUMP GOVERNORS

BY

FRANK W. STERLING

REDUCING VALVES

(See also p. 1001)

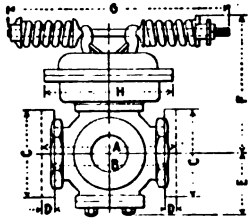
Many of the auxiliary machines and pipe lines on shipboard must operate at a pressure reduced from boiler pressure. This reduction is accomplished by inserting in the steam supply lines **pressure reducing valves** which wire-draw the steam.

Types. Pressure reducing valves are of two types, **inside spring** type and **outside spring** type; the essential difference lies in the fact that adjustment of the spring is not practical in the inside spring type and the valve must be designed for its approximate working conditions. On the contrary, the outside spring type allows of wide adjustment of its spring tension and the *delivery pressure* may thereby be controlled through a wide range.

The Outside Spring Type reducing valve, Fig. 1, operates as follows: Steam enters valve at *A* and flowing in the direction indicated by the arrows, passes out at *B*. In its course it enters chamber *D*, through port *E*, closing valve 4 against the opposing power of the springs 34. Any increased pressure on the diaphragms overcomes the resistance of the springs, lifting valve 4 toward its seats. Should the delivery or reduced pressure decrease, the springs overcome the pressure on the diaphragms and force valve 4 open. An equilibrium is thus instantly established. *The desired delivery pressure* is controlled by adjusting nuts 36.

Table 1. Pressure Reducing Valve, Outside Spring Standard 125 Lbs. and Under

(Foster Engineering Co.)



Size of Valve	A	B	C	D	E	F	G	H	Approximate net weights, pounds	
	Dist. bet. faces flanged	Dist. bet. faces screwed	Diam. of flanges	Thickness of flanges	Center to bottom	Center to top	Length spring bolt	Diam. of hood	Screwed	Flanged
¼	6	3	3½	7/16	1⅞	4⅛	4⅝	4⅜	4½	10
⅜	5	4¼	3¾	7/16	2¾	5½	12	6 3/16	12½	14
1	5 5/16	4⅝	4	½	2⅞	5½	12	6 3/16	14	17
1¼	5½	5¼	4½	9/16	3⅜	6¾	12	7	22	25
1½	6	5⅛	5	9/16	3½	7	12	8	27	33
2	7½	6½	6	⅝	4¼	9	15¼	10	49	59
2½	9	7½	7	11/16	4⅝	10⅛	15¼	10	76	86
3	10	8¾	7½	¾	5	10½	15¼	10	81	96
3½	11	10½	8½	13/16	5½	11	15¼	10	94	112
4	12	11½	9	1	6½	12	21½	12⅛	151	174
4½	14	12½	9¼	1 1/16	6½	12	21½	12¼	165	197
5	15	14¼	10	1⅛	6¾	12½	21½	12¼	190	206
6	17	14½	11	1 3/16	7½	13½	29½	14	279	290
7	18¼	....	12½	1¼	9¼	14	30	14⅛	.....	410
8	20¾	....	13½	1 5/16	9¼	14½	30	14⅛	.....	483
9	23½	....	15	1⅜	10½	15	30	14⅛	.....	592
10	23½	....	16	1 7/16	10½	15	30	14⅛	.....	630
12	27¼	....	19	1 9/16	12¼	16½	30	16¾	.....	1043
14	32	....	21	1 11/16	14½	22	30	19	.....	1490
16	35	....	23½	1 13/16	16	23	34	19½	.....	1615
18	37½	....	25	1⅞	16¾	23¾	34	21	.....	1968
20	48½	....	27½	2 1/16	21	26	36	22	.....	3215

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Types. Pressure reducing valves are of two types, **inside spring** type and **outside spring** type; the essential difference lies in the fact that adjustment of the spring is not practical in the inside spring type and the valve must be designed for its approximate working conditions. On the contrary, the outside spring type allows of wide adjustment of its spring tension and the *delivery pressure* may thereby be controlled through a wide range.

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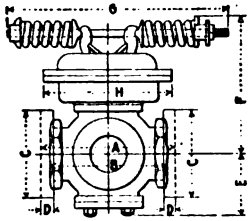


Size of Valve	A	B	C	D	E	F	G	H	Approximate net weights, pounds	
	Dist. bet. faces flanged	Dist. bet. faces screwed	Diam. of flanges	Thickness of flanges	Center to bottom	Center to top	Length spring bolt	Diam. of hood	Screwed	Flanged
¼	6	3	3½	7/16	1⅞	4⅛	4⅝	4¾	4½	10
¾	5	4¼	3¾	7/16	2¾	5½	12	6 3/16	12½	14
1	5⅜	4⅜	4	½	2⅞	5½	12	6 3/16	14	17
1¼	5½	5¼	4½	9/16	3⅜	6¼	12	7	22	25
1½	6	5¼	5	9/16	3½	7	12	8	27	33
2	7½	6½	6	⅝	4¼	9	15¼	10	49	59
2½	9	7½	7	11/16	4⅝	10⅛	15¼	10	76	86
3	10	8¾	7½	¾	5	10⅛	15¼	10	81	96
3½	11	10½	8½	13/16	5½	11	15¾	10	94	112
4	12	11½	9	1	6½	12	21½	12¼	151	174
4½	14	12½	9¼	1 1/16	6½	12	21½	12¼	165	197
5	15	14¼	10	1⅛	6¾	12½	21½	12¼	190	206
6	17	14½	11	1 3/16	7½	13½	29½	14	279	290
7	18¼	....	12½	1¼	9¼	14	30	14⅛	.....	410
8	20¾	....	13½	1 5/16	9¼	14½	30	14⅛	.....	483
9	23½	....	15	1⅜	10½	15	30	14⅛	.....	592
10	23½	....	16	1 7/16	10½	15	30	14⅛	.....	630
12	27¼	....	19	1 9/16	12¼	16½	30	16¾	.....	1043
14	32	....	21	1 11/16	14½	22	30	19	.....	1490
16	35	....	23½	1 13/16	16	23	34	19½	.....	1615
18	37½	....	25	1⅞	16¾	23¾	34	21	.....	1968
20	48½	....	27½	2 1/16	21	26	36	22	.....	3215

Number	Name of part
1	Body
2	Upper Valve-seat
3	Lower Valve-seat
4	Main valve (or clapper)
5	Bottom flange (or plug)
6	Valve (or clapper) stem
7	Valve stem nu'
8	Valve stem Jamb-nut
9	Valve stem Jamb-nut cotter-pin
10	Bottom and top flange screws
11	Gasket
12	Top
13	Top liner
14	Valve stem levers
15	Valve stem pin

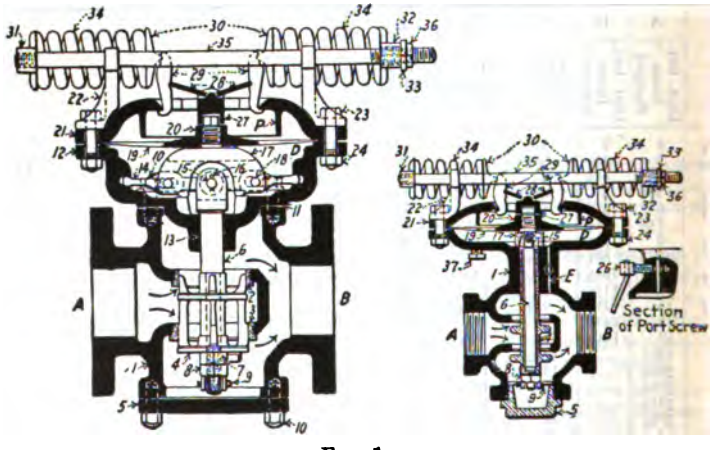


FIG. 1.

16	Cotter pin for 15
17	Diaphragm centre
18	Diaphragm centre pin
19	Diaphragm (set—consisting of one or more)
20	Diaphragm Jamb-nut
21	Hood
22	Spring bolt bracket
23	Spring bolt bracket bolts
24	Spring bolt bracket and hood bolt nuts
25	Hood bolts
26	Port screw
27	Toggle lever base
28	Toggle lever
29	Toggle lever links
30	Pivot washer
31	Plain yoke
32	Lock yoke
33	Lock-yoke wrench
34	Springs (two to a set)
35	Spring bolts
36	Spring bolt nuts
37	Diaphragm chamber plug

Table 2. Pressure Reducing Valves, Outside Spring Extra Heavy, 125 to 250 Lbs. (See Fig. in Table 1.)

A	B	C	D	E	F	G	H	Approximate net weights, pounds	
Dist. bet. faces flanged	Dist. bet. faces screwed	Diam. of flanges	Thickness of flanges	Center to bottom	Center to top	Length spring bolts	Diam. of hood	Screwed	Flanged
6	3	3½	9/16	1⅞	4⅛	4⅝	4⅜	4½	11
5	4¼	3¾	9/16	2¾	5½	12	6 9/16	12½	17
5¾	4⅝	4½	⅝	2⅞	5½	12	6 9/16	14	21
5½	....	5	¾	3⅛	6¾	12	7	....	30
6	....	6	13/16	3½	7	12	8	....	40
7½	....	6½	⅞	4¼	9	15¼	10	....	68
9	....	7½	1	4⅝	10⅛	15¼	10	....	92
10	....	8¼	1⅛	5	10½	15¼	10	....	138
11¾	....	9	1 3/16	5½	11	15¼	10	....	152
12½	....	10	1¼	6½	12	21½	12¼	....	201
14	....	10½	1 5/16	6½	12	21½	12¼	....	213
15	....	11	1⅜	6¾	12½	21½	12¼	....	235
17	....	12½	1 7/16	7½	13½	29½	14	....	321
18¼	....	14	1½	9¼	14	30	14½	....	450
20¾	....	15	1⅝	9¼	14½	30	14½	....	528
23½	....	16	1¾	10½	15	30	14½	....	640
23½	....	17½	1⅞	10½	15	30	14½	....	685
27¼	....	20	2	12¼	16½	30	16¾	....	1116
32	....	22½	2⅛	14½	22	30	19	....	1503
35	....	25	2¼	16	23	34	19½	....	1732
37½	....	27	2⅜	16¾	23¾	34	21	....	2124
48½	....	29½	2½	21	26	36	22	....	3380

Table 3. Pressure Reducing Valve, Inside Spring
(Foster Engineering Co.)

Sizes	Diam. of flanges	Dist. between faces	Dist. between screwed ends	Thickness of flanges	Center to bottom	Center to top	Net weights	
							Screwed	Flanged
All composition								
½	3½	5½	4	9/16	2¼	9⅛	11	15
¾	3¾	5½	4⅛	9/16	2¼	9⅛	13½	18
1	4	6	4½	⅝	2½	9⅞	14	19
1¼	4½	6½	4⅞	⅝	2½	10	17	23
1½	5	7	5	⅝	2¾	10¼	19	29
2	6	7½	6¼	⅝	3½	10¾	25	37
Iron body—composition trimmed								
2½	7½	10	8½	1	5¼	11½	65	73
3	8¼	10½	9¾	1⅛	6	11⅞	83	98
3½	9	11½	10¾	1 3/16	6½	12½	97	120
4	10	12½	11½	1¼	6⅝	13½	128	160
4½	10½	14	12½	1 5/16	7⅛	15¼	158	195
5	11	15	13¾	1⅜	7½	15¼	185	220
6	12½	17	14½	1 7/16	8⅝	16⅜	272	320
7	14	18¼	....	1½	10⅜	19¼	....	490
8	15	20¾	....	1⅝	11½	20¼	....	600
9	16	23	....	1¾	12⅛	21	....	900
10	17½	23½	....	1⅞	12⅝	21¾	....	1020
12	20	27¼	....	2	15	28½	....	1240

The Inside Spring Type reducing valve, Fig. 2, operates as follows: Steam entering at *A* passes through main valve port *H* to outlet *B*. The

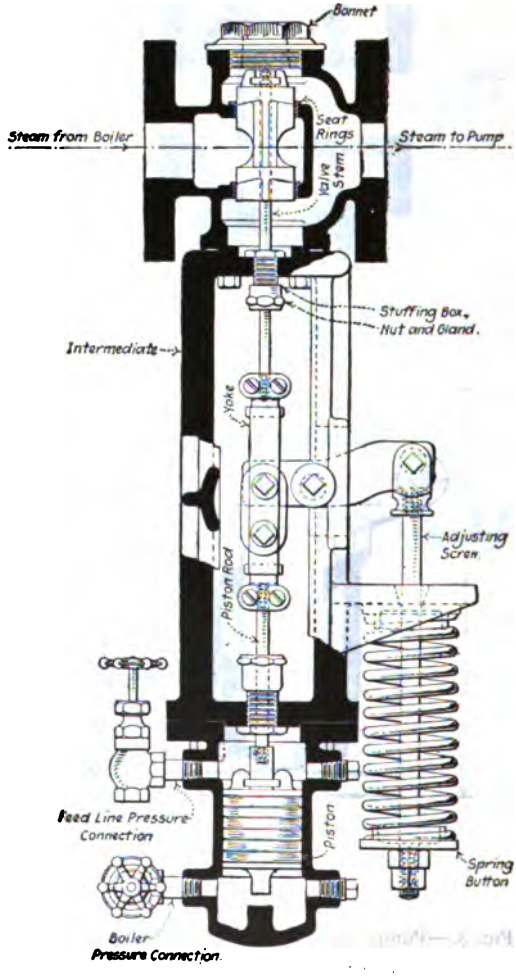


FIG. 4.—Pump Governor, Excess Pressure Type.

Auxiliary Operated Reducing Valve. The valve shown in Fig. 1 is designated a **direct operated reducing valve;** steam pressure acts on the diaphragm which is direct connected to the main valve and main springs. Fig. 2 illustrates an **auxiliary operated reducing valve;** in this case steam acts on the diaphragm but this is connected to the auxiliary valve 14, which auxiliary in turn controls steam to the piston 2 which operates the main valve. The **advantage of the auxiliary operated** valve is that very close regulation can be effected with very small movement of the auxiliary valve.

PUMP GOVERNORS

Purpose. Pump governors are essentially special forms of pressure regulators and are frequently designated as **"pump pressure regulators."** They are used to so regulate the pump operation that it will deliver the desired discharge pressure.

Types. There are two general types, the **constant pressure** type and the **excess pressure** type. The former is employed where it is desired to limit the discharge pressure of a pump. The excess pressure type is employed where it is desired to maintain a governing excess of pressure in the steam ends of a pump regardless of any increased hydraulic resistance the pump may encounter; these are the conditions met in boiler feeding. A balanced valve is an essential feature in pump governors.

Constant Pressure Governor. Fig. 3 shows an excellent constant pressure type, see Table 4. Monel valve seats, stems and pistons are desirable. The piston is under pump discharge pressure through connection at bottom of governor. The governor spring being set to the desired discharge pressure, any excess discharge pressure over this spring setting will operate to close the main valve; when the discharge pressure falls below the desired pressure the spring will open the main valve, thus admitting steam to the pump. The spring setting may be changed while the governor is in operation.

The Excess Pressure Governor, Fig. 4, consists of a valve direct connected to piston operating in a cylinder. One side of this piston is connected to boiler pressure and the other side is connected to the pump discharge pressure. The governor spring is set for the excess pressure in the discharge line, and this spring assists the feed line pressure so that the governor valve cannot be closed until the pump discharge pressure exceeds the boiler pressure by the desired amount.

Pump governors are made to operate with diaphragms in place of the auxiliary piston but they are not as successful as the piston operated type and are not approved for **Naval practice.** It is advisable to so fit a lever that the governor may be choked open; this can be used in lieu of a governor by-pass.

REDUCING VALVES AND PUMP GOVERNORS

BY

FRANK W. STERLING

REDUCING VALVES

(See also p. 1001)

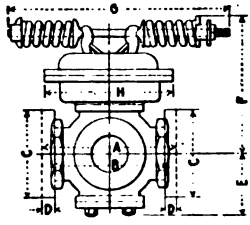
Many of the auxiliary machines and pipe lines on shipboard must operate at a pressure reduced from boiler pressure. This reduction is accomplished by inserting in the steam supply lines **pressure reducing valves** which wire-draw the steam.

Types. Pressure reducing valves are of two types, **inside spring** type and **outside spring** type; the essential difference lies in the fact that adjustment of the spring is not practical in the inside spring type and the valve must be designed for its approximate working conditions. On the contrary, the outside spring type allows of wide adjustment of its spring tension and the *delivery pressure* may thereby be controlled through a wide range.

The Outside Spring Type reducing valve, Fig. 1, operates as follows: Steam enters valve at *A* and flowing in the direction indicated by the arrows, passes out at *B*. In its course it enters chamber *D*, through port *E*, closing valve 4 against the opposing power of the springs 34. Any increased pressure on the diaphragms overcomes the resistance of the springs, lifting valve 4 toward its seats. Should the delivery or reduced pressure decrease, the springs overcome the pressure on the diaphragms and force valve 4 open. An equilibrium is thus instantly established. *The desired delivery pressure* is controlled by adjusting nuts 36.

Table 1. Pressure Reducing Valve, Outside Spring Standard 125 Lbs. and Under

(Foster Engineering Co.)



Size of Valve	A	B	C	D	E	F	G	H	Approximate net weights, pounds	
	Dist. bet. faces flanged	Dist. bet. faces screwed	Diam. of flanges	Thickness of flanges	Center to bottom	Center to top	Length spring bolt	Diam. of hood	Screwed	Flanged
¼	6	3	3½	7/16	1⅞	4½	4⅜	4⅜	4½	10
¾	5	4¼	3¾	7/16	2¾	5½	12	6 3/16	12½	14
1	5 3/4	4⅝	4	½	2⅞	5½	12	6 3/16	14	17
1¼	5½	5¼	4½	9/16	3⅛	6¾	12	7	22	25
1½	6	5¼	5	⅝	3½	7	12	8	27	33
2	7½	6½	6	⅝	4¼	9	15¼	10	49	59
2½	9	7¼	7	11/16	4⅝	10½	15¼	10	76	86
3	10	8¾	7½	¾	5	10½	15¼	10	81	96
3½	11	10½	8½	13/16	5½	11	15¼	10	94	112
4	12	11½	9	1	6½	12	21½	12¼	151	174
4½	14	12½	9¼	1 1/16	6½	12	21½	12¼	165	197
5	15	14¼	10	1⅛	6¾	12½	21½	12¼	190	206
6	17	14½	11	1 3/16	7½	13½	29½	14	279	290
7	18¼	....	12½	1¼	9¼	14	30	14½	.....	410
8	20¾	....	13½	1 5/16	9¼	14½	36	14½	.....	483
9	23½	....	15	1⅜	10½	15	30	14½	.....	592
10	23½	....	16	1 7/16	10½	15	30	14½	.....	630
12	27¼	....	19	1 9/16	12¼	16½	30	16¾	.....	1043
14	32	....	21	1 11/16	14½	22	30	19	.....	1490
16	35	....	23½	1 11/16	16	23	34	19½	.....	1615
18	37½	....	25	1¾	16¾	23¾	34	21	.....	1968
20	48½	....	27½	2⅛	21	26	36	22	.....	3215

Number	Name of part.
1	Body
2	Upper Valve-seat
3	Lower Valve-seat
4	Main valve (or clapper)
5	Bottom flange (or plug)
6	Valve (or clapper) stem
7	Valve stem nu'
8	Valve stem Jamb-nut
9	Valve stem Jamb-nut cotter-pin
10	Bottom and top flange screws
11	Gasket
12	Top
13	Top liner
14	Valve stem levers
15	Valve stem pin

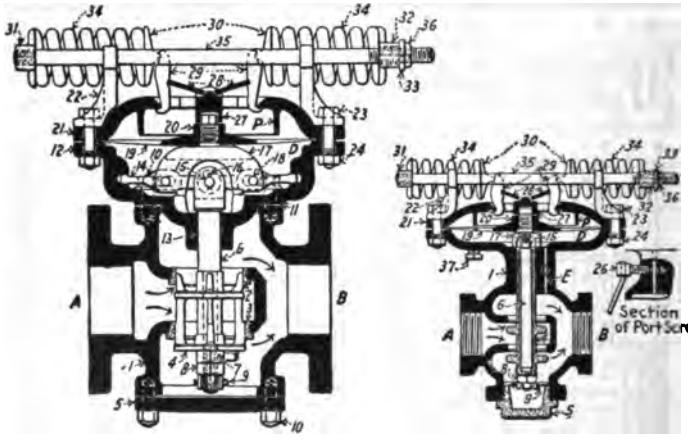


FIG. 1.

16	Cotter pin for 15
17	Diaphragm centre
18	Diaphragm centre pin
19	Diaphragm (set—consisting of one or more)
20	Diaphragm Jamb-nut
21	Hood
22	Spring bolt bracket
23	Spring bolt bracket bolts
24	Spring bolt bracket and hood bolt nuts
25	Hood bolts
26	Port screw
27	Toggle lever base
28	Toggle lever
29	Toggle lever links
30	Pivot washer
31	Plain yoke
32	Lock yoke
33	Lock-yoke wrench
34	Springs (two to a set)
35	Spring bolts
36	Spring bolt nuts
37	Diaphragm chamber plug

Table 2. Pressure Reducing Valves, Outside Spring Extra Heavy, 125 to 250 Lbs. (See Fig. in Table 1.)

A	B	C	D	E	F	G	H	Approximate net weights, pounds	
Dist. bet. faces flanged	Dist. bet. faces screwed	Diam. of flanges	Thickness of flanges	Center to bottom	Center to top	Length spring bolts	Diam. of hood	Screwed	Flanged
6	3	3½	9/16	1⅞	4⅛	4⅝	4⅜	4½	11
5	4¼	3¾	9/16	2¾	5½	12	6 9/16	12½	17
5¼	4⅝	4½	⅝	2⅞	5½	12	6 9/16	14	21
5½	....	5	¾	3⅜	6¾	12	7	....	30
6	....	6	13/16	3½	7	12	8	....	40
7½	....	6½	⅞	4¼	9	15¼	10	....	68
9	....	7½	1	4⅝	10⅛	15¼	10	....	92
10	....	8¼	1⅛	5	10½	15¼	10	....	138
11¼	....	9	13/16	5½	11	15¼	10	....	152
12½	....	10	1¼	6½	12	21½	12¼	....	201
14	....	10½	1 5/16	6½	12	21½	12¼	....	213
15	....	11	1⅜	6¾	12½	21½	12¾	....	235
17	....	12½	1 7/16	7½	13½	29½	14	....	321
18¼	....	14	1½	9¼	14	30	14⅛	....	450
20¾	....	15	1⅝	9¼	14½	30	14⅛	....	528
23½	....	16	1¾	10½	15	30	14⅛	....	640
23½	....	17½	1⅞	10½	15	30	14⅛	....	685
27¼	....	20	2	12¼	16½	30	16¾	....	1116
32	....	22½	2⅛	14½	22	30	19	....	1503
35	....	25	2¼	16	23	34	19½	....	1732
37½	....	27	2⅜	16¾	23¾	34	21	....	2124
40½	....	29½	2½	21	26	36	22	....	3380

Table 3. Pressure Reducing Valve, Inside Spring
(Foster Engineering Co.)

Sizes	Diam. of flanges	Dist. between faces	Dist. between screwed ends	Thickness of flanges	Center to bottom	Center to top	Net weights	
							Screwed	Flanged
All composition								
½	3½	5½	4	9/16	2¼	9½	11	15
¾	3¾	5½	4⅛	9/16	2¼	9½	13½	18
1	4	6	4½	⅝	2½	9⅞	14	19
1¼	4½	6½	4⅞	⅝	2½	10	17	23
1½	5	7	5	⅝	2¾	10¼	19	29
2	6	7½	6¼	⅝	3½	10¾	25	37
Iron body—composition trimmed								
2½	7½	10	8½	1	5¼	11½	65	73
3	8¼	10½	9¾	1⅛	6	11⅞	83	98
3½	9	11½	10¾	1 3/16	6½	12½	97	120
4	10	12½	11½	1¼	6⅝	13½	128	160
4½	10½	14	12½	1 5/16	7⅛	15¼	158	195
5	11	15	13¾	1⅜	7½	15¼	185	220
6	12½	17	14½	1 7/16	8⅛	16⅜	272	320
7	14	18¼	....	1½	10¾	19¼	....	490
8	15	20¾	....	1⅝	11¼	20¼	....	600
9	16	23	....	1¾	12⅛	21	....	900
10	17½	23½	....	1⅞	12⅝	21⅜	....	1020
12	20	27¼	....	2	15	28½	....	1240

The Inside Spring Type reducing valve, Fig. 2, operates as follows: Steam entering at *A* passes through main valve port *H* to outlet *B*. The

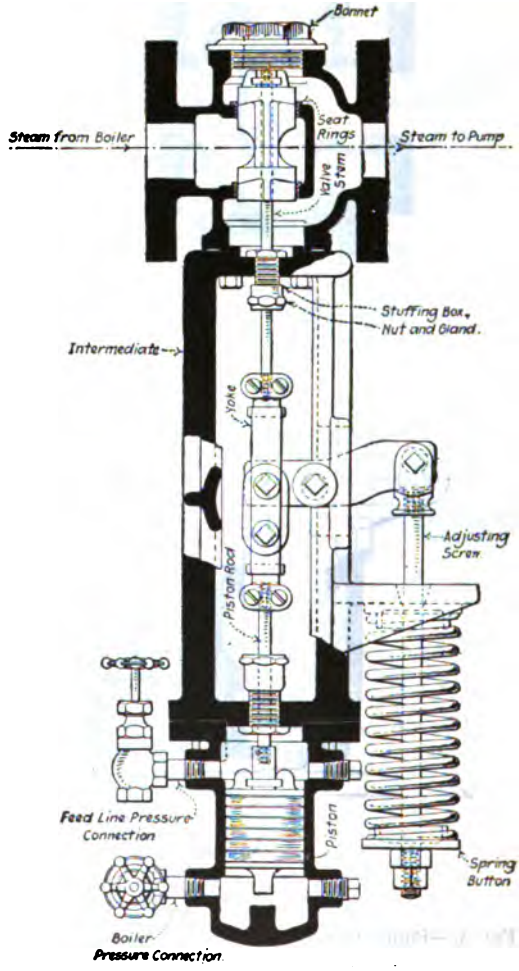


FIG. 4.—Pump Governor, Excess Pressure Type.

Auxiliary Operated Reducing Valve. The valve shown in Fig. 1 is designated a **direct operated reducing valve;** steam pressure acts on the diaphragm which is direct connected to the main valve and main springs. Fig. 2 illustrates an **auxiliary operated reducing valve;** in this case steam acts on the diaphragm but this is connected to the auxiliary valve 14, which auxiliary in turn controls steam to the piston 2 which operates the main valve. The **advantage of the auxiliary operated** valve is that very close regulation can be effected with very small movement of the auxiliary valve.

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STEAM TRAPS*

BY

F. G. HECHLER

REFERENCE: Gebhart, "Steam Power Plant Engineering," Wiley.

The purpose of steam traps is to remove all condensation from steam lines, separators, heating systems, etc., without permitting any steam to escape. They must function for long periods with practically no attention for repair or overhaul.

Classification

Steam traps are classified as return traps and non-return traps. **Return traps** are designed to return the condensate directly to the boiler without the aid of a pump. They are installed at a higher level than the boiler into which they discharge. When the trap is ready to discharge the float opens an equalizing valve and permits steam from the boiler to enter the trap, thus equalizing the pressures and permitting the condensate to drain by gravity from the trap back to the boiler. **Non-return** traps usually discharge to the atmosphere or into a low-pressure drain line. No equalizing valve is necessary, and, if desired, the condensate may be discharged into a tank placed above the trap. Non-return traps may be subdivided into bucket float, ball float, expansion and differential types.

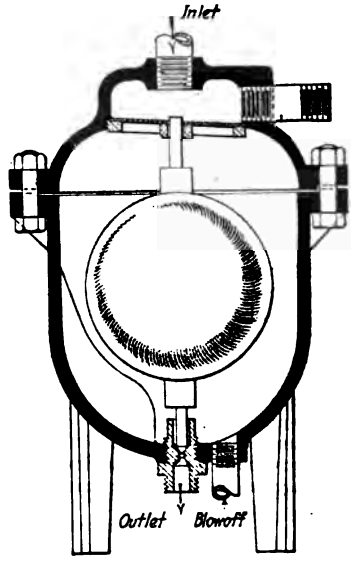

FIG. 1.—Simple ball float trap.

The bucket and ball float types are widely used for general marine service. Their outstanding characteristics are their prompt removal of all condensation as fast as formed and the ability to function satisfactorily over wide pressure ranges without requiring any readjustment. **The expansion or thermostatic type** finds its widest use in connection with heating systems or similar low-pressure work. This type requires some cooling of the condensate before it will function, which is undesirable for power lines because it requires additional piping for the purpose. Expansion traps will usually not operate satisfactorily over a wide pressure range without readjustment. Differential traps have only a limited use.

* Courtesy of The American Society of Naval Engineers.

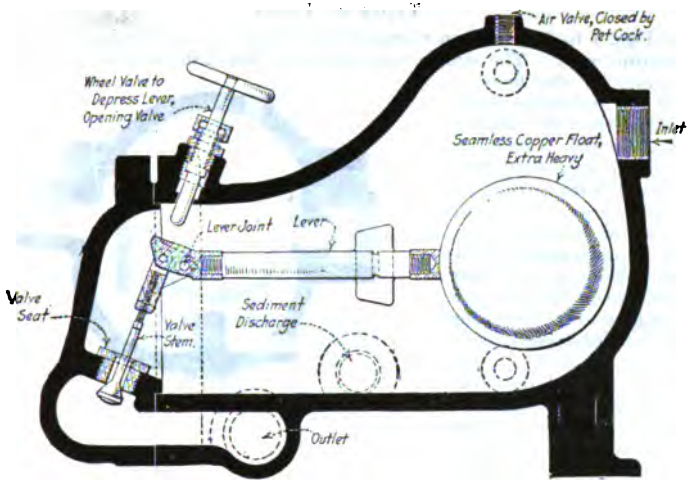

FIG. 2.—Ball float trap, lever type.

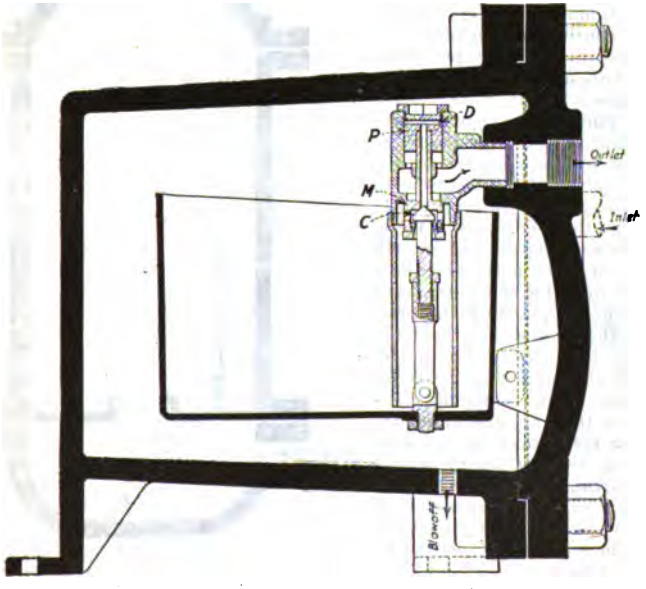

FIG. 3.—Bucket float trap, compound valve.

Types of Traps

In Figs. 1 to 7 are shown representative traps of the principal types used in marine work. The simple type shown in Fig. 1 depends directly on the buoyancy of the float to open the discharge valve against the unbalanced pressure which is equal to the valve area multiplied by the difference between the pressures in the trap and in the discharge line. This requires either a very large float or a very small valve, except for very low steam pressures. By using a system of levers, as in Figs. 2, 3, and 5, the effective pull of the float may be greatly magnified, permitting the use of larger valves and giving larger capacities with relatively small floats. The arrangement in Fig. 2 is such that the valve opens with the pressure instead of against it as is customary; this type of trap gives a continuous discharge.

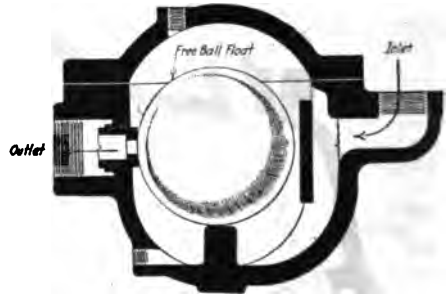

FIG. 4.—Ball float trap.

Bucket traps of the type shown in Fig. 3 are used more extensively in marine work than any other type. The design shown has a compound valve, consisting of a pilot valve *C* and the main discharge valve *M*. This type of trap discharges intermittently each time the bucket fills. When the pilot valve is opened by the descent of the bucket it admits pressure into the space *D* above the piston *P*, equalizing the pressure on the two sides of the main valve so that the further movement of the bucket opens it for discharge. The compound valve gives the trap a large capacity; a similar result may be obtained by using two or more simple valves arranged to open successively. Usually, however, only one valve is used as it is possible to obtain sufficient capacity by careful design.

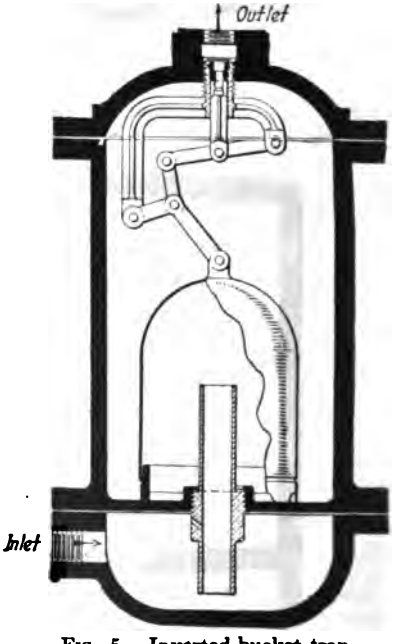

FIG. 5.—Inverted bucket trap.

An exceedingly simple trap is shown in Fig. 4. It is of the **ball float** type in which the unattached float itself forms the valve which opens and closes the

outlet depending on the water level. Air being heavier than steam and lighter than water will form a layer between the two and will be discharged as soon as the trap opens, thus avoiding all possibility of the trap becoming air bound.

The design shown in Fig. 5 is unusual, but appears to be satisfactory. When the condensate has all been discharged, steam enters the inverted **bucket** and renders it buoyant by displacing the water, and thus closes the discharge

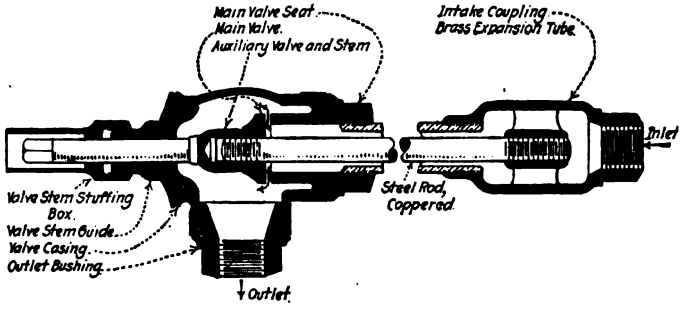

FIG. 6.—Expansion trap.

valve. When water again enters, the steam condenses and the bucket descends opening the valve. Any air that enters the trap escapes through the small hole in the top of the bucket, and is discharged with the water, hence it is claimed the trap can never become air bound.

Two types of **expansion** traps are shown in Figs. 6 and 7. In the design shown in Fig. 6 the relative expansion of the inner steel rod and the outer brass tube is depended upon to open and close the valve, while in Fig. 7 a volatile fluid, enclosed in a corrugated disk or chamber, furnishes the motive power. The latter type is widely used on radiators and heating coils; the first is said to be satisfactory for general use. In both types of expansion valves the adjustment may be such that the discharge will be either constant or intermittent.

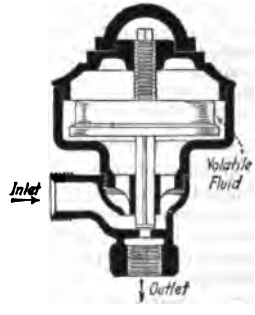

FIG. 7.—Expansion trap.

Design

The requirements for marine service are unusually severe; the trap must function satisfactorily at any pressure from atmospheric to the highest boiler pressure carried; its operation must not be adversely affected by the motion of the ship in a seaway; the valve mechanism must be simple and readily accessible for inspection and repair without having to disconnect any pipe line; the discharge valve must not be at the bottom of the trap where scale and sediment collect.

The **body of the trap** should be heavy and rigid, and flat surfaces should be avoided whenever possible. If the trap is intended for use with high steam pressures, the requirements of the American Standard for extra heavy

flanged fittings, p. 1286, should be complied with in determining the **thickness of body and of cover flanges**, as well as the size and the spacing of the **cover bolts**. Comparatively wide **gasket spaces** should be used to prevent the gasket from blowing out. Counter-bored recess gasket flanges are usually most satisfactory. The **inlet and outlet openings** should be on the body of the trap so the cover may be removed without breaking any pipe lines, and there should be a **bottom blow-off** for cleaning the trap of scale and sediment.

Buoyancy of the float in ball float traps must be sufficient to open the valve against the steam pressure. In this case the following relation must be satisfied:

$$W + (p_s - P_d)a < F$$

Where W = weight of float in lbs.; p_s = steam pressure in trap in lbs. per sq. in. gage; P_d = pressure against which trap discharges (usually atmospheric), lbs. per sq. in. gage; a = contact area of discharge valve, sq. in., F = effective buoyancy of float = displacement in cu. ft. times density of water in trap at temperature corresponding to the pressure in lbs. per cu. ft. (For density see p. 180.)

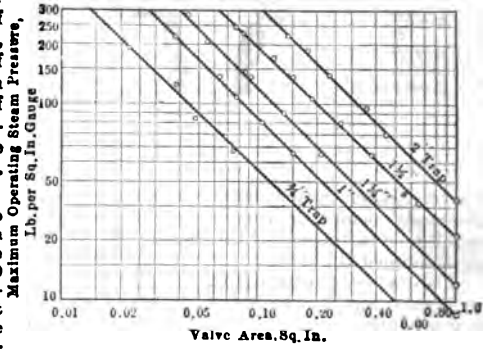

FIG. 8.

Example: A 1¼-inch trap of the simple float type shown in Fig. 1, and intended for use with pressures up to 300 pounds, had a float displacement of 0.1278 cu. ft.; the float weighed 4.17 pounds. The maximum size of the discharge valve calculated from the relation just given is 4.17 + (300–0) a<0.1278 times 52.6 or a <0.0085 sq. in., and the corresponding valve diameter is 0.104 in.

Small discharge valves are unsatisfactory because they are liable to become clogged with scale and because their discharge capacity is too small. Large floats are not desirable because they increase the size of the trap unduly, and because it is difficult to make a large float to withstand the high pressures. The use of levers to increase the float pull is usually resorted to for these reasons.

The effective pull of a bucket float for opening the valve may be found by filling the trap and bucket with water, attaching a spring balance to the valve stem and measuring the pull necessary to lift the bucket. Tests have shown that for bucket floats the relation between maximum operating pressure, valve area and bucket pull may be written as $p_s \times a^n = F$ where p_s, a, and F have the meanings given above, and where the exponent n has a value that is usually slightly less than unity (results varying from 0.93 to 1.00 have been found). When graphed on logarithmic paper this curve is a straight line whose slope gives the value of the exponent n. The relations between p_s and a as determined experimentally for a line of bucket float traps varying in nominal size from ¾ inch to 2 inch are shown in Fig. 8. The p intercept for an area of 1 sq. in. is the pulling power of the bucket as determined with the spring balance. In practice the valve areas should be somewhat less to insure satisfactory operation for slight over-pressures that may occur, and also to permit grinding in the valve and truing up the seat when

necessary. The allowance in area should be from 10 to 15 per cent; valve areas determined in this manner will be consistent and will give the maximum safe capacity. The design of the bucket float requires care; the addition of a small weight at the outer end of the bucket may increase the maximum operating pressure of the trap with a given valve as much as 20 per cent.

Trap Ratings and Capacities

There is no logical reason why steam traps should be rated, and also usually purchased, on a basis of the size of the inlet and outlet connections. Traps of the same nominal size may have widely different capacities at the same pressures as shown in Table 1. When specifying capacity it is necessary to state the pressure at which it is required. Capacities should always be determined with hot water delivered to the trap by steam at the necessary pressure; capacities determined with cold water and a pump will be too high.

Steam traps (except expansion) will operate satisfactorily at any pressure lower than the designed pressure without change or adjustment, but with a smaller capacity. To secure maximum capacity the valve should always be adapted to the operating pressure. When a trap is likely to be used over a considerable range of pressures it should be provided with several interchangeable valves of different size. For pressures up to 300 pounds it has been found sufficient to use three valves and seats designed for pressures of 30, 150 and 300 pounds. Table 2 gives the capacities as determined by test of a line of bucket traps at pressures of 30, 100 and 200 pounds. These capacities are the maximum quantity the traps can pass when discharging continuously under full steam pressure.

Table 1. Discharge Capacity of Steam Traps

Make of trap	Type	Nominal size, in.	Weight, lbs.	Capacity at 250 lbs. press.	Remarks
Vance	Bucket	1¼	145	26,360	Compound valve.
Ideal	Ball float	1¼	280	7,040	
Kieley and Mueller	Bucket	1½	200	20,250	Compound valve.
Sarco	Expansion	¾	14	2,700	At 200 lbs. press.
Johns-Manville	Ball float	1¼	44	3,480	
Armstrong	Inverted bucket	1	67	3,200	
Anderson	Ball float	1¼	165	6,230	
Corliss Valve	Ball float	1¼	...	4,250	
Swartwout	Bucket	1¼	145	4,100	
McDaniel	Ball float	1¼	136	13,800	
Strong	Bucket	1¼	176	4,800	
Fisher	Bucket	1¼	151	4,800	
Wright	Ball float	1¼	109	7,350	Three valves opening consecutively.
Sterling	Ball float	1	84	3,600	

Table 2. Variation of Capacity With Size of Trap

Nominal size, diameter inlet and outlet, in.	Capacity, lbs. per hr. under steam pressures of; lbs. per sq. in. gage		
	30	100	200
¾	6,180	4,400	2,400
1	8,600	8,200	6,400
1¼	14,600	9,900	9,100
1½	12,600	13,700	15,500
2	17,600	25,400	25,400

The above capacities were obtained by using an appropriate valve for each steam pressure.

Installation, Care and Operation

When installed on shipboard, traps are seldom horizontal and the constant change in inclination, due to the pitching and rolling of the vessel in a seaway, may interfere with their action. Many of the **bucket and ball float traps** lose their water seal and open the discharge valve, permitting steam to blow through when inclined in a longitudinal direction through an angle of about 15 degrees with the horizontal. This difficulty is to some extent obviated by making the operating mechanism as compact as possible; it has also been found with bucket traps that it is possible to increase the safe angle of inclination up to about 25 degrees by making the bucket with an inclined upper edge similar to that shown in Fig. 3. Whenever possible, traps should be installed on shipboard with their longitudinal axis fore and aft, thus decreasing the liability of steam blowing through. The operation of the traps shown in Figs. 4 and 5 is not affected by any inclination likely to occur. Expansion traps are also free from this defect as they operate equally well when installed in any position.

When the pressures are subject to considerable variation, **expansion traps** are not recommended for use unless provided with an automatic pressure compensating device. With some traps of this type it is difficult to maintain a given setting even under constant pressures, and in some cases the temperature of the surrounding air has an appreciable effect on the operation. Expansion traps may be adjusted to give either a constant or an intermittent discharge; they do not become air-bound, but sediment and grease prove troublesome. Since they have no storage capacity, 8 or 10 feet of pipe should be provided for cooling the condensate somewhat before it reaches the trap.

When **continuous discharge traps** are operating at low capacities the **wire drawing** at the valve is likely to prove destructive to the valves and seats; these should be arranged so they may be readily examined and renewed when necessary. The continuous discharge trap is well suited for use with long discharge lines, preventing water hammer and sudden shock likely to occur with the intermittent type under this condition. Whenever possible traps should be located so that the condensate will flow to them by gravity. If such a location would make the trap inaccessible, it may be placed at a higher level if the line to be drained carries a pressure of 5 lbs. or more. The drain line should always be connected to the lowest part of the line or apparatus to be dripped. **A trap should be connected to drain only a single line unless the pressures are known to be identical**; this is a rare condition and is likely to be disturbed from time to time. Traps may discharge to a higher level than the trap if necessary; it is usual to assume that the **condensate may be elevated** two feet for each pound of excess pressure in the trap over that in the discharge line. Traps should always be connected up with unions and fitted with a by-pass so that the trap may be overhauled without interfering with the proper drainage of the line. A test valve or cock should be placed in the discharge line close to the trap to determine how the trap is functioning. A blow-off line not smaller than ¾ inch should be provided for cleaning the trap.

Regular inspection of all traps should be made, and they should be promptly overhauled when improper functioning is detected. Steam traps are prone to leak due to cutting of the seat or scale lodging under the valve. Because under these circumstances there is no difficulty due to an accumulation of water, repairs are often neglected, permitting a serious steam leakage to occur.

Fittings

Gage glasses are desirable aids for determining the condition of a steam trap. Gage glass fittings should be of the ball check type so they will be self-closing in case of accident. Under high steam pressures tubular gage glasses frequently break. An **air cock** is necessary with most float traps to relieve the air when starting, and at such other times as may be necessary. A **removable strainer** placed at the trap inlet, or in the pipe line just before the trap, is often of value in preventing scale and sediment reaching the working parts of the trap.

Testing

No uniform code for testing steam traps has been adopted. New types of steam traps submitted for use in the Naval Service are subjected to the following tests: (See Jour., A.S.N.E., Vol. 28, No. 1, February, 1916.) (*a*) **Hydrostatic pressure** test of 150 per cent of the maximum working pressure (ball floats are separately tested to 500 pounds per sq. in.). (*b*) **Functioning** test with steam pressures from 25 to 300 pounds gage. (*c*) **Capacity** test to determine the maximum quantity of water the trap can discharge under various steam pressures. (*d*) **Rolling** test with the trap mounted on a rocking platform to simulate the motion of a vessel in a heavy seaway. (*e*) **Endurance** test with the trap installed for at least three months in the power plant to determine whether it will function properly a reasonable length of time.

Specifications

The specifications of the Bureau of Steam Engineering, Navy Department, for steam traps are as follows: (1) All traps will be of either open bucket or ball float, positive action type, unless otherwise specially approved by the Bureau, so designed that they can be overhauled without disconnecting the pipes, and the bonnets removed without disturbing the lagging. They will be located where they are easily accessible for examination. (2) The trap must be of a type that has given satisfaction in actual service on board a Naval vessel, or passed a satisfactory test at the U. S. Naval Engineering Experiment Station, Annapolis, Md. If a ball float is used, the ball must be able to stand an external pressure of not less than 500 lbs. per sq. in. (3) The blow-off will be arranged for hand operation, and must have a free opening for discharge, not less than ½ in. in diam. The drain (discharge valve) must be high enough above the bottom of the trap to prevent being clogged with sediment. All pipe connections to traps will be made with union couplings or by flanges. All traps will have by-pass pipes and valves for convenience of overhauling. All traps will be fitted with valves for testing.

PIPE COVERING AND LAGGING

BY

P. NICHOLLS

HEAT LOSSES FROM BARE METAL SURFACES

The laws of heat loss from bare surfaces to air are complicated due to the many variables that may enter into the action. They involve direct radiation to surrounding objects, convection losses caused by the air layers immediately in contact with the surface being heated and, by the natural circulation thus produced, carrying it away and being replaced by cool layers, and forced convection losses due to wind or air currents.

The law of direct radiation as stated by Steffan and Boltsman is $E = K(T^4 - T_0^4)$, where T and T_0 are the absolute temperatures of the hot body and surrounding bodies respectively and K is a factor depending on the nature of the surface. This law is not true for other than black bodies, and lacks experimental confirmation for iron surfaces.

The convection loss factor depends on the nature, shape and position of the surface. The freer the possibility of circulation, the greater will be the loss. The loss per square foot decreases with the height of a vertical surface, and is greater in a flat surface facing up than in one facing down. It is greater, per square foot, for a small horizontal pipe than for a large one, and for a horizontal than for a vertical pipe.

Windage increases the losses as much as 200 and 300 per cent, but as the actual value depends greatly on conditions comparatively little experimental work has been done. In a bare surface with a high conductivity material between the air and the source of heat, the conditions approach those of a boiler or condenser and the losses are only limited by interior and exterior surface actions.

The most exhaustive experiments on fundamental laws were made by Peclet in the middle of the last century, but his work was conducted on a laboratory scale, with small apparatus. The form of the empirical laws he deduced have been confirmed by tests with full size apparatus, but considerable variation from his constants have been found by later experimenters.

Table 1 gives bare pipe losses based on constant determined by the *Mellon Institute* applied to Peclet laws. These losses are given in B.t.u. per linear foot per hour, with air at 70 deg. F. These values are also translated into pounds of coal lost per hour on the assumption of 13,000 B.t.u. per lb. of coal and 70 per cent efficiency for the boiler

Table 1. Losses from Horizontal Bare Iron Steam Pipes per Linear Foot per Hour—Air Temp. 70 deg. F.
(Coal taken as 13,000 B.t.u. per lb.—boiler efficiency 70 per cent)

Gage press.	Hot water		10 lbs.		80 lbs.		120 lbs.		160 lbs.		200 lbs.		200 lbs. & 100 deg. F. Sup.	
Temp.	180 deg. F.		239.4 deg. F.		324 deg. F.		350 deg. F.		370.7 deg. F.		387.9 deg. F.		487.9 deg. F.	
Pipe size, in.	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal	B.t.u.	Lbs. coal
½	70	0.0076	121	0.0133	212	0.0233	245	0.0269	272	0.0299	296	0.0326	461	0.0506
¾	82	0.0090	143	0.0157	52	0.0277	291	0.0320	325	0.0357	354	0.0389	552	0.0607
1	98	0.0107	170	0.0187	301	0.0331	348	0.0382	388	0.0427	423	0.0465	664	0.0730
1¼	117	0.0129	205	0.0225	363	0.0399	420	0.0462	469	0.0516	512	0.0562	809	0.0889
1½	130	0.0143	227	0.0249	403	0.0443	468	0.0514	522	0.0574	570	0.0626	902	0.0991
2	156	0.0172	273	0.0300	487	0.0535	564	0.0618	63	0.0694	690	0.0758	1096	0.1205
2½	183	0.0202	321	0.0353	573	0.0630	665	0.0731	745	0.0818	814	0.0894	1297	0.1426
3	218	0.0239	382	0.0420	683	0.0751	793	0.0872	888	0.0976	971	0.1066	1551	0.1705
3½	246	0.0270	430	0.0472	770	0.0847	894	0.0983	1002	0.1101	1094	0.1201	1755	0.1930
4	273	0.0300	479	0.0526	857	0.0942	996	0.095	1116	0.1226	1220	0.1341	1956	0.2150
4½	300	0.0330	526	0.0579	944	0.1038	1097	0.1205	1230	0.1351	1345	0.1478	2159	0.2370
5	334	0.0367	582	0.0640	1043	0.1146	1212	0.1331	1359	0.1493	1486	0.1634	2390	0.2627
6	390	0.0429	684	0.0752	1227	0.1349	1428	0.1570	1604	0.1761	1751	0.1925	2820	0.3100
7	446	0.0490	782	0.0859	1404	0.1544	1634	0.1795	1831	0.2012	2002	0 2200	3227	0.3546
8	500	0.0550	878	0.0965	1588	0.1745	1835	0.2016	2059	0.2261	2251	0.2474	3642	0.4000
9	555	0.0610	975	0.1071	1751	0.1925	2040	0.2240	2288	0.2512	2502	0.2751	4040	0.4440
10	618	0.0679	1085	0.1192	1950	0.2143	2270	0.2495	2542	0.2794	2786	0.3060	4492	0.4940
12	729	0.0801	1280	0.1406	2301	0.2530	2680	0.2945	3011	0.3310	3290	0.3617	5310	0.5840
14	795	0.0875	1397	0.1535	2515	0.2764	2930	0.3220	3290	0.3620	3600	0.3958	5810	0.6385
16	908	0.0998	1595	0.1753	2870	0.3155	3340	0.3670	3752	0.4125	4100	0.4508	6630	0.7280

Table 2. Losses From Flat Bare Iron Surfaces, Steam and Direct Heated Per Sq. Ft. per Hour—Air 70 deg. F.
(Coal taken as 13,000 B.t.u. per lb.)

Temp. diff. deg. F	Steam gage pressure	Actual surface temp. air = 70 deg. F.	B.t.u. lost per sq. ft. per hour	Lbs. of coal wasted per hour: From steam heated surface boiler eff. = 70 per cent	Lbs. of coal wasted per hour: From fire heated surface
50	Hot water	120	87	0.0116	0.0067
100	Hot water	170	190	0.0151	0.0146
110	Hot water	180	215	0.0286	0.0165
169	10 lbs.	239	375	0.0500	0.0288
200	27.3 lbs.	270	479	0.0637	0.0369
254	80 lbs.	324	685	0.0913	0.0527
280	120 lbs.	350	798	0.1062	0.0615
300	160 lbs.	370	897	0.1198	0.0692
317	200 lbs.	387	982	0.1310	0.0755
350	160 lbs. + 50 deg. F. Sup.	420	1162	0.1548	0.0895
400	160 lbs. + 100 deg. F. Sup.	470	1493	0.1990	0.1150
418	200 lbs. + 100 deg. F. Sup.	488	1625	0.2160	0.1250
450		520	1890	0.2520	0.1455
500		570	2383	0.3170	0.1830
550		620	3050	0.4060	0.2350
600		670	3930	0.5230	0.3020

The losses from a flat bare metal surface depend also on its position and area. In a vertical surface the average loss per square foot decreases with the height. Peclet's empirical law shows that the change in value is not great above six feet.

Values that can be used for average conditions are given in Table 2.

Losses Through and Savings from Heat Insulation

Calculation of heat insulation values are based on the fact that the heat passing through the insulation must equal the dissipation of heat from the outer surface, commonly called the "total emission." The **power of emission** of a surface will depend on the surface conditions, its temperature, and the temperature of the surrounding air and objects, and the surface will reach such a temperature that the **rate of emission** will equal the rate of transfer through the insulation due to the temperature difference between the inner and outer surfaces. This is expressed by the following formulæ:

For flat surface

$$H = \frac{EC}{E - CL}(T_1 - T_0)$$

For pipes

$$H = \frac{R_2\, EC}{R_1\, C - E.\, R_2 \log_e \frac{R_2}{R_1}}\,(T_1 - T_0)$$

where

H = total heat lost per hour per sq. ft. of hot surface.
L = thickness of insulation in inches.
R_1 = inner radius of insulation in inches.
R_2 = outer radius of insulation in inches.
T_1 = temperature of hot surface.
T_0 = temperature of air.
C = coefficient of conductivity, B.t.u.'s per sq. ft. per inch thickness per hour per degree F. difference.
E = coefficient of emissivity per sq. ft. of outer surface per hour per degree F. difference.
C is a variable depending on the temperature, and for all practical materials increasing with it.
E is a variable and will depend on the nature of the surface. The form of the value of E as determined by *Peclet*, *McMillan* and the *Mellon Institute* is closely, $E = a - b\,(T_2 - T_0)$, where T_2 is the temperature of the surface, T_0 of the air and a and b are constants. Values found for the constants differ and depend on the location.

Due to the difficulty in determining the true laws for E and C, results have often been expressed in terms of a total conductance, C_1, between the inner temperature and the air, or

$$H = C_1\,(T_1 - T_0)$$

This is, however, entirely arbitrary, and can only be used to compare results of exactly similar and equal geometrical conditions.

The application of fundamental coefficients to the above equations is long and tedious. Tables 3 and 4 give the heat transmitted through 85 per cent Magnesia for a limited number of practical steam pressures and pipe sizes. These values are calculated from the fundamental laws, and values of E and C for each temperature found by the *Mellon Institute*.

The tables give the B.t.u. that will be transmitted through the insulation and the B.t.u. saved by its use, for thicknesses corresponding to the Commercial List thicknesses.

Table 3.—Heat Losses Through 85 Per Cent Magnesia On Vertical Flat Surfaces And Heat Saving Over Bare Surface Losses (Air Temp. 700 Deg. F.)

Temp. dif.	Steam gage pressure	Actual hot surface temp. (Air at 70 deg.)	B.t.u. per sq. ft. per hr.													
			1 in. thick		1½ in. thick		2 in. thick		2½ in. thick		3 in. thick		4 in. thick		5 in. thick	
			Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.	Lost B.t.u.	Save B.t.u.
50	Hot water.....	120	18	69	13	74	10	77	8	79	7	80	5	82	4	83
100	Hot water.....	170	40	150	28	162	22	168	18	172	15	175	12	178	9	181
110	Hot water.....	180	44	171	32	183	25	190	20	195	17	198	13	202	10	205
169	10 lbs.....	239	72	303	51	324	39	336	33	342	28	347	21	354	17	358
200	27.3 lbs.....	290	88	391	61	418	47	432	38	441	33	446	26	453	20	459
254	80 lbs.....	324	115	570	81	604	64	621	53	632	45	640	35	650	28	657
280	120 lbs.....	350	128	670	90	708	71	727	59	739	49	749	38	760	30	768
300	160 lbs.....	370	138	759	98	799	77	820	64	833	53	844	41	856	33	864
317	200 lbs.....	387	148	834	105	877	82	900	68	914	57	925	44	938	35	947
350	160 lbs. & 50 deg. F. Sup	420	167	995	118	1044	92	1070	78	1084	64	1098	49	1113	40	1122
400	160 lbs. & 100 deg. F. Sup	470	197	1296	139	1354	108	1585	89	1404	75	1418	57	1436	47	1446
418	200 lbs. & 100 deg. F. Sup	488	209	1416	148	1477	115	1510	94	1531	79	5546	61	1564	50	1575
450	.....	520	228	1662	161	1729	125	1765	103	1787	86	1804	66	1824	54	1836
500	.....	570	261	2122	185	2198	143	2240	117	2266	98	2285	75	2308	61	2322
550	.....	620	295	2755	207	2843	161	2889	131	2919	111	2939	85	2965	69	2981
600	.....	670	329	3601	230	3700	178	3752	145	3785	123	3807	95	3835	77	3853

Table 4. Heat Losses Through 85 Per Cent Magnesia Pipe Covering and Heat Saving over Bare Pipe Losses both in B.t.u. per Hour per Linear Foot of Pipe

(Temperature of air taken as 70 deg. F.).

Gage pressure		Hot water		10 lbs.		80 lbs.		120 lbs.		160 lbs.		200 lbs.		200 lbs. and 100 deg. Sup.	
Temp. deg. F.		180		239.4		324.0		350.0		370.7		387.9		487.9	
Pipe size, in.	Thickness*	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.
	S7/8	25	57	41	102	67	185	75	216	81	244	88	266	123	429
	1½	20	62	32	111	52	200	58	233	63	262	69	285	95	457
¾	D1¾	18	64	30	113	49	203	55	236	59	266	64	290	88	664
	2	16	66	28	115	46	206	51	240	56	269	60	294	84	468
	3	14	68	24	119	39	213	43	248	97	278	51	303	71	481
	S7/8	30	68	47	123	75	226	84	264	91	297	99	324	138	526
	1½	22	76	37	133	59	242	66	282	71	317	77	346	107	557
1	D1¾	21	77	34	136	54	247	60	288	65	323	70	353	98	566
	2	19	79	32	138	51	250	57	291	61	327	66	357	92	572
	3	16	82	27	143	42	259	47	301	50	338	55	368	77	587
	S7/8	35	82	55	150	91	272	102	318	111	358	119	393	165	644
	1½	26	91	42	163	68	295	76	344	82	387	89	423	123	686
1¼	D1¾	24	93	39	166	63	300	70	350	76	393	82	430	113	696
	2	22	95	37	168	59	304	66	354	71	398	76	436	106	703
	3	18	99	29	176	48	315	54	366	58	411	63	449	87	722
	S7/8	37	93	59	168	97	306	109	359	118	404	128	442	181	721
	1½	27	103	45	182	74	329	82	386	89	433	96	474	133	769
1½	D1¾	25	105	42	185	67	336	75	393	81	441	88	482	122	780
	2	23	107	40	187	64	339	71	397	77	445	82	488	114	788
	3	19	111	31	196	51	352	58	410	63	459	67	503	93	809
	S1 1/16	39	117	64	209	103	384	116	448	126	505	136	554	191	905
	1½	32	124	52	221	84	403	94	470	101	530	109	581	152	944
2	2	28	128	46	227	72	415	81	483	87	544	94	596	129	967
	D2 1/16	27	129	44	229	70	417	79	485	86	545	92	598	127	969
	3	22	134	35	238	57	430	65	499	70	561	76	614	103	993

2½	S1½	46	137	73	248	118	455	133	532	145	600	156	658	217	1080
	1½	37	146	59	262	95	478	105	560	114	631	123	691	171	1126
	2	31	152	50	271	80	493	90	575	97	648	104	710	145	1152
	D2	30	153	48	273	78	495	88	577	95	650	102	712	143	1154
	3	24	159	39	282	63	510	71	594	77	668	83	731	114	1183
3	S1½	52	166	84	298	136	547	153	640	165	723	180	791	255	1296
	1½	42	176	68	314	108	575	120	673	130	758	140	831	196	1355
	2	36	182	59	323	94	589	105	688	114	774	122	849	170	1381
	D2	34	184	56	326	92	591	103	690	110	778	120	851	166	1385
	3	28	190	44	338	72	611	80	713	87	801	94	877	130	1421
	4	22	196	38	344	61	622	68	725	74	814	80	891	111	1440
3½	S1½	58	188	93	337	152	618	170	724	184	818	199	895	277	1478
	1½	47	199	74	356	120	650	133	761	144	858	156	938	220	1535
	2	37	209	62	368	102	668	114	780	124	878	134	960	185	1570
	D2	35	211	60	370	100	670	112	782	120	882	130	964	180	1575
	3	30	216	48	382	77	693	86	808	93	909	101	993	141	1614
	4	25	221	41	389	68	702	75	819	82	920	88	1006	120	1635
4	S1¼	61	212	98	381	157	700	176	820	190	926	205	1015	290	1666
	1½	50	223	80	399	130	727	146	850	158	958	171	1049	238	1718
	2	42	231	68	411	110	747	124	872	135	981	145	1075	198	1758
	D2¼	38	235	62	417	102	755	114	882	124	992	135	1085	185	1771
	3	32	241	52	427	84	773	94	902	103	1013	110	1110	153	1803
	4	28	245	44	435	70	787	79	917	86	1030	93	1127	129	1827
4½	S1¼	65	235	105	421	173	771	196	901	210	1020	226	1119	316	1843
	1½	56	244	89	437	142	802	160	937	173	1057	187	1158	259	1900
	2	47	253	74	452	118	826	133	964	145	1085	156	1189	217	1942
	D2¼	41	259	68	458	110	834	123	974	136	1094	144	1201	197	1962
	3	35	265	56	470	90	854	101	996	110	1120	118	1227	163	1996
	4	30	270	46	480	76	868	85	1012	93	1137	100	1245	138	2021
5	S1¼	72	262	115	467	183	860	204	1008	220	1139	239	1247	342	2048
	1½	59	275	95	485	151	892	168	1044	182	1177	198	1288	281	2109
	2	49	285	80	502	127	916	141	1071	153	1206	166	1320	231	2159
	D2¼	44	290	73	509	119	924	132	1080	143	1216	155	1331	214	2176
	3	37	297	60	522	99	944	110	1102	120	1239	129	1357	177	2213
	4	31	303	50	532	81	962	92	1120	100	1259	107	1379	148	2242

* In column of thickness, "S" means "Standard" and "D" means "Double Standard."

Table 4. Heat Losses Through 85 Per Cent Magnesia Pipe Covering and Heat Saving over Bare Pipe Losses both in B.t.u. per Hour per Linear Foot of Pipe. (*Continued*)

(Temperature of air taken as 70 deg. F.)

Gage pressure		Hot water		10 Lbs.		80 Lb.		120 lb.		160 lb.		200 lb.		200 lbs. and 100 deg. Sup.	
Temp. deg. F.		180		239.4		324.0		350.0		370.7		387.9		487.9	
Pipe size, in.	Thickness*	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.	Loss B.t.u.	Save B.t.u.
	S1⅛	85	305	138	546	222	1005	248	1180	270	1334	290	1461	407	2413
	1½	68	322	110	574	176	1051	198	1230	214	1390	231	1520	320	2500
	2	56	334	91	593	145	1082	164	1264	176	1428	189	1562	261	2559
6	D2¼	52	338	85	599	135	1092	150	1278	162	1442	175	1576	242	2578
	3	42	348	68	616	111	1116	124	1304	135	1469	145	1606	201	2619
	4	35	355	57	627	92	1135	102	1326	110	1494	119	1632	166	2654
	S1¼	85	361	138	644	225	1179	252	1382	273	1558	296	1707	414	2813
	1½	75	371	124	658	198	1206	220	1414	239	1592	256	1744	362	2865
	2	62	384	102	680	164	1240	182	1452	198	1633	212	1790	292	2935
7	D2½	53	393	86	696	137	1267	154	1480	166	1665	179	1823	248	2979
	3	47	399	76	706	122	1282	136	1498	146	1685	159	1843	220	3007
	4	37	409	62	720	100	1304	114	1520	124	1707	134	1868	182	3045
	S1¼	95	405	156	722	250	1338	280	1555	302	1757	327	1924	460	3182
	1½	85	415	137	741	220	1368	245	1590	265	1794	285	1966	401	3241
	2	70	430	113	765	181	1407	201	1634	218	1841	235	2016	328	3314
8	D2½	60	440	94	784	153	1435	171	1664	187	1872	200	2051	275	3367
	3	52	448	85	793	135	1453	151	1684	163	1896	176	2075	245	3397
	4	42	458	67	811	110	1478	125	1710	135	1924	145	2106	200	3442
	5	35	465	58	820	94	1494	105	1730	114	1945	121	2130	168	3474
	S1½	108	447	170	805	278	1473	314	1726	340	1948	366	2136	506	3534
	1½	92	463	150	825	240	1511	268	1772	288	2000	311	2191	440	3600
	2	75	480	121	854	195	1556	220	1820	237	2051	255	2247	352	3688
9	D2½	65	490	104	871	168	1583	186	1854	202	2086	217	2285	301	3739
	3	55	500	90	885	146	1605	164	1876	177	2111	191	2311	265	3775
	4	45	510	73	902	119	1632	134	1906	144	2144	155	2347	215	3825
	5	39	516	63	912	100	1651	113	1927	121	2167	132	2370	185	3855

	S1¼	116	502	192	893	304	1646	336	1934	366	2176	395	2391	558	3934
	1½	103	515	168	917	268	1682	298	1972	325	2217	348	2438	481	4011
	2	82	536	135	950	218	1732	244	2026	265	2277	284	2502	388	4104
10	D2½	70	548	112	973	183	1767	205	2065	222	2320	239	2547	328	4164
	3	60	458	98	987	160	1790	180	2090	193	2349	209	2577	289	4203
	4	50	568	80	1005	130	1820	145	2125	160	2382	170	2616	234	4258
	5	40	578	68	1017	110	1840	121	2149	131	2411	141	2645	200	4292
	S1½	120	609	195	1085	311	1990	346	2234	375	2636	404	2886	560	4750
	2	95	634	155	1125	248	2053	276	2404	300	2711	323	2967	450	4860
12	D3	70	659	114	1166	182	2119	204	2476	220	2791	237	3053	330	4980
	4	58	671	92	1188	146	2155	165	2515	180	2831	194	3096	270	5040
	5	50	679	79	1201	126	2175	141	2539	153	2858	165	3125	230	5080
	S1½	130	665	210	1187	332	2183	372	2558	402	2888	437	3163	611	5199
	2	103	692	168	1229	270	2245	302	2628	330	2960	351	3249	482	5328
14	D3	72	723	120	1277	200	2315	225	2705	242	3048	260	3340	352	5458
	4	60	735	96	1301	160	2355	179	2751	192	3098	206	3394	282	5528
	5	52	743	85	1312	136	2379	150	2780	160	3130	177	3423	240	5570
	S1½	150	758	240	1355	380	2490	424	2916	460	3292	495	3605	685	5945
	2	120	788	190	1405	305	2565	342	2998	372	3380	397	3703	545	6085
16	D3	84	824	139	1456	221	2649	246	3094	269	3483	285	3815	392	6238
	4	69	839	111	1484	179	2691	200	3140	215	3537	231	3869	315	6315
	5	58	850	93	1502	151	2719	170	3170	185	3567	198	3902	269	6361
	S1½	180	955	288	1702	458	3192	512	3668	556	4134	602	4538	840	7510
	2	140	995	228	1762	370	3280	418	3762	455	4235	490	4650	670	7680
20	D3	101	1034	162	1828	268	3382	300	3880	328	4362	350	4790	480	7870
	4	82	1053	132	1858	215	3435	240	3940	260	4430	280	4860	381	7969
	5	70	1065	112	1878	179	3471	199	3981	212	4478	232	4908	325	8025
	S1½	210	1145	339	2041	548	3812	611	4389	662	4963	718	5442	1000	9000
	2	162	1193	269	2111	435	3925	489	4511	530	5095	569	5591	795	9205
24	D3	119	1236	195	2185	312	4048	350	4650	378	5247	405	5755	563	9437
	4	95	1260	156	2224	250	4110	280	4720	302	5323	329	5831	458	9542
	5	80	1275	130	2250	210	4150	235	4765	253	5372	273	5887	381	9619

These values are based on an air temperature of 70 deg. F. Corrections for other air temperatures can be applied by multiplying the figure given in the table by:

$$\frac{\text{(bare surface temp.)} - \text{(actual air temperature)}}{\text{(bare surface temp.)} - 70}$$

The difference between these values and the loss that would occur with bare surfaces gives the saving effected by the insulation, and the efficiency of the insulation is expressed as:

$$\text{Insulation Efficiency} = \frac{\text{(Loss from bare pipe)} - \text{Loss insulated}}{\text{(Loss from bare pipe)}}$$

The coal or money saved by a given insulation can be calculated from the tables, but a distinction must be made between surfaces which are heated directly by the fuel, such as furnace walls and those which are steam heated.

If each lb. of fuel contains H B.t.u., then lbs. of fuel saved per hour for direct heated surfaces =

(for flat surfaces) $$\frac{\text{(B.t.u. saved per sq. ft.)} \times \text{(area in sq. ft.)}}{H}$$

(for pipes) $$\frac{\text{(B.t.u. saved per lineal foot)} \times \text{(length in feet)}}{H}$$

For steam heated surfaces these values must be divided by the boiler efficiency, which may run from 50 to 70 per cent. A fair average is 60 per cent.

As a further refinement the results of losing heat from steam must be considered. If the steam is superheated its temperature is lowered, with the necessity of a larger boiler investment to make up for the drop, or the efficiency of the engine is decreased. If the steam is not superheated, the loss of heat will mean that some of the steam will be condensed and will be either dropped in the pipe or caught by the separator before reaching the engine. It will then be water at the same temperature as the steam, and if all the heat in it is returned to the boiler it will be saved. If it is not returned to the boiler it will all be lost and this amount must be added to that actually dissipated from the pipes.

To express the fuel saved in true money saving, to the money value of the fuel saved must be added the cost of handling and carrying this fuel and its percentage of overhead.

Economic Thickness of Insulation. There is a certain thickness of insulation which will give the maximum net saving. The money saving per year can be calculated for any thickness when the number of hours the surfaces are under heat is known. The yearly charge due to the cost of applying the insulation is also known and consists of the sum of the interest, depreciation and repair charges. The Economic Thickness is that which gives the greatest difference between these values, or the thickness at which the rates of change in the two values are equal.

The Economic Thickness thus differs for each steam cost. The value for 85 per cent Magnesia and for four steam costs are shown in Figs. No. 1 to 4. These curves are based on a continuous service of 8760 hours per year. Where the pipes are cold part of the time, multiply the steam cost by the hours of operation per year divided by 8760, and use this value for the actual steam cost. In computing these curves 20 per cent of the list cost of the covering has been allowed for the cost of application, and 13 per cent of the total cost for the annual charges, 6 per cent interest, 5 per cent depreciation, 2 per cent miscellaneous.

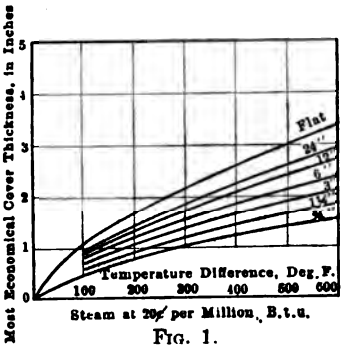

FIG. 1.

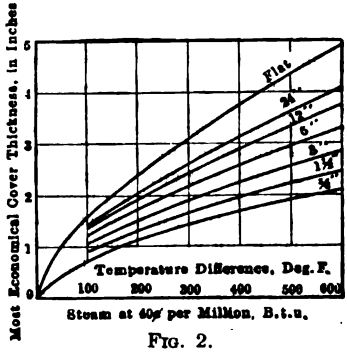

FIG. 2.

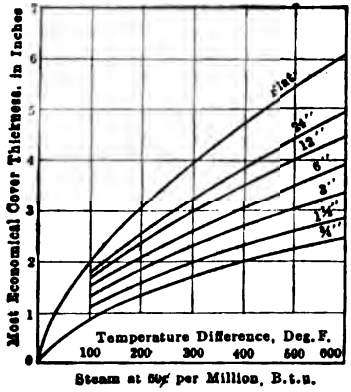

FIG. 3.

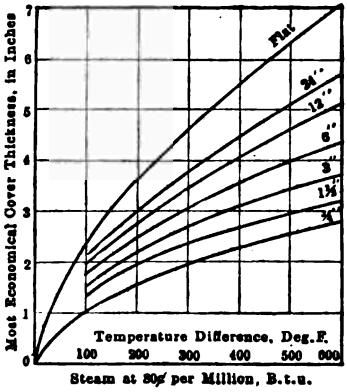

FIG. 4.

Table 5 connects the steam cost per million B.t.u. with the price of coal on the following basis:

B.t.u. per lb. of coal	= 13,000
Boiler efficiency	= 70 per cent
Labor of handling and overhead	= $1 per ton of coal

Table 5

Cost of coal in bunkers per 2240 lbs.	Cost of steam per million B.t.u.	Cost of coal in bunkers per 2240 lbs.	Cost of steam per million B.t.u.
$2.00	$0.15	$7.00	$0.40
3.00	0.20	8.00	0.45
4.00	0.25	10.00	0.55
5.00	0.30	12.00	0.65
6.00	0.35	14.00	0.75

Commercial Insulations

Commercial Insulations are classed as follows:

1. **Loose vegetable or animal fibres** felted or moulded, used only at low temperatures on refrigerating, cold water or hot water pipes or surfaces. Such insulations are **wool** and **hair felt,** and **cork,** by themselves or **combined with asbestos** or roofing papers or containers.

2. **Asbestos insulations** made from loose fibres, moulded to shape and the surface hardened with a binder, or formed into paper and built into layers **with intervening air spaces.** Available for all low and moderate temperature surfaces up to the limit of steam temperatures.

3. **Moulded powders** with or without binders. The efficiency of such insulations largely depends on the proportion of entrained air contained in the moulded product, and the larger the percentage of voids the greater will be its efficiency as an insulator. Those powders whose crystals have the smallest absolute conductivity and the best reflecting surface have increased efficiency.

Plaster of Paris, lime, gypsum and other materials have been used. Plaster of Paris being an acid salt corrodes metals, and lime and gypsum have naturally low efficiency due to their high density when moulded.

Moulded cork scrap is used for low temperatures and refrigerating conditions.

Infusorial earth moulds to a low apparent density but, unless used in block form as cut from its bed, has poor binding qualities and requires an artificial binder to give it cohesion. Being an oxide of silica it can in its natural form be used on the highest temperatures.

Basic Hydrated carbonate of magnesia moulds into shapes with 90 per cent voids, and due to the interlocking of the minute crystals possesses considerable inherent strength. Commercially it is known as **85 per cent magnesia** and is mixed with 10 to 15 per cent of asbestos fibre to give it added strength. It can be machined accurately to shape and is available for the highest superheated steam temperatures.

Standard Commercial Sizes

Block Insulation is made in various sizes according to the material, but moulded materials are standardized for 6 in. × 36 in., or 3 in. × 18 in., with any thickness from ½ in. to 4 in.

Pipe Covering is made in lengths of 36 in., in sectional and segmental forms. Sectional forms are used for small and medium pipes and are split longitudinally for ease of application. Segmental blocks are used for larger diameters and are wired in place.

Manufacturers of Pipe Covering have adopted a Standard List of sizes which is known as the "1907 List." This list is given in Table 6. The sizes are known by their names, such as "3 in. Double Standard."

Application of Insulation. Double layer Insulation has the advantage that it closes up the cracks better than single layer, and is therefore desirable for high temperatures. For particularly fine work it is common to put a layer of Plastic Cement, which is smoothed to a good finish and covered with a sheathing paper, followed by an 8 oz. canvas jacket, which is sewed on.

Flat surfaces can be covered with insulation blocks and smoothed off with a layer of Plastic Cement, and irregular surfaces are usually covered with Plastic Cement only.

Table 6. Standard Pipe Covering Sizes

Pipe size, inches	Standard	1½ in.	2 in.	Double standard	Double 1½ in.
	Thickness, inches	Thickness, inches	Thickness, inches	Total thickness, inches	Total thickness, inches
½	⅞	1½	2	1¾	3
¾	⅞	1½	2	1¾	3
1	⅞	1½	2	1¾	3
1¼	⅞	1½	2	1¾	3
1½	⅞	1½	2	1¾	3
2	1 1/32	1½	2	2 1/16	3
2½	1 1/32	1½	2	2 1/16	3
3	1 1/32	1½	2	2 1/16	3
3½	1 1/32	1½	2	2 1/16	3
4	1⅛	1½	2	2¼	3
4½	1⅛	1½	2	2¼	3
5	1⅛	1½	2	2¼	3
6	1⅛	1½	2	2¼	3
7	1¼	1½	2	2½	3
8	1¼	1½	2	2½	3
9	1¼	1½	2	2½	3
10	1¼	1½	2	2½	3
12	1½	1½	2	3	3
14	1½	1½	2	3	3
16	1½	1½	2	3	3
18	1½	1½	2	3	3
20	1½	1½	2	3	3
24	1½	1½	2	3	3
30	1½	1½	2	3	3

Where the insulation is liable to damage a thin coat of hard finish Cement should be used, and for stronger protection sheet iron is employed.

Fittings, valves, and flanges are covered with Plastic Cement or Blocks and Plastic Cement being finished off with canvas, or a smooth cemented surface.

High pressure steam line flanges should have removable insulation coverings, which are usually made up in two parts.

All feed-water heaters, hot-water tanks, receivers, separators, etc., should be covered to a proper thickness for their temperature, using Plastic Cement or Blocks as most convenient.

A good Cement will stick to surfaces, especially if put on with the surfaces hot, and where there is a large flat surface it is good practice to imbed chicken wire, which is fastened to the surface in a convenient manner.

Pipe Covering Specifications

The **Magnesia Association of America** has issued a standardized specification, covering the method of application for all classes of apparatus.

The thickness of covering to be used would be settled by the engineer and the Association recommends that such thicknesses should be the economic sizes obtained on the principle enumerated on p. 1356 under the heading "Economic Thicknesses." These might be adapted to marine use; copies may be had from the Association on application.

SECTION 12

MARINE ELECTRICAL INSTALLATION

BY

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CONTENTS

	Page
Methods of Installation:	
Classification Society Rules	1363
Drawings	1364
Wire Sizes	1364
Protection of Conductors	1365
Types of Conductors	1366
Generating Sets	1367
Marine Switchboards	1371
Lighting System:	
Circuits	1371
Fittings	1372
Running Lights	1373
Lamps	1374
Searchlights	1375
Power System	1376
Signalling Systems:	
Engine Telegraphs	1377
Telephones	1377
Fire Alarms	1377
Submarine Signals	1378
Radio Telegraph	1378

MARINE ELECTRICAL INSTALLATIONS

BY

H. A. HORNOR

LIST OF SYMBOLS

⊗	Steamtight Deck Fixtures.
◉	N. W. T. Key Receptacle.
◎	Pendant Fixture.
—o	Bracket Fixture.
✱	5 Light Electrolier.
—×—	Perkins 5 amp. Indicating Snap Switch.
▭	Distribution Panel (Number of circuits as marked).
▭	5 amp. Navy Standard Switch and Receptacle.
●	Feeder Junction Box.
o	Branch Junction Box.
⊙	Watertight Branch Box not fused.
~◺	Desk Light, Navy Standard.
⊠	Watertight Switch, Navy Standard.

REFERENCES: J. W. Kellogg, "Uses of Electricity on Board Ship," Marine Engineering, 1903, vol. 8, p. 342. H. A. Hornor, Trans. Amer. Inst. Elec. Eng., 1915, vol. 34, p. 1515. H. A. Hornor, "Electricity's Part in Building and Navigation of Ships," Electrical Engineering, 1918, vol. 51, p. 28.

METHODS OF INSTALLATION

Rules and Requirements. In order to obtain insurance a vessel must be classed by some organization whose business it is to properly survey and maintain a record of the vessel. In addition to this the vessel in all its parts must comply with the navigation laws of the governing country and to such specific rules as apply to international canals. In general, the constructional details must conform to these rules and for particular details the owner prepares his own specifications as an instrument of contract between himself and the builder. By far the largest number of vessels are classed by the **Lloyd's Register** of British and Foreign Shipping although in recent years in this country many vessels are classed by the **American Bureau of Shipping.** These Societies have provided guidance rules on electrical installations which cover protection against fire risk and exposure of the component parts of the installation to the injurious effects of salt air and sea water. They are liberal in their approval of designs which are submitted to them in line with the progress and development of electrical engineering and are willing at all times to submit any suggestions to practical service tests. **Copies of the rules are obtainable upon application to the Societies.** They must be familiar to those in responsible charge of the work of installation and to the engineer who lays out the drawings.

The following **specifications** may serve as a sample for the electrical equipment of a general freight vessel using only electric lighting: The electrical plant will consist of two vertical marine generating sets of 10-kw. capacity. The generating set will consist of one compound-wound, direct-current generator, rated at 110 volts, direct-connected to a single cylinder vertical engine arranged to give its full rated output at 110 lbs. steam pressure. A main switchboard will be provided with the proper instruments for observing and controlling voltage, and necessary switches for the proper distribution of current. Distribution panels will be provided for controlling lights in the quarters.

Approximately one hundred 110-volt, 25-watt masda lamps will be furnished, hand portables to be supplied where necessary; four 4-light cargo clusters to be furnished. The 2-wire system of distribution will be used. Lead-armored cable will be used throughout, and all work to be done in accordance with Lloyd's requirements. In the engine room, fire room, on open decks and any place liable to a mechanical injury, steamtight guarded fixtures will be used; the switches, junction boxes, etc., in these places must be watertight. In the quarters non-watertight fixtures of a standard design, as well as non-watertight fittings will be acceptable. Fixtures must be adapted to the location. One 18-inch searchlight complete must be provided. One set of galvanized iron steaming lights to be equipped for oil and electric lamps, except where noted, will be furnished as follows: one port light, one starboard light, two masthead lights (electric only), two riding lights, one overtaking light. In order to assure that these steaming lights are functioning at all times, there will be provided one 5-circuit tell-tale panel located in the pilot house. Spare brushes for the generators, spare assorted fuses, and 25 extra 25-watt masda incandescent lamps will be furnished.

The communicating system will consist of mechanical repeating telegraphs with large brass dials for the engine room, and white enamel with glass front in the pilot house. A speaking tube will be provided between the bridge and pilot house, between pilot house and captain's cabin and between pilot house and engine room. Mechanical whistle pulls will be provided in pilot house. Alarm bells will be provided in accordance with standard requirements.

Drawings. On the reverse side of the deck plans as provided by the hull division, which plans comprise a general outline of the hull, bulkheads and various pieces of permanent furniture, the electrical draftsman lays in with pencil the location of the main switchboard, distribution panels, outlets for lights, receptacles, switches, etc. The number of feeders are usually given and he selects the run of leads from the switchboard to the terminal apparatus.

Wire sizes are governed by the Underwriters Rules and the fundamental safety proposition that the **permissible watts per circuit** is based on the voltage of the system times the safe carrying capacity of the wire. For example, No. 14 B. & S. wire is allowed to carry 12 amperes which multiplied by 110 volts equals 1320 permissible watts. Wires for the **lighting system** are proportioned for a 3 per cent drop in voltage from the switchboard to the farthest outlet but except in very special installations this requirement is rarely checked up in the finished installation. A voltage drop of 5 to 8 volts is permissible for **motor circuits** but the distances on shipboard are usually so short that the safe conductivity required for the motor starting current covers any question of drop in voltage. In regular practice such calculations are not made.

Electrical circuits are all designated by single solid black lines; these lines are inked in after the calculations have been made and the layout tentatively approved. Each line is then designated by thin black arrow lines giving the size of wire, the type of protection, the total number of wires (usually two) comprising the circuit. Each lighting outlet and all the various appliances are designated by symbols as illustrated.

Electrical drawings are made to the same scale as **general arrangement drawings**, usually ⅛ in. = 1 ft. They are at the best a guide for the foreman on the ship. The nature of the installation permits a great deal of flexibility in the attachment of the

wires to the ship's hull structure and it is not the practice to make **detail drawings** of any part of the installation. In certain very large vessels having an extensive lighting system, especially in passenger ships, the drawings may be increased to a scale of ¼ in. = 1 ft., in order that the draftsman will not crowd out too much of the general arrangement plan where he has a number of lines grouped together, and also in order to aid the electrical foreman by permitting more outlines of other installations such as ventilation, sanitary piping, heating, etc.

The draftsman is an important link in the final accomplishment of good workmanship on the vessel, as he and the foreman must confer together during the process of construction so as to avoid interferences and yet maintain as straight runs as possible for the electrical leads. The draftsman must be cognizant with (1) the current required for the operation of the searchlights, motors and incandescent lamps; this information he can obtain through a familiarity with the manufacturers apparatus; (2) the kilowatt capacity of the generating sets in order that he may determine the wire sizes connecting the generator or generators to the main switchboard; (3) the other installations specified for the vessel and usually detailed by the engine and hull drafting rooms.

Protection of Conductors. **Conduits** are used below decks where the electric leads may be subjected to mechanical injuries and on exposed weather decks. **Wooden moulding** is permitted in living quarters. Wooden mouldings must be designed in accordance with the **National Electrical Code** for separation of the conductors from each other and from the steel. Moulding grooves receive two or three coats of shellac before the electric wires are laid in them. The Marine Committee unanimously disapprove the use of both conduits and wood moulding.

Practice dictates certain **fundamental requirements for conduit work**, as follows: Hot galvanizing is not permitted, all bends must be made cold, no pipe fittings allowed, couplings to have both ends right hand thread, unions to have spherical ground joints with inner surface perfectly smooth, and must be installed at bends, slip-joints and loop boxes to be installed on long leads, conduit connections must be made up with white or red lead, ends of conduit must be reamed before assembling, number of wires pulled into a given sized conduit must be proportioned so as to avoid the use of grease or occasion strains on the wire, only double-braided wire can be installed in conduit, all mechanical work on the conduit must be completed before the conductors are pulled in.

Conduits, although they provide against mechanical injury to the conductors, have not proved entirely satisfactory in preventing an accumulation of the waters of **condensation.** The conduits passing through different compartments at varying temperatures produce in themselves accumulations of moisture which in time affect the rubber insulation on the conductors. This effect may become so pronounced and the intensity of the grounds on the conductors may be such as to require the re-wiring of certain compartments every three or four years. Many ingenious but unsuccessful suggestions have been made and tried in order to overcome this defect. Experiments in Europe were carried on for the perfection of a special conduit alloy which would prevent the formation of condensation. Either the material itself was too expensive to manufacture in the shape desired or the experiments made were not successful. The material was never introduced into practice in this country.

Lead-covered, steel-armored conductor is approved by the U. S. Navy, and this design has been successfully adopted for merchant marine work. The design consists essentially of a lead sheathing over the approved rubber-covered braided wire with a taping over the lead sheath, and over all a fine galvanized-iron, steel piano wire of several strands woven as in basket making; hence its designation as **"basket weave."** This type of protection provides

the necessary flexibility as well as protection to the conductor during installation, permits it to be bent sharply around protrusions in the steel hull, prevents entirely any condensation within and secures complete watertightness of the conductor; it restricts grounds to terminal points and connection blocks.

With this type of protection it would seem advisable to eliminate wooden mouldings for use on shipboard, this especially in view of the fact that the lead-covered, steel-armored conductor makes a perfectly presentable job in the living quarters of the ship and provides safety, being installed in the open and readily inspected for repairs. It is economical to install on wooden partitions and wood enclosed portions of the steel vessel and makes a most excellent material for the electrical installation on wooden vessels.

The first cost of commercial lead-covered, steel-armored conductors does not greatly exceed the combined first cost of conduit and wire.

Type of Conductors. All copper conductors larger than No. 12 B. & S. gage are required to be stranded and no single conductor smaller than No. 14 B. & S. gage solid is permitted in the installation. The grade of copper used is such that the specific purity of the copper conductor is no longer of any great importance. The stranding of the conductor is most important in marine work especially in large sizes for feeders of any great length as it is difficult to thread the feeders through the various compartments and divisional bulkheads without running a great risk of impairment to the conductor. It also gives the necessary flexibility for forming the conductor around the various obstructions either of the hull structure or other installations. The stranded copper conductor is **insulated** by a wall of rubber following the general practice of shore installations except that it is customary to specify a particularly good commercial grade usually in excess of the requirements of the **Board of Fire Underwriters.** A heavier wall is usually provided than that required for the low potential used (110 or 220 volts).

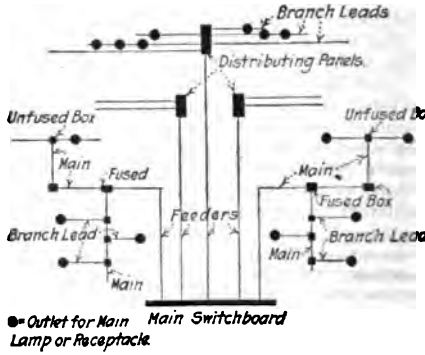

FIG. 1.—Lighting Distribution.

Distribution. The **2-wire full metallic system** of distribution is now the established practice. The **single-wire system** has been used but would not be considered to-day especially in view of the general use of radio-telegraphy. The distribution on shipboard follows generally centralization of control as common in land practice. **Feeders** are led from the main switchboard to distribution cabinets from whence are taken branches to groups of lights. **Special feeders** are led to the searchlight, radio-telegraph apparatus, and special motor applications. The branches from the distribution panels are provided with the necessary cut-outs and the panels are usually enclosed in sheet metal boxes. In the development of the application of **electric motor auxiliaries** the introduction of a commercial system of 3-phase alternating current brings with it a **3-wire distribution.** If such motor systems develop

into higher potentials undoubtedly the lighting system will be reduced through transformers to a lower potential which will still maintain the 2-wire distribution.

With conduit protection for the conductors the practice to-day is to install two separate single conductors carrying them both in the same conduit. So in the case of 3-phase alternating current, three conductors would be installed in the same conduit. With lead-covered, steel-armored conductors the practice on a 2-wire distribution is to use **twin conductor cables** made up with jute to produce circularity, and for the 3-phase to use the three conductors. In alternating current practice the conductors must be run together in order to prevent inductive effects.

Generating Sets. The present merchant practice is to use **reciprocating engines** for marine generating sets. The marine engineer has always had a preference for this type of apparatus. The designers have endeavored to supplant the reciprocating engines by means of the steam turbine. Direct-connected steam turbine generating sets were introduced several years ago and the design then employed was not found by certain owners to give as good results in service as reciprocating engines. Attempts were then made to introduce the geared turbo-generating set but these installations were very few and the results of their service have never been published. Where weight and space are governing factors turbo-generating sets are used in the U. S. Navy.

Table 1. Marine Generating Sets

Geared turbine generators						Direct-connected turbine		Reciprocating engine		
A.c., kv.-a.	Speed of turbine	Speed of generator	D.c., kw.	Speed of turbine	Speed of generator	Kw.	Speed of set	Kw.	Speed of set	Weight of generator
30	7200	1200	15	7200	1200	5	4000	5	500	
50	7200	1200	25	7200	1200	7½	4000	7½	450	1350
75	7200	1200	35	7200	1200	10	4000	10	450	1775
100	7200	900	50	7200	1200	15	3600	12½	450	1875
150	7200	900	75	7200	900	....	....	15	400	2525
200	7200	1200	100	7200	900	....	....	17½	375	2625
Note: All standard a. c. voltages, *i.e.*, 240, 480, 600, 2300 volts.			150	7200	900	....	....	20	350	2700
			200	7200	900	....	....	22½	350	2725
			Note: All standard d. c. voltages.					25	350	2750
								30	325	2800
								35	325	3000
								40	300	4000
								50	275	6250
								60	275	6450
								75	275	7500
								100	300	8500

There are probably two general reasons for the consistent preference for the reciprocating generating set (1) the familiarity of marine engineers with reciprocating engines, (2) the insignificance of the increase in efficiency of the unit due to the fact that the exhaust from these sets is used for heating the boiler feed water. These sets are arranged for both condensing and non-condensing operation. The exhaust is carried either to the boiler feed water heater, the main or auxiliary condenser or to the atmosphere, thus making for flexibility under emergency conditions.

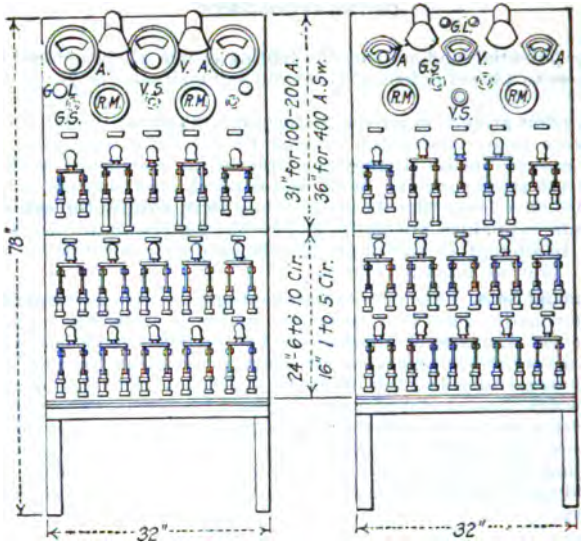

FIG. 2. FIG. 3.

Switchboards. Two Circuit Generator Sections with Fuses; Ten Circuit Feeder Sections.

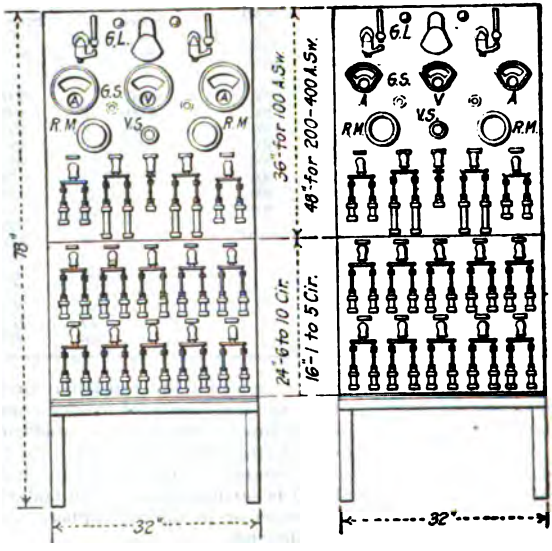

FIG. 4. FIG. 5.

Switchboards. Two Circuit Generator Sections with Circuit Breakers; Ten Circuit Feeder Sections.

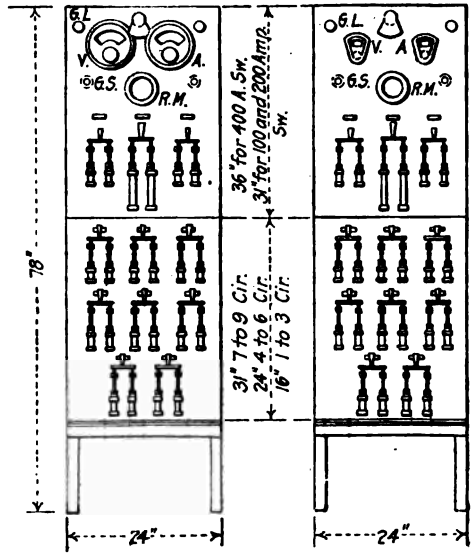

FIG. 6. FIG. 7.

Switchboards. Single Circuit Generator Sections with Fuses; Eight Circuit Feeder Sections.

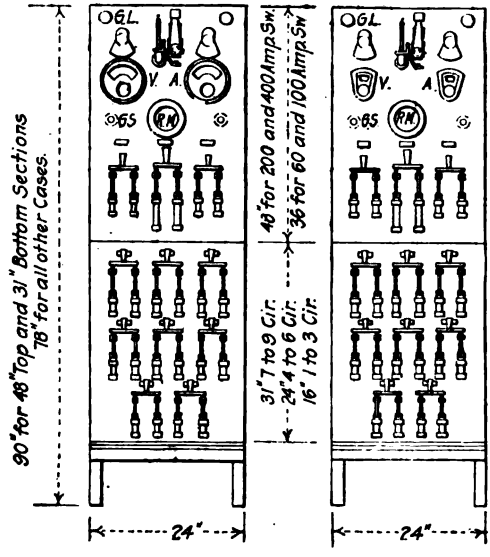

FIG. 8. FIG. 9.

Switchboards. Single Circuit Generator Sections with Circuit Breakers;

The **steam connections** to these sets must be made with flange connections and not with threaded connections as practised on land, this in order to reduce the number of steam joints, also the possibility of leaky joints due to the vibrations set up in the hull of the vessel by the main propelling engines. Generators are designed with special

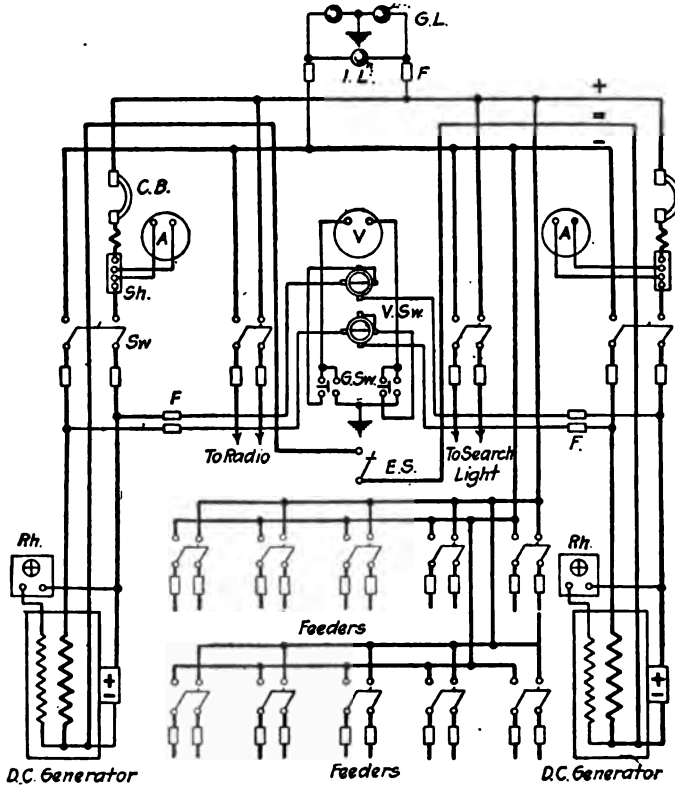

FIG. 10.—Switchboard Wiring.

A = Ammeter	**I.L.** = Illuminating lamp
C.B. = Circuit breaker	**Rh.** = Rheostat
E.S. = Equaliser switch	**Sh.** = Shunt
F. = Fuse, Cat. No. 111382	**Sw.** = Switch
G.L. = Ground detector lamp	**V.** = Voltmeter
G. Sw. = Ground test switch	**V. Sw.** = Voltmeter switch

precautions to prevent oil from creeping along the shaft, are provided with non-corrosive parts throughout, and are usually wound for flat compounding. Reciprocating engines are provided with both gravity feed and forced lubrication so that they are capable of running for fairly long periods without attention. The navigating lamps

permit the use of gasoline engine generators or storage batteries for an auxiliary means of emergency lighting and emergency supply for the radio-telegraph but, although there has been some development of such apparatus, sufficient data is not available to determine its adaptability to this service. The storage battery is entirely adequate for this purpose.

Marine switchboards, in general, are similar to shore boards; the material is slate, and all fittings must be non-corrosive; they must be as compact as possible because of restricted space available. Circuit breakers, and other switchboard apparatus must not extend beyond the switchboard top and must be non-corrosive.

Two general methods for **connecting generators** to the distributing circuits are (1) generators are connected to separate bus-bars with double throw-switches so that loads may be arbitrarily divided between the sets; (2) generating sets connected to run in parallel so that they automatically share the load. In small installations generator connections are made through fused knife-switches, but for large generating sets manually operated double-arm circuit breakers are substituted.

Switchboards for Merchant Ships, as furnished by the General Electric Co., are illustrated in Figs. 2 to 9. Panels are of the sectional type mounted on a frame made up of 2½ by 1½ by ⅜-in. angles with a horizontal angle for supporting the weight of the panels independent of the supporting bolts. The upper portion is the generator section and the lower part is the feeder section. The material of the sections is dull black marine finished slate.

Generator sections are provided for controlling either one or two 125-volt machines in capacities up to 25 kw. In addition to the switches and instruments for the generator circuits each generator section contains two 60-amp. feeder switches with 60-amp. fuse clips for controlling the search-light and radio circuits.

Feeder sections are available with various numbers of circuits which can be combined with the generator sections, as indicated, to take care of various conditions which may be required in individual cases. In order to provide a rugged construction all feeder switches are of 60-amp. capacity.

Each switch is labeled with the name of the circuit. All instruments are calibrated at the factory to be accurate to within 1 per cent of full scale reading. All panels meet the requirement of Lloyd's rules, and those with circuit breakers will meet the requirements of the American Bureau of Shipping.

LIGHTING SYSTEM

Lighting Circuits. In an ordinary freight vessel lighting circuits are extremely simple and are led as described on page 1366 under *distribution.* In **passenger vessels** the circuit is so laid out as to provide each stateroom from two different feeders so that light will be assured at all times and also to admit of repairs with minimum inconvenience. **Branch leads** from the distribution panels are tapped and lights taken off from **sub-branches** or the branch leads are carried around and the fixtures attached directly to the branch leads. This latter arrangement of circuits is called **stringing lights** and permits of the use throughout of the same size wire, in all cases No. 14 B. & S. gage.

In **engine room, boiler room** and like places, the taps are made in cast iron junction boxes. In **living quarters** they are made in the wooden mouldings, and in a lead-covered, steel-armored installation they are made

at the fixture outlet in the wooden block which also serves to secure the fixture. Recently **condulets** have been used instead of the round pattern cast iron box shown in Fig. 11. In watertight construction both these fittings must be supplied with a rubber gasket and sufficient holding screws to give proper compression to the cover.

Practice differs in the method of making the tap on the branch leads. Some experts consider that a twisted mechanical connection of copper wire well soldered, taped, and varnished, satisfies all requirements. Others employ a connection block, fastening the wire by means of screws to the connection block, thus avoiding the practice of splicing or the use of solder. Both practices are mainly dependent upon good workmanship. If the connection block is properly designed and sufficient slack is arranged for in the branch box, the working of the vessel will not cause the wire to slip or break away from contact, on the other hand a spliced joint avoids such a condition if it be well made.

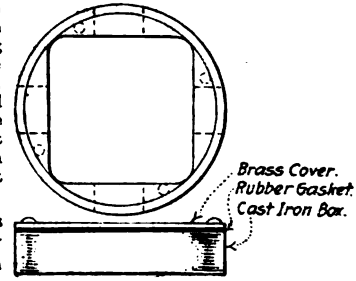

FIG. 11.—Branch Junction Box.

Non-watertight fittings such as switches, receptacles, etc., are usually identical in design and construction with those used in land practice. **Watertight fittings** are usually Navy standard in the best practice. There has been only one design of receptacle for watertight work that approaches the satisfactoriness or reliability of the Navy standard watertight receptacle. This receptacle provides an automatic means of closing the opening into which the plug enters, and has the added advantage over the Navy standard receptacle of insuring against failure to replace the cap over the plug opening.

In accordance with law, passenger steamers are provided with circuits in passageways and living quarters from the emergency generating set. In very large passenger vessels a special night circuit known as a **police circuit** is arranged in a similar manner. In ocean going freight steamers lighting circuits are carried into the cargo holds and terminate in receptacles. These circuits are usually fed from distribution panels convenient to the engine room so that, when the holds are completely filled, these particular circuits can be cut out, in order to insure that all electric lines in the cargo spaces are "dead" while the ship is under way.

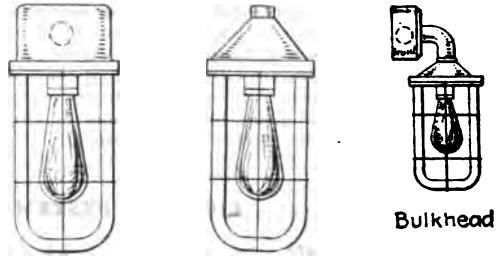

FIG. 12.—Steamtight Fixtures.

Lighting fixtures. In the machinery spaces and on exposed decks **steamtight fixtures** are installed. These fixtures are similar in appearance to the Navy standard fixture except that the guards are much lighter and the fixtures are frequently made of a smooth cast iron. A variety of forms suit various

locations and mountings. Three well-known types, Fig. 12, are so-called from the facility with which they can be used in the installation, namely, **deck fixtures** for mounting directly overhead on the deck, **bulkhead fixtures** arranged with a short bracket, flange and box for mounting on the bulkhead, and **drop fixtures** arranged with a centrally tapped hole for directly attaching to the end of the conduit so that it may be dropped from the deck above and adjusted to any desired height. The steamtight fixture consists mainly of three principal parts, a box for containing the lamp socket, a glass globe (in the merchant service arranged with a screw thread) and a guard which screws on the box and protects the glass globe from mechanical injury. This fixture is very adaptable and can have modifications to suit a variety of purposes. For example, it can be used on outer decks and be arranged with a screen so as to prevent reflection from annoying the quartermaster at the wheel.

In **vessels designed for** the alternate carriage of **cattle and bulk cargo between decks,** the rules of the Classification Societies, though permitting permanent fixtures for the cattle, require that these fixtures be entirely removed and the opening securely closed when bulk freight is carried in the same spaces. The steamtight fixture may fulfill these requirements by removing its guard globe and lamp and closing it with a screw cover. Special fixtures are designed with a cast iron hinged cover which encloses the entire fixture. In **packet freight steamers** the lower holds and between deck spaces are lighted with permanent fixtures of the steamtight type located well up between the beams and further protected by means of thin strips of steel placed longitudinally from beam to beam in wake of the fixture. For the **ocean going freight steamers** portable clusters, bowl shaped reflectors painted white and containing from 4 to 6 lamps, are lowered into the holds while loading or unloading by night. Such a fixture also serves for a gangway light and for a propeller light to prevent small craft from damaging the propeller while the vessel is docked. **Portable hand lamps** are provided for use around the engine and boiler rooms. A very light and simple device which satisfies all the requirements can be made by fitting up an ordinary weather-proof moulded socket with a light protective guard.

In **living quarters** a light metal fixture is used which consists usually of a very plain base, sufficient to cover the lamp receptacle, and a bracket for holding a shade or globe. This fixture must be compact as the head room in the ship's cabin is restricted. Staterooms, which are painted white enamel and are small, usually require but a single unit for good lighting. In the officers' cabins a bracket desk light is provided. In the more elaborate **passenger steamers** and especially in the de luxe apartments, increased lighting fixtures, such as bunk reading lights, bracket lights over the shaving mirror and cheval glass are provided. The light metal fixture, if not of an especially good quality, will soon be affected by salt air and for certain passenger vessel work frequent renewal is necessary. In order to reduce this cost attention has recently been given to the use of **belleek china fixtures** not only for the lighting fixtures but also for receptacles and switches. This practically eliminates the use of metals except for the joiner work. The belleek china fixtures are very easy to keep clean, are neat in appearance, are tough in texture and are not easily broken. **Automatic door switches** are required on all doors to insure lights are out when an occupant leaves a compartment.

Running Lights. In accordance with the navigation laws every steam vessel carries the following lights: mast head, port, starboard, stern and range. A separate feeder is led from the main switchboard to a control

panel called a tell-tale located in the pilot house. Each of the running light lanterns is provided with two incandescent lamps. These lamps are so wired to the tell-tale board that if one of the lamps goes out a visible (single lamp) and audible (buzzer) signal is given on the tell-tale board and a switch can be thrown to place the reserve lamp in circuit. In the event that there is anything wrong with the reserve lamp the same signals will continue. As the electric leads to the navigating lights are not only in exposed positions but mounted on structures that are subjected to continuous vibrations while the ship is under way, these leads are liable to derangement. As the tell-tale board is small the connections are necessarily very closely spaced. These conditions may give rise to bad short-circuits which may result in the utter uselessness of the circuits; they are safeguarded by the introduction of fuses on both sides of the throw-over switch. These fuses, very easy of replacement, are mounted on the front of the board. Blowing of a fuse causes the same signals as the derangement of a lamp.

There have been a number of designs other than that described for this purpose, but they have not operated as well in service and in every case, it is understood, have been abandoned. In the English practice such a device is not usual. The practice in the English merchant marine is similar to that in the Navy, namely, an ordinary distribution panel located in the pilot house.

Lamps. Circuits for the lighting system are calculated on the basis of the old 16 c.p. carbon filament incandescent lamp but the tungsten lamp is now universally used. For the ordinary cargo steamer the 25-watt lamp is sufficient for general lighting purposes. In small passageways even a smaller sized lamp could be used. Forty watt Masda lamps are sometimes fitted in cargo clusters or hand portables. The same size lamp socket also permits the use of 60-watt lamps. Forty watt lamps are used in the running light lanterns. For lighting the engine room hatch even larger sizes of the Mazda lamps, either 250 or 500 watts, are sometimes used. It was at first doubted whether the tungsten filament lamp could be successfully used on shipboard due to the vibratory action set up in the hull structure by the propelling machinery, but it developed in the manufacture of the 110-volt Masda lamp that the filament was as strong as or stronger than the unanchored carbon filament that was then successfully used in service. The 220-volt Masda lamp is not satisfactory for marine use.

Methods of Illumination. The modern methods of illumination do not apply to the lighting systems of the ordinary freight vessel. There is really no question of proper illumination strictly involved. The problem is a practical one connected with distribution of light, and this fact has materially contributed to the successful application of the incandescent lamp. The question of general illumination arises very infrequently in merchant practice. Most portions of the vessel requiring light are subdivided into relatively small compartments having highly reflective surfaces so that the coefficient of absorption is extremely low; consequently the illuminant is working under most advantageous conditions, and the loss in lumins from the light source is an insignificant amount and eliminates the question of illuminating efficiency. Even with the smallest commercial lamp the lighting of the compartment would be excessive from the standards of illuminating efficiency.

The only explanation for the insistence of those who live on board ship on relatively large wattage, with accompanying high lumin intensity, must be psychological. In elaborately decorated river excursion steamers and in transatlantic passenger steamers illumination problems are more in order. Such vessels in their interior decoration strive for imitation of land conditions, and here, as on land, indirect fixtures and semi-indirect fixtures, cove lighting and concealed lighting have a field.

Searchlights. The merchant freight vessel usually carries one 18-inch pilot house control searchlight. The 18-inch projector has long been accepted as standard, it being the minimum size that will give sufficient projection and the maximum size necessary. The searchlight is useful in picking up buoys and when docking or undocking at night. The horizontal electromagnet arc lamp is common practice. The ship's voltage of 110 is reduced through an adjustable resistance usually located in the dynamo room near the main switchboard. A special ammeter is mounted on the main switchboard, and by means of a voltmeter switch the machine voltmeter is also available for use in adjusting the searchlight circuit. The searchlight feeder is carried directly to the pilot house and is controlled there through a double-pole single-throw lever knife-switch. The arc lamp searchlight gives good service on shipboard, but as its use is infrequent it requires more than ordinary attention.

In vessels equipped throughout with **alternating current** it is not considered expedient to rely upon the poor efficiency of the alternating-current arc lamp. Here the **incandescent searchlight** is substituted for the customary arc lamp. The incandescent lamp 18-inch projector, for distances up to ¾ of a mile, will operate as successfully as the arc lamp of the same size. Beyond this it is not as efficient. The efficiency of the two types differs widely The arc lamp 18-inch projector consumes approximately 35 amperes and the incandescent type approximately 4½ amperes.

On vessels carrying radio-telegraph apparatus it is essential that the portable leads connecting the searchlight lamp to the connections in the base be covered with some metallic protection grounded to the hull of the ship to avoid making the lamp inoperative due to the inductive influences of the radio-telegraph. Large passenger steamers generally carry two or more searchlights. Excursion steamers plying on inland waters carry a size larger than 18 inches in order to throw a broader beam on the shore.

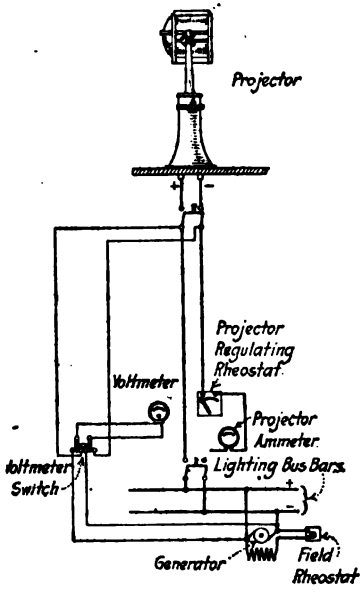

FIG. 13.

Dredges and canal boats usually carry two searchlights one forward and one aft, and these projectors are arranged with special lenses to comply with the law so that they can divide the beam thus retaining a dark area in the middle of the canal. (Rules and Regulations of the Suez and Panama Canals.)

Searchlight Connections. Diagram of connections, Fig. 13, shows a very desirable method of connecting both the hand and pilot house control projectors, as it provides for the location of all instruments and means of adjustment in the dynamo room under the eye of the attendant.

With the voltmeter switch the voltmeter may be used to show the generator voltage, and also by means of two pilot wires passing to the projector itself, the voltage at the projector. This latter connection is used when adjusting the lamp in case of disarrangement of the feeding magnet spring.

POWER SYSTEM

Motor Circuits. All feeders are led directly from the main switchboard without break to the control apparatus of the motor. A switch and cut-out is required in the line to protect the starting box and motor. **Drop in voltage** in the line does not enter the problem as the distances are short. The conductor sizes must allow for starting requirements. Motors of 0.25 H.P. or less can be directly connected to the line.

In direct-current practice at the standard voltage of 110 **the 2-wire system is used** and a separate wire is run for each side of the circuit, or for very large motors several wires are run for each leg.

In alternating-current practice on the 3-wire, 3-phase system the conductors are made up in the form of a cable in order to avoid induction troubles. If the motor application is large several cables may be used in parallel.

Motor Controller. For small motor applications up to 10 H.P. the usual land type of starting box enclosed in a sheet metal case is used. Above 10 H.P. drum controllers are used. For special applications, where the apparatus is exposed to salt air and salt water, the drum controller is mounted in a watertight case. Resetting devices and contactors for special functions are employed as on land. At the present time for simplicity of operation of such deck machinery as winches, mechanical levers similar to those used on steam winches actuate the controller. Spare parts are carried but it is desirable to make such devices uniform so that in case of the derangement of one piece of apparatus the controller can be shifted from another application.

Motors. At present commercial land motors have been applied with some degree of success. Experiments are being conducted with both watertight and non-watertight types on exposed decks. If **non-watertight** motors are used the first cost will be reduced to the shipbuilder but he in turn will have to place protective metal coverings around the motors. The tendency is for an insistence upon a few special requirements of the manufacturer without affecting too much the present standard commercial designs. It is important that oil should not run along the armature shaft when the ship heels and the windings should be protected by compounds that will make them impervious to moisture. No parts must be used that will suffer corrosion by salt air or salt water.

Direct versus Alternating Current Motors. All types of motors are applicable on shipboard. For certain types of vessels, such as bulk oil carriers, it is preferable to avoid the use of commutating machinery although if such motors are fully enclosed and carefully ventilated, there is no objection. For this particular type of vessel the motor is not any more serious than the controller and in some designs the control apparatus has been located in the engine room so as to insure against oil explosions.

Direct-current Motors have the advantage of good speed control which is easy to obtain by use of resistance methods and also the braking action of dynamic control. For hoisting machinery, such as anchor windlass, capstan, and deck winches, such control characteristics are of great value. The disadvantage resides in the increased weight and increased cost. This is magnified in designs where the electric propelling machinery is alternating current as in this case it requires direct-current generating sets instead of utilising static transformers.

Alternating-current Motors have not as good speed changing characteristics as has the direct-current motor, but the **induction motor** avoids the use of commutators and provides very good torque characteristics. In electrically propelled ships, auxiliaries so equipped increase the overall efficiency of the plant. For steering gear equipment, if the speed changes are obtained through hydraulic gears, the induction motor runs at constant speed and functions as required by the load. Hydraulic gears are furnished in convenient form suitable for any of the auxiliary applications and this would lead to the advantages gained by reduction in cost, reduction in weight, and interchangeability. This latter feature can be attained economically by selecting one average sized motor and applying it to all auxiliaries.

SIGNALLING SYSTEMS

Engine Telegraphs. The signals from the bridge and pilot house to the engine room are purely mechanical in character. This system consists of transmitters connected by low brass wire and triple-link chain passing over brass pulleys. A reply system assures that the signal is properly received. In some cases an electrical attachment is made so that an alarm is sounded in the engine room in the event that the engine moves in an opposite direction to the signal given. The instruments are electrically lighted.

Other **Mechanical Telegraphs** similar in their operation to engine telegraphs are used for docking ship, steering, rudder indicator, and other functions.

Bell pulls or **gong and jingle signals** are installed in case of derangement of the main engine telegraphs. Due to the spring action of these devices, low brass wire is not suitable and copper wire is substituted.

Telephones. In vessels in which the distance from pilot house to forward engine room bulkhead is more than 150 ft. the navigation laws require a telephone. Ordinary land telephones are not satisfactory for marine practice. There are manufactured special watertight, loud-speaking phones which have been successfully used for many years.

Voice tubes of brass 1½ in. in diameter are used to connect pilot house and engine room, navigating bridge and pilot house, bridge and captain's cabin, the engineer's cabin and starting platform. These tubes are provided with whistle mouthpieces for signalling purposes.

Call bell arrangements on ordinary cargo steamers consist of an annunciator in the galley with push buttons from the officers' mess and officers. cabins. This system becomes more elaborate and extensive in large passenger steamers Special alarm bells with actuating devices in the pilot house are required by law for passenger or crew sleeping quarters where no guards are maintained day and night.

Fire alarm systems are not usual on ordinary cargo ships. On combined cargo and passenger steamers such systems are usually installed. These systems are usually special in character and the work of installation is done by the special company taking the contract.

Electric Whistle Operator. Although provision is always made to actuate the main steam whistle by mechanical means there are now on the market several good methods of electrically operating the main steam whistle valve. Most of the systems are arranged for "at will" as well as automatic operation, usually by clock mechanism. Contact may be so adjusted that the whistle can be blown in the regulation manner for foggy weather as well as adjusted for giving the proper signals when the vessel is towing or being towed.

Submarine Signals. The **Submarine Company** manufactures and provides the necessary apparatus, consisting in general of brine tanks for location in port and starboard fore peak and the necessary telephone receivers in the pilot house, which will detect signals for sunken bells located along the coast of those countries having much navigation. This signalling apparatus also provides means of detecting the approach of other vessels and the direction of approach during thick weather.

Radio-telegraph. All the various radio-telegraph apparatus are practical for installation on merchant vessels. The important point is to be careful in designing the electric equipment to prevent inductive effects on all leads. Special precautions must be taken in providing ample ground connections for the wireless antennæ and all metallic stay lines of the rigging must be grounded to the hull.

A separate and continuous feeder is led from the main switchboard to the wireless motor generator and a similar direct line is run from the emergency lighting set on vessels where this is a requirement.

SECTION 13

LUBRICATION AND LUBRICANTS

BY

G. R. ROWLAND, Supervising Engineer, The Texas Co., Mem. A.S.M.E., S.A.E., A.S.A. and N.A.S.E.

CONTENTS

	Page
Lubrication	1381
Friction	1381
Lubricants:	
Mineral Oils	1383
Animal, Fish and Vegetable Oils	1383
Fixed Oils and Compound Oils	1383
Greases	1384
Graphite	1385
Selecting Lubricant for Service	1386
Engine and Machine Oil	1386
Forced Lubrication Oils	1388
Marine Engine Oils	1388
Cylinder Oils	1391
Air Compressor Oil	1392
Ice Machine Oil	1392
Motor Oils	1392
Physical Constants of Oils:	
Viscosity	1394
Flash and Fire	1394
Emulsification Test	1395
Water and Sediment	1395
Cold Test	1396
Specific Gravity	1397
Color	1397
Odor	1398
Carbon	1399

LUBRICATION AND LUBRICANTS

BY

G. R. ROWLAND

Lubrication can perhaps be best understood by considering the case of a spindle revolving at a speed of 10,000 revolutions per minute in a bath of oil. The oil film immediately touching the suface of the spindle will revolve in the bath at nearly the surface speed of the spindle, the oil clinging to the spindle through its adhesive power. Succeeding films or layers of oil revolve at slightly less speed, while the film touching the sides of the bath will be as nearly stationary as its cohesiveness and the general movement of the oil will allow. The friction of the particles of oil sliding past one another is referred to as **internal or fluid friction.**

If a gradual side pressure is applied to the spindle, forcing it toward one side of the bath, some of the oil will be pressed out, and when the pressure becomes great enough the metal surfaces of the spindle and side of the bath or base will actually come together. This will produce heat and wear from metal friction. If the pressure is constantly kept at the point where the metal surfaces actually come together, an oil having more body or viscosity should be used. The heavier-bodied oil will resist the pressure and keep the surfaces apart, and the spindle will again turn in the centre of the bath. As long as the oil is heavy enough to keep the metal apart no metal wear takes place. Again increasing the side pressure on the spindle will force the heavier oil out, and heating will again occur. To overcome a steady pressure of this nature, a still heavier-bodied oil must be used in order to keep the surfaces apart.

In the above example the frictional heat of the spindle base will be slightly increased as the heavier oil is used under heavier pressure, but in no case will the heat be as high as that resulting from the destructive abrasion of metal on metal due to the use of too light an oil. Also, the **minimum amount of power will be required to operate the spindle with the lightest oil in use,** and the power required will slightly increase as the heavier oil is used; but, as in the case of temperature, the power will at no time be as high as when the surfaces are together through too great pressure or too light an oil. This case illustrates the **effect of pressure upon the oil film and upon the viscosity or body of the oil.** The same rules apply generally to all bearing lubrication.

Friction is one cause of loss of power in an engine. In all engines there are so many moving parts that it is of the greatest importance that friction should be reduced as much as possible. This is done (1) by making the surfaces in contact smooth and of ample size, (2) by making them of different metals, and (3) by the use of suitable lubricants. Most of the marine engines in use at the present time are vertical multiple cylinder machines built without fly-wheels or governors. Up until about twenty years ago the moving parts of marine engines were lubricated with various kinds of animal oils, principally lard oil. Of late years, or since the introduction of mineral oils, animal oils have been almost completely eliminated due to the fact that there has been a vast improvement in mineral oils and also that lubrication is to-day upon a more efficient basis. In most all reciprocating engines the oil is reclaimed, filtered and used over again which is impossible to accomplish to any great extent with animal oils.

Coefficient of Friction is defined as the ratio between the total friction and the total load. Theoretically, total friction in a perfectly lubricated bearing is independent of the load that may be put on the bearing. Practically this is approximately true. Fig. 1, from "*Practical Lubrication*,"

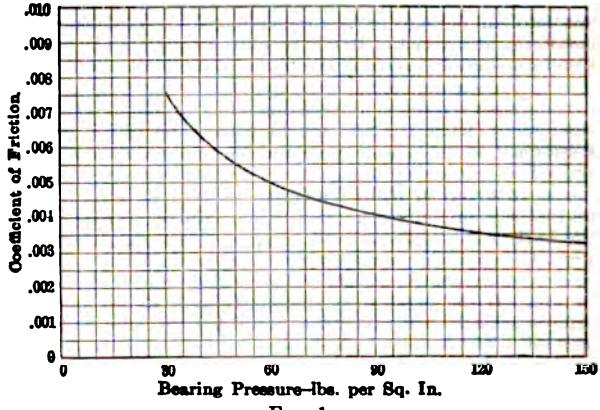

FIG. 1.

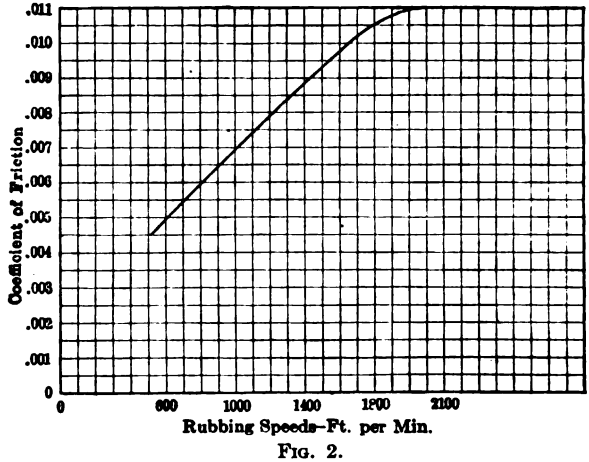

FIG. 2.

Bryan, Jour. A.S.N.E., Vol. 27, shows how the coefficient of friction drops very appreciably with increase of bearing pressure.

The danger of increasing bearing pressure is not that it directly increases the friction, but that it tends to squeeze out the oil from the bearing, and this results in increased oil consumption. Where forced lubrication is used this will increase the amount of oil circulated with the attendant losses.

"**Increasing the Rubbing Speed of the Journal** greatly increases the friction. The curve in Fig. 2 illustrates graphically the amount of this increase. It will be noted that at rubbing speeds above 2100 feet per minute the curve appears to flatten. This speed is exceptional, however, and at the speeds ordinarily encountered in the Service [Navy] the friction increases greatly as the rubbing speed increases."—*Bryan, "Practical Lubrication," Jour. A.S.N.E., Vol. 27.*

LUBRICANTS

Good lubricants are substances, semi-fluid or fluid, capable of forming and maintaining films of sufficient thickness between two rubbing surfaces to prevent actual friction between the surfaces, substituting for it the fluid friction of the lubricant itself. **Marine lubricants** may be classed as mineral oils, animal oils, vegetable oils, and greases.

Mineral oils are manufactured from crude petroleum and shale. The petroleum lubricating oils used in the United States are made from crude oils which may be classed in the following manner according to their natural characteristics: (1) **Paraffine oils,** composed chiefly of hydrocarbons of the paraffine series and found chiefly in Pennsylvania, West Virginia and Eastern Ohio; (2) **asphaltic oils,** composed of hydrocarbons of the napthene series, found principally in California, Texas and Louisiana, and (3) **Semi-asphaltic oils,** found chiefly in Kansas, Oklahoma and Illinois.

Lubricating oils made from the paraffine crudes, have light gravities, high flash and fire tests, high congealing points and low viscosities unless blended with steam cylinder oils. Asphaltic base oils have heavy gravities, lower flash and fire tests, congeal only at temperatures below zero degrees F., and can be made in any desired body without blending.

Animal, Fish and Vegetable Oils are sometimes used, either alone or as usually is the case, as a compounding for mineral oils. The more important follow:

Whale Oil	made from whale "blubber,"
Lard Oil	made from the fat of hogs,
Neats Foot Oil	made from beeves hoofs,
Tallow Oil	pressed from beef tallow,
Sperm Oil	made from head of sperm whale,
Cottonseed Oil	made from cottonseed,
Rapeseed Oil	made from rapeseed,
Castor Oil	made from the castor bean,
Rosin Oil	made by destructive distillation.

Certain of the above, such as lard and tallow oils, not only have value when compounded with mineral oils, but for some purposes, good results can only be obtained by so compounding.

Fixed Oils and Compounded Oils. Before the introduction of mineral oils for use in steam cylinders **tallow** was used exclusively, being melted in a tallow pot which was kept on top of the steam chest. After the tallow was properly melted, it was introduced into the cylinder through a tallow cup, usually fitted with a valve and screw cover, and located on top of the steam chest. Lubrication by this method was crude and very intermittent. However, it probably suited the times and the condition of the engines. **Sperm oil** was formerly used, both mixed and unmixed, with mineral oils, for light machinery and spindles. **Porpoise jaw oil** is still used for the lubrication of watches and delicate mechanism, for which a thoroughly non-drying and fairly limpid oil having an extremely long life is required.

The lubricating properties of animal and vegetable oils are excellent, as the heavier oils, such as castor, are able to withstand very high pressure, and the lighter oils, such as sperm, have very low coefficients of friction. Their disadvantages are (1) high cost, (2) oxidising tendency, which necessitates frequent cleaning of machinery and (3) acids form that pit and corrode.

Nearly all lubrication of a general nature can now best be cared for by a straight mineral oil of the proper viscosity. When properly manufactured such an oil can be used over and over again, which is not the case with fixed oils.

Greases. In general lubricating practice there are certain mechanical conditions which cannot be handled to advantage by lubricating oils, and for which it is necessary or desirable to use a semi-solid lubricant called grease. Grease, as the term is used for commercial purposes, is a **mineral oil thickened with soap** made from an animal or vegetable oil, such as tallow oil, horse fat, lard oil, cottonseed oil, rapeseed oil, palm oil and rosin oil. In practically all greases mineral oil constitutes by far the larger part, frequently as much as 90 per cent of the finished product, the **consistency of grease** being from soft to hard, depending upon the percentage of soap in the mixture.

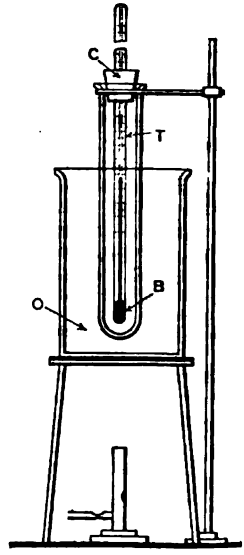

FIG. 3.

Grease should be used only where oil cannot be retained in and around the bearing, or where the bearing is too inaccessible to be oiled regularly, or with very low speeds and high pressures. In all cases of high speed and wherever oil can be retained in and around the bearing, grease should not be used because it will increase the bearing temperature and the power consumption. Greases should not be used on piston rods or in cylinders as they will liberate fatty acids which will eventually get into the boiler.

Cup greases are primarily intended for use in **compression cups** which force the grease directly to the bearing surface. As their melting point is comparatively low, but little frictional heat is required to make them flow. Where compression cups are not used, frictional heat of the bearing must be depended upon to make the grease flow. **Pin cups**, having a copper wire to transmit heat from the bearing to the grease, may be used or the grease may be introduced direct to the bearing through slots. **Special greases**, of the same consistency as cup grease but with a higher melting point, should be used for relatively high temperatures. The required hardness and melting point increases with the bearing temperature expected, and a **very hard grease** with a high melting point is used for extremely high pressures, and for extremely high surrounding temperatures.

The melting point of grease is the temperature at which the grease begins to flow in liquid form. There are several methods of determining the melting point and each method will give a different temperature as the melting point. The following method is now being used very generally with satisfactory results, see Fig. 3. Some of the grease to be tested is spread with the fingers onto the bulb *B* of a thermometer which ranges from about 100 deg. F. to about 400 deg. F. The thermometer is then fitted by means of a cork *C* into an empty test tube *T*, so that the thermometer hangs free. The test tube is then placed in an oil bath *O* heated by a Bunsen burner, and the flame so regulated

that the temperature will rise from 2 to 4 degrees per minute. When a drop forms on the bulb of the thermometer and falls to the bottom of the test tube, the temperature is recorded as the melting point of the grease.

A grease is never better than the mineral oil used as its base. In other words, the **lubricating value of a properly manufactured grease depends upon the mineral oil** used in its manufacture. The soap acts as a thickener for the mineral oil, and it is the latter that does practically all of the lubricating.

If a grease is not properly manufactured, the oil will frequently separate under service conditions, leaving the soap in the cups and on the bearings in solid or powder form. This is caused by a defect in manufacture that cannot be discovered by the casual inspection of a sample of grease. Cup greases and general lubricating greases should not separate when subjected to working temperatures, and when cooled should go back into a perfectly homogeneous mixture.

In the **manufacture** of most greases the animal or vegetable oil is heated to the required temperature in a steam jacketed kettle. Cream of lime or other alkali dissolved in water is then added, and this mixture is boiled and stirred until it is completely saponified, or turned into soap. Heating is continued until nearly all of the moisture is driven off. Part of the mineral oil is then stirred in, and the stirring is continued with the addition of more oil until the proper consistency is reached. Lime and soda greases are made by first mixing the fixed oil and mineral oil and then adding the alkali. The finished product should contain no lumps and should be homogeneous in character.

The **appearance of a grease** is frequently very misleading. To reduce the price manufacturers may resort to the use of the cheapest mineral oils obtainable, or to improper reduction that leaves excessive moisture in the finished grease, or may increase the weight, thus cheapening the product, by the addition of powdered soapstone, chalk, talc and other fillers. None of these faults can be determined by the appearance of the grease, yet none should be tolerated in a high-grade product.

Graphite is classed as a lubricant because it reduces friction, but its action is different from that of oils. If a highly polished bearing surface is examined under a high-power microscope it will be seen that the so-called smooth surfaces really consist of alternate ridges and depressions which are too small to be detected with the eye. If graphite is used on such a bearing it will soon fill all the depressions and give a bearing surface that is much smoother; this will reduce the friction. A steady supply, however, would soon clog the bearing.

Graphite and oil are often used together in small gasoline engines and will give good results where the graphite does not accumulate and clog the bearings. Oil holes and pipes are very liable to stoppage from this cause. For this purpose graphite, if used at all, should be in a **deflocculated** form, the **amorphous** form not being suitable.

Tests were conducted by *G. J. Meyers, U. S. N.*, at the *U. S. Navy Academy* to determine the lubricating value of 0.25 per cent graphite added to oil. This is stated to be the percentage for which the least friction is claimed. Runs made on the testing machine without and with graphite added to the oil gave the following results:

Load in lbs. per sq. in. of bearing surface	32.70	49.30	67.20	83.90	101.50
Critical speed, straight mineral oil	1.46	1.79	2.18	3.16	4.09
Critical speed, oil plus 0.25 per cent graphite	1.47	1.68	2.23	3.18	4.12

from which he concluded that "comparing the results, it is seen that the presence of graphite had no appreciable effect on the lubrication."—*Jour. A.S.N.E., Vol.* 29.

SELECTING LUBRICANT FOR SERVICE

In general, but more especially with high and medium speed machines, the best practical **guide as to the suitability** of a lubricating oil is the amount of frictional heat produced with that oil in use. Frictional heat indicates wasted power, and, in the case of a bearing, is determined by the temperature of the bearing in operation and the temperature of the room in which the machine is running. An indication of improvement in lubrication is the reduction of the amount of frictional heat. However, there are so many other factors which enter into the determination of the **ultimate value of a lubricant** that the amount of frictional heat as indicated by the bearing temperature is not in itself conclusive proof of the superiority of one lubricant over another.

For instance, in **Naval practice**, where ordinarily a low viscosity oil would reduce the frictional heat, it might have a tendency to cause a hot or burned-out bearing that would endanger a fighting unit, which loss might be avoided by using an oil of sufficiently high viscosity. Also in **Diesel engine lubrication**, due to the fact that unused cylinder oil drains out and mixes with oil in the general lubricating system, it is desirable to use one oil sufficiently heavy for use in the cylinders, though this oil might cause additional friction in the bearings.

Lubricating oils may be divided into two general classes: The first class, called **engine and machine oils**, including dynamo, turbine, spindle oils, etc., is intended for **external lubrication.** The second class, **steam cylinder and valve oils**, is used for **internal lubrication** of steam cylinders and valves. **Motor oils** and oils for internal combustion engines may be considered as a separate class, though they are not essentially different from some of the highest class turbine, engine and machine oils.

Engine and Machine Oils should meet the following **requirements**: (1) The oil should be of sufficient viscosity or body to keep the bearing surfaces apart at working temperatures; (2) it should possess such qualities as will reduce frictional losses to a minimum; (3) it should remain fluid at such low temperatures as will be met under ordinary service conditions; (4) it should meet service requirements as to durability, separation from water, etc.; (5) the flash point should be sufficiently above the working temperature to allow a reasonable margin of safety; (6) it should have no tendency to decompose or to form deposits which will gum up the machine and increase the friction; and (7) it should contain no impurities which will corrode or pit the bearing metal.

In selecting an oil for any particular purpose, the viscosity should be determined by the conditions under which the oil is to be used. The purpose of lubrication is to secure and maintain a film of oil sufficient to keep the bearing surfaces apart at **working temperatures.** If an oil used on a bearing is too light in viscosity at working temperature to sustain the load it will permit metal contact, with consequent bearing wear and loss of power. On the other hand, if an oil which has too great body at working temperature is used, there will be unnecessary fluid friction, and again there will be loss of power. The more viscous the lubricant, the greater the bearing pressure which can be sustained; the **viscosity of the lubricant should, therefore, be in proportion to the bearing pressure.** Heavy, slow moving machinery requires an oil of high viscosity; light, swiftly moving machinery requires just the opposite. Also, high temperature conditions require an oil of higher viscosity than ordinary conditions. Most of the resistance or lost power in running machinery is caused by the internal friction or the fluid friction of the lubricant used.

The **general law governing lubrication** is that within certain limits a reduction in the viscosity of a lubricant will bring about a reduction in the coefficient of friction, or, in other words, a reduction in frictional heat and in lost power.

In deciding then whether to use a high (heavy), intermediate or low (light) viscosity oil on a particular bearing there are three factors to consider: (1) Temperature; (2) bearing pressure; and (3) journal rubbing speed.

The temperature to consider is the normal working temperature of the bearing, and the viscosity is chosen accordingly. If two bearings are identical, one working at 100 deg. F. and the other at 150 deg. F., there should be an oil of certain viscosity at 100 deg. F. for the first bearing and one having the same viscosity at 150 deg. F. for the second, if these temperatures are to be maintained.

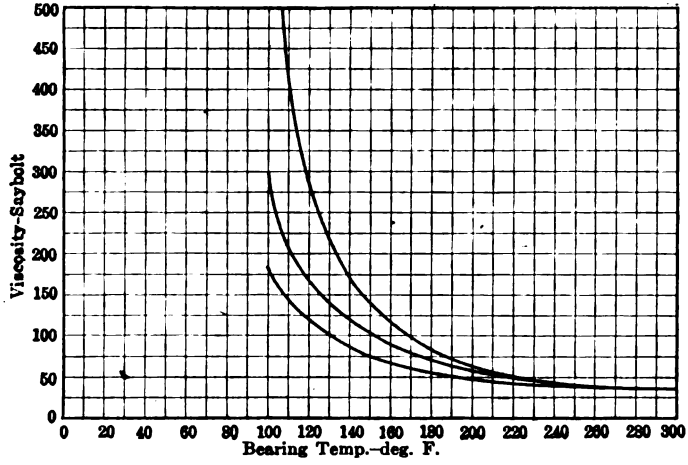

FIG. 4.

The question of the **proper heat** in a bearing is one that is open to much discussion, some authorities preferring to allow a bearing to run at high temperatures as long as it is in good condition and is getting plenty of good clean oil. The first effect of heating a bearing is to thin the oil and thus decrease the friction. The effect of cooling the bearing is to increase the friction. As a bearing heats up the viscosity of oil will change with the temperature and this variation is shown in Fig. 4 by curves which are drawn for typical oils.

The principal reason why a hot bearing is so feared is that it is generally an indication that something is wrong with the bearing or that it is not getting sufficient oil. Possibly this is a good reason for not allowing bearings to run at very high temperatures, but there is no excuse for going to extremes in the other direction. There is a turbo-generator in the power plant at the *U. S. N. Eng. Exp. Station at Annapolis, Md.*, the bearings of which run regularly at temperatures between 190 deg. F. and 210 deg. F. and this has never given any trouble, although the oils used in it have ranged from the heaviest forced-lubrication oil to the lightest ice-machine oil in use in the Naval Service. In one case a temperature of 205 deg. F. was maintained for 30 days.

Cooling a Bearing below a fair working heat by flooding it with oil results in a waste of oil, and keeping it cool by circulating an unnecessarily large amount of cooling water around it results in a waste of power. However, in case a bearing heats up due to dirt or other foreign matter, an increased supply of oil may be necessary to wash this dirt out of the bearing.

With regard to **bearing pressure**, the important condition is that an oil shall have body enough to prevent it from being squeezed out under the maximum load to which the bearing will be subjected. If an excessive amount of the chosen oil is required, this quantity can generally be reduced by substituting a heavier oil. Oil will thereby be saved but friction will be increased slightly. The oil selected will necessarily represent a compromise between the different factors.

In a **high-speed bearing** it is necessary to use a thin oil, on account of the great friction due to this high speed. Bearings of this type are generally designed so as not to have a very great bearing pressure, so that a light oil can be used without danger of it being squeezed out.

In **turbine bearings** the best conditions are obtained by the use of light oils. Here the speed is high and the bearing pressure is not only small, but is also steady and uniform in character. In a **reciprocating engine** the rubbing speed is much slower and the pressure on the bearings is intermittent in character. The reciprocating action intensifies the bearing pressure and the squeezing out effect but has the saving quality that while this pressure is on one side of the bearing the oil film is given an opportunity to form on the opposite side and be ready to meet this.

On account of the above action that takes place in a large reciprocating engine bearing a fairly heavy oil is needed to give an increased "cushioning" effect and to prevent wasting the oil. The limit of viscosity in forced-lubrication systems is set by the inability of the oil pumps to handle a too heavy oil. With engines not fitted with forced-feed lubricating systems it is set by the ability to feed properly.

Engine and Machine Oils are divided into two classes: (1) those suitable for use in **forced feed** systems and (2) those suitable for **gravity or wick feed** where forced lubrication is not used; in the Naval Service these oils are termed **"Marine Engine Oils."**

Forced Lubrication Oils must be straight mineral oils. One of the primary requisites of a forced feed oil is that it must not saponify and must separate readily from water. Every effort should be made to keep water out of such oil as the oil must be kept in continuous circulation and the effect of a water leak is cumulative.

Extreme care must be taken to eliminate **dirt.** Frequently oil will break down in a system due to the fact that it is constantly circulated at high speeds with water and dirt, the primary cause being due to the size and construction of the system and not to the oil itself.

Tables 1, 2, and 3 give the characteristics of acceptable oils for forced feed lubrication.

Marine Engine Oils. For gravity and wick feed lubrication, where forced feed closed systems are not used, the oil is used only once and it must be heavy enough to remain in the bearings and not be forced out easily, otherwise excessive waste will result. Also, since water often gets into the bearings or is splashed onto the crosshead, some provision must be made to prevent the oil from being washed away from the bearing surfaces. For the foregoing reasons, oils for this purpose are usually composed of **mineral oil compounded** with a certain percentage of animal or vegetable oil—ge-

Table 1. Heavy Force Feed Oils

Physical Constants

U. S. N. Eng. Exp. Station

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Pour test, deg. F.	Saybolt viscosity at		
						100 deg. F.	130 deg. F.	210 deg. F.
A	0.936	19.57	383	440	−3	750	234	58
P	0.908	24.18	414	467	35	570	251	66
A	0.930	20.50	380	430	0	512	204	55
A	0.886	28.01	457	512	30	454	215	63
A	0.936	19.57	389	440	0	433	200	51
P	0.877	29.60	416	440	32	417	209	63
*P	0.893	26.78	520	572	54	1630	672	130
*A	0.941	18.78	400	446	10	1643	520	80

* Extra heavy oil for motor cylinders only.

Table 2. Medium Force Feed Oils

Physical Constants

U. S. N. Eng. Exp. Station

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Pour test, deg. F.	Saybolt viscosity at		
						100 deg. F.	130 deg. F.	210 deg. F.
P	0.890	27.39	428	484	5	240	120	50
A	0.937	19.41	338	376	−7	322	137	49
P	0.904	24.87	418	449	10	255	122	49
A	0.929	20.70	379	434	−5	342	150	48
P	0.898	25.90	416	463	30	271	135	51
A	0.933	20.05	371	406	−5	275	121	47
P	0.887	27.84	442	502	20	270	130	51
A	0.930	20.54	340	376	0	308	144	52
P	0.875	30.00	470	532	5	280	140	53

Table 3. Light Force Feed Oils

Physical Constants

U. S. N. Eng. Exp. Station

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Pour test, deg. F.	Saybolt viscosity at		
						100 deg. F.	130 deg. F.	210 deg. F.
A	0.926	21.19	342	380	−5	218	107	48
P	0.876	29.82	425	470	−3	183	99	46
A	0.927	21.02	330	372	−10	202	115	44
P	0.883	28.55	420	468	25	202	104	47
P	0.876	29.82	408	460	23	214	114	43
P	0.878	29.54	428	478	5	199	108	49
P	0.874	30.18	428	464	10	182	100	46
P	0.881	28.97	422	472	20	205	107	48

Table 4. Marine Engine Oils

Physical Constants

U. S. N. Eng. Exp. Station)

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Pour test, deg. F.	Saybolt viscosity at		
						100 deg. F.	130 deg. F.	210 deg. F.
P	0.920	22.17	447	516	−15	540	242	65
A	0.940	18.94	365	412	5	679	293	72
P	0.907	24.36	434	484	15	511	236	69
A	0.940	18.94	360	406	5	653	267	65
P	0.918	22.51	412	447	20	585	274	79
*P	0.915	23.01	447	502	1	477	205	58
*P	0.883	28.55	434	477	30	445	213	64
*A	0.930	20.50	386	438	5	508	206	55

* Straight mineral oils not compounded with fixed oils

erally about 15 to 20 per cent of blown rapeseed oil. This is the only kind of engine oil used in the U. S. Navy that is not a pure mineral oil.

Animal and vegetable oils maintain their viscosity with increase of temperature much better than do mineral oils; also, when stirred with water, they form thick emulsions, or to use the common engineering term, they "saponify." The effect of this saponification is to make the oil more adhesive. On this account a mineral oil can be washed off a crosshead slide much more easily than can a vegetable or animal oil.

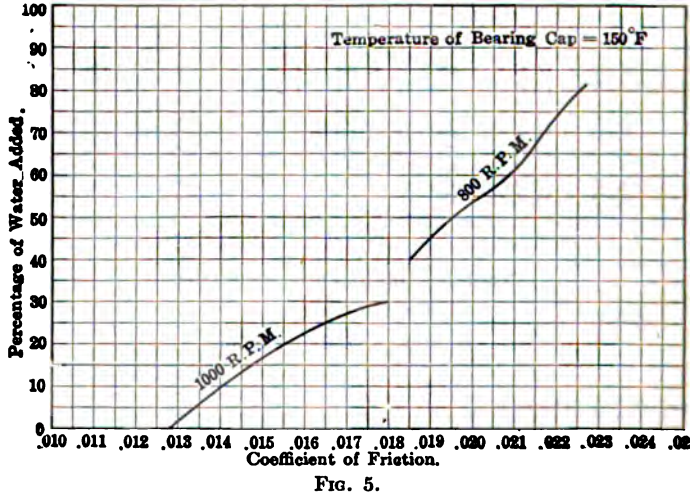

FIG. 5.

However, a mixture of oil and water will not give as good lubrication as oil alone. This is shown graphically in Fig. 5, taken from "*Practical Lubrication*," *Bryan, Jour. A.S.N.E., Vol*, 27, which curve was obtained by measuring the coefficient of friction of a marine engine oil first alone and then with various percentages of water added to it, the mixture being thoroughly

stirred to make it saponify. This curve shows that small amounts of water in a bearing will greatly increase the friction. The emulsion formed at the bearing will prevent the oil from washing off the bearing surface; however, the best plan is to use an oil that will emulsify and then keep water out of the bearings.

For **general lubricating oils** the only **requirement as to flash point** is that the flash should be sufficiently high to allow a reasonable margin for safety. As ordinary bearing temperatures do not average more than 150 deg. F., a flash test about 300 deg. F. would give a margin of more than 150 degrees.

Table 5. Steam Cylinder Oils

Physical Constants

U. S. N. Eng. Exp. Station

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Cold test, deg. F.	Saybolt viscosity at		
						130 deg. F.	210 deg. F.	300 deg. F.
P	0.904	24.87	490	542	45	760	129	54
P	0.908	24.19	550	614	42	834	154	59
P	0.894	26.60	557	610	55	947	230	61
P	0.902	25.00	575	630	40	1250	195	68
P	0.897	26.08	570	624	50	1022	183	63
P	0.921	22.00	524	588	53			

Table 6. Steam Superheat Cylinder Oils

Physical Constants

U. S. N. Eng. Exp. Station

Base	Specific gravity	Baumé	Flash point, deg. F., open cup	Fire point, deg. F., open cup	Cold test, deg. F.	Saybolt viscosity at		
						130 deg. F.	210 deg. F.	300 deg. F.
P	0.915	23.04	646	700	30	....	312	91
P	0.913	23.34	600	680	45	1887	295	81
P	0.897	26.08	609	670	35	....	236	72
P	0.902	25.21	597	650	50	1411	236	72
P	0.907	24.36	602	660	46	....	230	69

Cylinder Oils. Mechanical conditions involved in steam cylinder lubrication require special oils for that purpose. **Unfiltered** cylinder oils are usually dark green or black; **filtered** oils are light green and bright red. Steam cylinder oils are, as a rule, **very viscous**, some of them scarcely flowing at ordinary temperatures. The viscosity of these oils is taken at 210 deg. F. (ordinary oils are taken at 100 deg. F.).

The conditions to be met in steam cylinder lubrications are quite different from those occurring in external lubrication. In the former case the oil is atomised and used but once and is then carried off with the exhaust steam. The chief consideration in **selecting steam cylinder oil** should be (1) its viscosity, and (2) whether or not it should be compounded.

The **proper viscosity** for a steam cylinder oil is determined by (1) the speed and type of engine, (2) method of effecting lubrication, (3) the distance from the steam chest to the point of introduction of the lubricant to the steam line and (4) the amount of moisture in the steam. It is necessary that the oil

be atomized before it reaches the valves to afford good lubrication. As a low viscosity oil atomizes more readily than a high viscosity oil, and as time and distance are required to atomize the heavier oils, it follows that the greater the moisture in the steam and the shorter the steam line from the steam chest to the point of introduction of the lubricant the lower the permissible viscosity.

In ordinary cases, where there is a considerable amount of moisture in the steam, the cylinder oil should be **compounded** with a certain amount of animal oil so that the oil will more readily form an emulsion and adhere to the cylinder walls. On the other hand, a **straight mineral cylinder oil** should be used where it is desirable to keep the animal matter out of the exhaust steam, as for example when the steam is condensed and returned to the boilers and used for other purposes.

When superheated steam was first employed, it was thought necessary to have cylinder oils with a **flash point** higher than the total temperature of the steam. In some instances superheated steam reaches a temperature of 700 deg. F. It is possible to have a cylinder oil made from certain stocks with a flash point of 620 deg. F., but in practice with high superheat temperatures this oil would tend to coke and cause damage.

Air Compressor Oil, see p. 1205.

Table 7. Ice Machine Oils

Physical Constants

U. S. N. Eng. Exp. Station

Oil number	Base	Specific gravity	Baumé	Flash pt., O.C.	Fire pt., O.C.	Cold point	Viscosity		
							100°	130°	210°
1	A	0.929	20.70	295	334	−26	105	65	39
2	P	0.883	28.55	368	418	−20	105	67	40
3	P	0.900	25.56	371	406	−20	107	67	40
4	A	0.923	21.70	328	360	−15	110	66	39
5	P	0.872	30.55	379	415	−10	128	75	42
6	P	0.895	26.42	373	413	− 5	114	70	40
7	A	0.919	22.34	310	340	−15	97	68	38

Ice Machine Oil. Oil used in the expander valve chest and cylinder of dense air machines should have a very low cold point and should not have too great a viscosity at the low temperatures at which the expander cylinder works. In order to get these qualities it is necessary to use a light oil.

In the compressor cylinder and valve chest the temperature is higher. In some cases it is so much higher that ordinary ice-machine oil will not have sufficient viscosity. In this case a heavier oil should be used in the compressor cylinder and valve chest. It is a good plan to use regular ice-machine oil where possible, as there will, of course, be some leakage of the oil from the compressor side to the expander side and a heavy oil will very likely gum there.

Motor oils and oils for internal combustion engines are subjected to extraordinary conditions. Their **requirements** are therefore somewhat different from other lubricating oils, and may be classed as follows: (1) The **viscosity** should be such that the film of oil will be sufficient to keep the metal surfaces apart; (2) the oil should possess such characteristics as will permit it to form a **seal** between the piston rings and cylinder walls so as to secure a **high degree of compression**; (3) it should be of such a nature as

to be **converted into vapor without decomposition,** even under extreme heat, so that the oil which gets into the combustion chamber will pass off with the exhaust as vapor without leaving an excessive carbon deposit; (4) it should be of such a nature that whatever **carbon** is deposited is **soft,** and easily removed; (5) the **cold test** should be low enough to meet all temperature conditions.

The **proper viscosity** for a motor oil is determined by the piston ring clearance and the conditions of the rings and cylinder walls. A wide clearance or worn rings, either of which causes poor compression, requires a high viscosity oil in order to secure the best results. If the piston rings are closely fitted to the cylinders a light-bodied oil should be used. If **too light an oil** is used too much oil will get past the piston rings on the suction stroke, and on the compression stroke some of the fuel mixture will leak past the piston rings and condense in the crank case, reducing the viscosity of the oil. If **the oil is too viscous** there will be a loss of power. Good compression depends very largely upon proper lubrication.

Lieutenant G. S. Bryan, U. S. Navy, Jour. A.S.N.E., Vol. 27, discusses the **flash and fire points of motor cylinder oils** as follows: "It is important that the flash point shall be higher than the temperature of the inner surface of the cylinder . . . with the water boiling in the jackets the temperature of the inner surface of the cylinder walls will be about 267 deg. F. The temperature of the layer of oil that is in immediate contact with the cylinder walls, which is the part that regulates the friction, cannot be much higher than this. I do not know of any motor oils that have a flash point lower than 325 deg. F. If the cylinder walls get up to this temperature in a water-cooled motor there is something radically wrong, and the remedy is not to get another oil of higher flash point, but to locate the trouble and remove it.

It is an old theory that was never founded on solid facts that a high flash-point is a necessity in a motor oil or the oil will burn up without giving any lubrication. The point was overlooked that when we have a maximum temperature of the gases in the cylinder of 2700 deg. F. and an average temperature of 950 deg. F., an oil with a flash point of 450 deg. F. will offer but little more resistance to burning than would one of 350 deg. F.

Either oil will burn if kept for any length of time in contact with the hot gas . . . A thin film of oil smeared on a hot (300 degrees) piece of iron or steel will burn for several seconds if ignited. Few motors ever run at less than 120 revolutions per minute, and at this rate the average point of lubricated surface on the cylinder wall would be exposed to the action of the flame for only one quarter of a second . . . there is no danger of all the oil film being burned in that short time, though there is no doubt that some of it is burned, whether the flash point is 300 deg. F. or 500 deg. F. . . . A flash point of 300 deg. F. would be sufficiently high for almost any water-cooled motor . . . 300 deg. F. should be sufficiently high for Diesel engines except in unusual cases."

Experiments made by *The Texas Co.* show that the **flash point has nothing to do with the lubricating value of a motor oil.** An examination of the flash point of the oil in the crank case, after it had operated in the motor of a car that had run fifty miles, revealed the fact that the flash was only 150 deg. F., while the original flash point was 445 deg. F. The original high flash of the oil did not indicate the sealing qualities necessary for a good motor oil, and with this oil the seal between the piston rings and the cylinder walls was not such as to prevent the gasoline vapor from passing into the crank case during the compression stroke. This gasoline mixture contaminated the oil, made it lighter in body and reduced the flash to only 150 deg. F., thus proving that a high flash oil would not remain so unless it possessed other and more valuable characteristics, and that an oil having these essential qualifications could well be of a lower initial flash point if its flash would remain constant.

PHYSICAL CONSTANTS OF OILS

In order to make specifications for the purchase of lubricants and also as a guide for their suitability for certain purposes, a series of tests are often necessary. A list of those generally made are as follows: (1) Viscosity; (2) flash and fire; (3) emulsification; (4) water and sediment; (5) cold test; (6) gravity; (7) color and odor; and (8) carbon.

In determining the suitability of a lubricating oil for any particular purpose, the **most important single characteristic** to be considered is its viscosity. Color and gravity have nothing, and flash and fire practically nothing, to do with the lubricating value of an oil. While the cold test of an oil has an important bearing on its suitability for feeding uniformly under low temperatures, generally speaking it has nothing to do with the lubricating value, once the oil has reached the bearing surface.

The Viscosity of an oil is the resistance offered by its particles or molecules in sliding past one another; in other words, its **internal or fluid friction.** Viscosity denotes the body or relative fluidity of the oil, and is determined by the time it takes a certain amount of oil to flow through a standard orifice.

The relative viscosity at atmospheric temperature of two or more oils in sample bottles can be roughly determined by holding the samples together and quickly turning upside down, and then noting the drop from the bottoms of the inverted bottles. The oil which drops first has the lower viscosity, or the thinner body; if they drop together, the viscosity or body is about the same. Again, the samples may be shaken and the relative viscosity determined by watching the ascent of equal sized air bubbles, the more sluggish the movement, the greater the viscosity of the oil.

For the accurate determination of viscosity it is necessary to use an instrument called a **viscosimeter.** The viscosimeter in general use in this country is the **Saybolt Standard Universal,** page 258. The standard instrument in England is the *Redwood*, in France the *Barbey*, in Russia the *Lamansky-Nobel*, and in Germany the *Engler*. There are also a number of other instruments in existence, as well as a number of modifications of the instruments already mentioned. Other instruments which have been used to a limited extent in this country are the *Tagliabue*, the *Scott*, and the *Doolittle*. For **conversion curves** from Saybolt to Redwood and Engler, see page 260.

Paraffine Base Lubricating Oils, when finished, seldom have a viscosity of over 250 seconds Saybolt, hence when higher viscosities are necessary for certain purposes, they must be blended with some other product of higher viscosity; the residue from Pennsylvania crude is usually selected for this purpose, although various animal and vegetable oils are sometimes used. **Asphaltic base oil** can be finished at practically any viscosity without blending. In the case of oils from the same crudes, the oil heavier in body will generally be heavier in gravity, but between oils manufactured from different crudes there is no viscosity-gravity relation.

The Temperature at which the Viscosity Test is Made is a matter of the greatest importance as all oils lose viscosity with a rise in temperature. The temperatures used for the Saybolt Universal are 210 deg. F. for steam cylinder oils and 100 deg. F. for engine and machine oils, and, as a rule, 100 deg. F. for internal combustion lubricating oils. The temperature of the oil during the test must be kept uniform as a variation of one degree in temperature would make an appreciable difference in the viscosity. Moreover in referring to the viscosity of any oil the temperature at which the viscosity was taken must be stated

Flash and Fire. When an oil is heated sufficiently, it gives off vapors which are inflammable, and which form an explosive mixture with air. The danger of fire or explosion, especially in the case of kerosene or burning oils,

led to the introduction of a test which would differentiate those oils which were safe for illuminating purposes from those which were so volatile as to make them dangerous. Later, when the mineral lubricating oils were introduced, the same tests were extended to these heavier oils, not because there was any particular danger of fire in the case of a properly manufactured oil, but rather as a guide to the refiner in his various methods of distillation. The Cleveland Open Cup Tester and the Pensky Martin Closed Cup Tester are specified for Navy Contracts. Lubricating cuts from Pennsylvania crudes have higher flash points than similar cuts from asphaltic base crudes.

The **Flash Point** of an oil is the temperature at which an oil, upon being slowly heated, will give off sufficient vapor to form over its surface a mixture with the air, which mixture, upon the introduction of a small flame, will ignite and go out. As the temperature of the oil rises, more vapor is given off, and when the production of vapor is sufficiently rapid, the oil takes fire and burns with a continuous flame. The **fire point** of an oil may be defined as the temperature at which, upon the application of a flame, the oil will burn continuously. For **method of determining** flash and fire points see page 268.

Emulsification Test. The emulsifying properties of all oils used in the Naval Service, except steam cylinder oils, are determined by means of an emulsifying machine at the *Naval Eng. Exp. Station*. This machine consists of a small motor which turns a copper paddle, a revolution counter, a water bath, an electric heater for the water bath and a rheostat for controlling the motor speed.

Four emulsification tests are made on each oil. Forty cubic centimeters each of distilled water, salt water, normal caustic soda solution, and boiling distilled water is placed in separately, thoroughly cleaned 100 c.c. graduated cylinders; then to each cylinder is added 40 c.c. of the oil to be tested.

The cylinders containing the oil and solutions are placed in the water bath, whose temperature is maintained at 130 deg., and the mixture in each cylinder is stirred by the copper paddle for 5 minutes at 1500 R.P.M. The temperature of the water bath is kept at 130 deg. and the cylinders containing the mixed oil and solutions are kept in the water bath, and if complete separation of the oil and the solution takes place within one-half hour, the time required for complete separation, after stirring, is noted.

The percentage of actual emulsion remaining in each cylinder, after the lapse of ½, 1, 4 and 24 hours is noted.

Lubricating oils used for forced feed or ice machine lubrication must settle out completely from all solutions within 30 minutes.

Compounded marine engine oils for use on reciprocating marine engines should show practically complete emulsion at the lapse of ½ hour and not very much separation at the end of 24 hours.

Water and Sediment in lubricating oil is a matter that is often brought forcefully to the attention of those who handle petroleum products. Not all impurities in petroleum products, however, are included under this heading. Others may be due to improper refining, or to improper or careless handling of the finished products, but their detection usually requires the services of a laboratory.

Water in any petroleum product (1) may have been present when the oil was shipped by the manufacturer, (2) it may have entered during transit, or (3) it may have entered at the consumer's plant. Almost invariably refined petroleum products are free from water when they leave the refinery. Many complaints of water in lubricating oils have been investigated and in nearly every case the water was of purely local origin.

The source of **sediment** in oil is more difficult to locate. It may be white sediment due to zinc corrosion in the case of oil shipped in steel drums, or it may be sediment due to glue in the case of oil shipped in barrels. The barrel is lined with glue and bungs should not be driven in, and care should be exercised in boring holes in the barrels not to introduce wood chips. Waste may be introduced in various ways.

Water or sediment may enter the oil through water leaks from piston rods. Various boiler compounds may come over when the boilers prime and eventually find their way into the lubricating oil. Due to modern facilities, inspection and care, it is practically impossible for moisture in oil to go undetected at the filling plant. To prevent water from getting into the oil during transit and storage, barrels should be shipped and stored on the side or bilge; better still, they should never be allowed to stand in the open.

Cold Test. Nearly all oils are liquid at ordinary temperature. If the temperature is gradually reduced from 60 deg. F. the oil will contract until it finally ceases to flow. **Paraffine base oils,** owing to the cost and difficulty of removing the paraffine wax, usually do not have as low a cold test as the **asphaltic base oils,** which do not have to contend with wax. **Russian oils** are low cold test oils.

Low cold test oils are necessary for use in or on equipment subjected to extremes of temperature, such as ammonia cylinders of **ice machines.** Reference to table 7 shows that when properly treated for removal of wax, paraffine base oils show satisfactorily low cold tests. The cold test may be divided into two stages, (1) the cloud test and (2) the pour test. The **cloud test,** which is only carried out on paraffine base oils, is to determine the temperature at which the oil will become slightly cloudy or partially opaque due to the formation of particles of paraffine in the oil. The **pour test,** which is made on both paraffine and asphaltic base oils, is to determine the lowest temperature at which the oil will flow or pour.

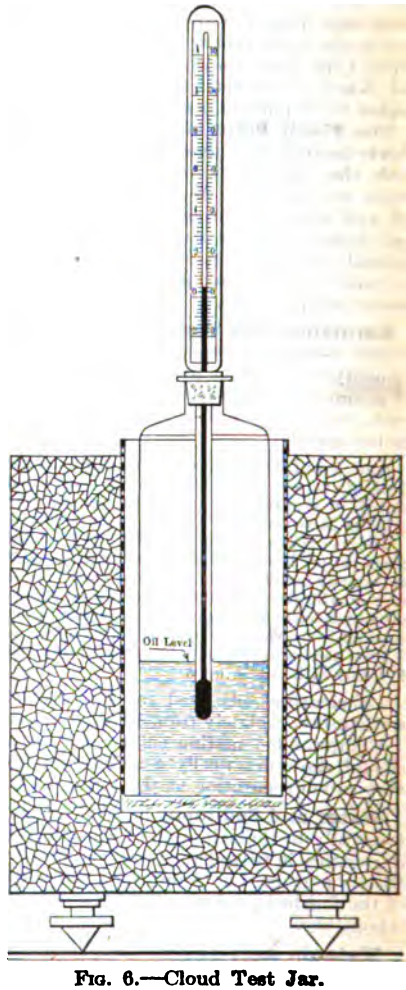

FIG. 6.—Cloud Test Jar.

The Cloud Test, Fig. 6. A cylinder or glass jar about 1¼ in. in internal diameter and about 5 in. long, or a regular 4-ounce sample bottle, is filled to a height of about 1¼ in. with the oil to be tested. A cold test thermometer is inserted into a cork placed in the mouth of the jar, so that the lower end of the thermometer is ½ in. from the bottom of the jar.

The cold test jar is then placed in a metal or glass jacket 4 to 5 in. high, care being taken to see that the test jar is placed in the centre of the jacket so that it does not touch the sides at any point. A disk of felt, cork or wax, ¼ in. thick, is placed in the bottom of the jacket. The whole apparatus is placed in a refrigerating mixture and at every drop in temperature of 2 deg. F., when near the expected cloud test, the jar is removed and inspected. When the lower half of the sample becomes opaque the thermometer reading is taken as the "cloud test."

The Pour Test may be a continuation of the cloud test. At each drop in temperature of 5 deg. F. the jar is removed and tilted until the oil begins to flow When the oil has become solid around the thermometer and will not flow when the jar is tilted to the horizontal, the previous 5 degree point is taken as the "pour test." Preferably, the cold should be so applied as to complete the cold test in approximately one-half hour.

FIG. 7.—Pyknometer.

Specific Gravity gives no indication of the lubricating value of an oil. To the expert it will give some indication of the nature of the crude from which it was obtained Light gravity crudes are entirely paraffine base crudes, and the products therefrom are of correspondingly light gravity. The Texas, except from the Electra field in the northern part, and the California crudes are heavy gravity, and the products therefrom are correspondingly heavy in gravity.

Specific gravity can be determined in various ways, one of which is by means of the pyknometer, or specific gravity bottle, Fig. 7. A special bottle with a perforated stopper is used in the laboratory, but with care an ordinary bottle can be used. The bottle should first be thoroughly cleaned and rinsed and then filled with cold distilled water and heated to 15.6 deg. C., or 60 deg. F. At this temperature the water should just fill the bottle. The full bottle should be carefully weighed, and after emptying and drying it, the empty bottle should be weighed. The bottle should then be filled with the oil to be tested, heated to 60 deg. F., again having the bottle exactly filled with oil and weighed. In each case the weight of the empty bottle must be deducted from the total weight found. The specific gravity of the oil is the ratio between the actual weight of the oil and the actual weight of the water.

Color, generally speaking, **is no guide to the suitability** of any class of lubricating oil for any particular work. An oil may be either pale or red, regardless of its grade. Oils of one refiner can, as a rule, be classed generally by colors, but, owing to the considerable difference in the crudes employed and to the manipulation of these crudes, the products of one refiner cannot be compared with those of another on the basis of color alone.

Prejudice has always existed among oil users in regard to color, especially among users of dark colored oil. Such engineers are prone to regard light colored oils with distrust because, whereas a thin film of dark colored oil can be seen clearly, light colored oils do not so readily indicate that the bearings are lubricated.

The **color of an oil changes** with continuous use. An **extra pale turbine oil** in continuous use will gradually become darker, and after many

months will be a deep wine color. The same change takes place with a **red turbine oil**, but it is not so noticeable. **Diesel engine oils** also darken, due largely to unconsumed parts of the fuel washing down to the crank case.

When an oil changes in color through use, it does not mean that the oil is becoming worn out or that its lubricating value is being materially affected. It is simply an indication that under the stress of service the oil is taking up various foreign particles which affect the color; the formation of asphaltenes and oxidation products is taking place. Changes in color of an oil, therefore, are due to surrounding conditions and not to the oil itself.

For ordinary purposes the color of an oil can be determined by its appearance in an ordinary sample bottle. Color as well as the general appearance of an oil will vary with the size of the bottle, and it is therefore essential that for comparative purposes the same sized bottle be used. The 4-ounce bottle is the standard used by all oil companies and is therefore the most convenient to use.

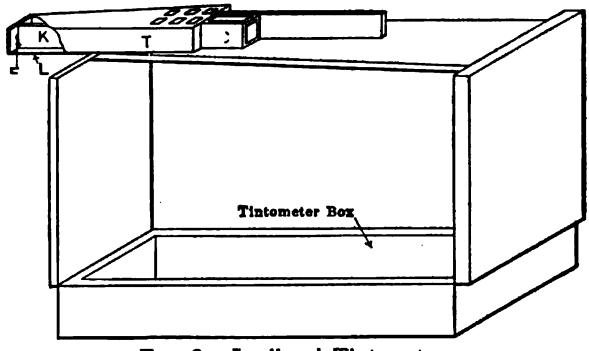

FIG. 8.—Lovibond Tintometer.

In the laboratory, where **color** must be **accurately determined**, special instruments are used, the most important being the Lovibond Tintometer, Fig. 8. This consists of a covered trough or box *T* divided longitudinally by a partition terminating in a vertical knife edge *K* opposite an eyepiece *E* fixed at one end. Two channels *L* are thus formed, which diverge slightly from the end at which the eyepiece is placed. The oil to be examined is placed in a rectangular glass cell *C* which is placed at the end of the right-hand channel. Stained glass slides of known depth of color are then inserted into the small slots at the other end of the channel until the color of the oil is exactly matched. These slides are all numbered on a scale known as the five hundred series, and the depth of color of the oil is expressed numerically. For dark oils a half-inch cell is used, for pale oils a 6-in. cell, and for kerosene and gasoline an 18-in. cell.

Odor is a matter of importance in all lubricating oils.

Lubricating oils are classified in regard to odor as normal and compounded. **Normal neutral oils** are straight mineral oils, and **compounded oils** are mineral oils combined with fixed oils. The different fixed oils, such as lard, rape and neats foot, have characteristic odors which can frequently be recognised in the compounded oil, so that it is usually possible to tell by the odor of an oil whether it is a straight mineral oil and often, in case it is a compounded oil, what fixed oils have been used in compounding.

All straight mineral oils have a characteristic odor, which, in well-refined oils, is almost imperceptible. Oils that have been shipped or stored in wooden barrels occasionally have a rancid odor, due to the presence of mositure which, even in small quantities, will dissolve the glue lining of the barrel.

Carbon. Petroleum or any petroleum product is a mixture of hydrocarbons which are chemical compounds, the carbon and hydrogen being combined chemically in different proportions to form entirely new and different substances. To say that an oil contains carbon means as much as to say that a piece of chalk contains carbon or that water contains hydrogen gas. To be sure carbon is one of the chemical constituents, but it has absolutely lost its identity, and before the carbon can be reclaimed the compound must be decomposed or broken up into its constituent elements.

When any substance containing carbon is subjected to high temperature without a free supply of air, the substance may be decomposed. If so, the gaseous elements are liberated or combine with other elements to form new compounds, and a residue is left which is chiefly carbon. In most instances, however, this would not be pure carbon, but would consist of various other substances as well. Since this decomposition takes place to a greater or less degree when lubricating oils are subjected to certain conditions the question of the decomposition of hydrocarbons and the deposit of carbon as a residue is an important one in the lubricating problem.

All deposits from a lubricating oil are not due to carbonization. The **deposits** which are sometimes found **in bearings** or oiling systems are due to ordinary wear, and consist of small pieces of metal due to metallic wear or iron rust combined with dust and deposits from the surrounding air, and do not contain carbon from the oil unless the bearing has burned out. In the case of **steam cylinder** lubrication the **deposits** are due rather to boiler water impurities and to small amounts of metal, mostly from rust, which are bound together by the oil. In course of time this oil partially evaporates and decomposes, leaving a hard deposit which contains only a small amount of oil and a relatively small percentage of carbon. An analysis of an average deposit from a steam cylinder might show that the residue contained 2 per cent of unchanged oil, 1 per cent of decomposed hydrocarbons, 7 per cent of carbon, and 90 per cent of ash, consisting of metallic substances, rust, dust, boiler impurities and dirt.

While the **deposits** found **in internal combustion engines** are by no means all carbon, still the percentage of carbon in each deposit is large, and this question is of the greatest importance. Carbon deposit in the cylinder and on the top of the piston and on the valves cause a great amount of inconvenience and damage, and therefore special stress is laid on the amount and nature of the carbon deposited in internal combustion work.

A great proportion of all the trouble with internal combustion engines is caused by hard carbon deposits, which wear away the cylinder, score the rings, make the valves leaky, destroy the spark plugs, and act generally against the efficiency of the engine. Furthermore, in cleaning out this hard carbon, the surfaces are frequently scratched and damaged.

SECTION 14

MEASURING HORSEPOWER OF MARINE ENGINES

BY

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CONTENTS

	Page
Power	1403
Indicators and Indicator Cards	1404
Essential Requirements of Good Indicator	1405
Indicator Springs	1405
Indicator Cord	1406
Reducing Motions	1406
Taking the Card	1408
Indicator Cards	1409
Combined Indicator Cards	1410
Effect of Varying Cut-off on Power Distribution	1411
Steam Consumption from Indicator Card	1412
Calculating the I. H. P.:	
M. E. P.	1412
Planimeters	1413
Manograph	1416
Torsion Meters:	
Shaft Calibration	1417
Gary-Cummings Torsion Meter	1420
Hopkinson Thring Torsion Meter	1423
Denny-Edgecombe Torsion Meter	1425

MEASURING HORSEPOWER OF MARINE ENGINES

BY

H. R. GARY

REFERENCES: Dyson, "Practical Marine Engineering," Aldrich, Pub. Co. Holmes, "Experimental Engineering," U. S. Naval Institute. Proceedings American Society of Naval Engineers.

Power is the number of units of work performed in a unit of time. **The horsepower,** the unit of power employed by engineers, is 33,000 ft.-lbs. per min. (=550 ft.-lbs. per sec.). In marine work horsepower may be expressed as (1) theoretical, (2) effective, (3) indicated, (4) shaft or (5) nominal.

Theoretical horsepower (H.P.) lies wholly in the province of the designing engineer and for power measurements may be neglected.

Effective horsepower (E.H.P.) is the actual tow-rope power required to move the clean hull with appendages through the water.

Indicated horsepower (I.H.P.) is the power developed by the steam in the cylinders. It does not include losses arising from resistance in the machinery; used for expressing horsepower of reciprocating engines.

Shaft horsepower (S.H.P.), sometimes called **brake horsepower (B.H.P.),** is the power actually transmitted at the shaft and is measured by the amount of torque or twist therein.

Nominal horsepower, a term now obsolete, was used in *Watt's* time interchangeably with I.H.P.

Power measurements are essential, especially aboard ship where fuel-carrying capacity is limited, to economical operation and in determining the efficiency of and detecting errors in design and erection.

Steam Engine Indicator Use. The use of the indicator to determine I.H.P. is confined to reciprocating, internal combustion and Diesel engines where it is possible to measure the pressure on the piston head, a special type of indicator, page 1415, being used for internal combustion engines where R.P.M. is very high. The indicator records the varying steam pressures in a cylinder during a complete revolution of the shaft, or double stroke of the piston, and from these the effective pressure at any point of the stroke may be ascertained and the **mean effective pressure** throughout the stroke be calculated. The I.H.P. of an engine may be calculated from this mean effective pressure, the piston area, the length of stroke and the R.P.M.

The indicator also may be used to **check valve settings**; the action of the steam distributing valve, whether admission is early or late; point of cut-off; exhaust opening and closure; compression; the nature of the expansion curve; the drop of pressure between the boiler and the cylinder; whether full pressure is maintained during admission or if steam is wire-drawn; the adjustment and condition of slide valve and piston; leakage in the valve after closing; the amount of back pressure or vacuum, and, in connection with a **feed water test,** the actual amount of steam consumed and the economy with which the engine operates.

Torsion Meter Use. Since it is manifestly impossible to measure the pressure on the turbine blades, as the pressure on each of the thousands of

blades would have to be measured separately, just as the pressure on each piston head has to be measured in a reciprocating engine, the torsion meter is used to measure the **power being transmitted through the shaft,** expressing it in S.H.P. Torsion meters will not give reliable information for reciprocating engines because of the variable torque in such shafting.

Torsion meters are designed to measure the shaft horsepower only, that is, the power that is actually transmitted through the shaft, differing in this respect from indicated horsepower which takes cognisance of the power generated by a known pressure of steam acting on a piston area and length of stroke, and part of which is dissipated in friction between this point and the point of application on the shaft.

S.H.P., as obtained by torsion meter, **can be reduced to I.H.P.** by dividing the former by a factor representing the engine efficiency. The factor 0.92 is used in ordinary calculations.

INDICATORS AND INDICATOR CARDS

A steam engine indicator, Fig. 1, consists essentially of a piston *A* of carefully determined area fitted into a cylinder *B*, the fit being sufficiently

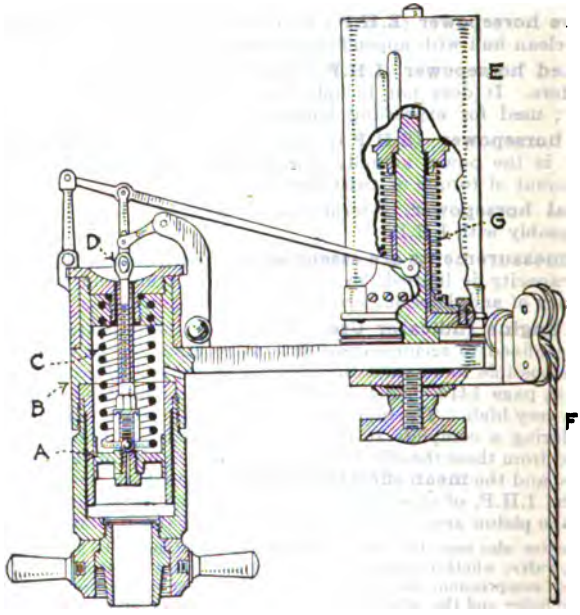

FIG. 1.—Inside Spring Indicator.

close to permit vertical movement of the piston without sensible friction. The lower end of the cylinder is open and fitted so that it may be attached to a cylinder of a steam engine in such a manner that the under side of the piston is simultaneously subjected to the same steam pressures as is the head of the engine piston. The upward movement of the indicator piston, due to the

steam pressure, is resisted by a special spring *C* of known resilience. A piston rod *D* projects upward through the cylinder cap and actuates a simple lever carrying at its free end a pencil point, whose vertical movement bears a constant ratio to that of the indicator piston.

A drum *E* of cylindrical form carries the **"card"** or paper on which the record is made and is attached to the cylinder *B* in such manner that the pencil point may be brought in contact with its surface and thus record any relative movement of paper and pencil; the drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord *F* attached to the cross-head and in the opposite direction by a spring *G* within itself.

When the indicator is properly adjusted and free communication opened with the cylinder of a steam engine in motion, the pencil moving vertically as the steam pressure varies, and the card moving horizontally under the pencil as the drum is rotated by the reciprocating motion of the engine, an enclosed diagram will be drawn on the paper representing the pressure of steam in the cylinder at all piston positions; the upper line shows the pressure urging the engine piston forward, and the lower the pressure retarding its movement on the return stroke.

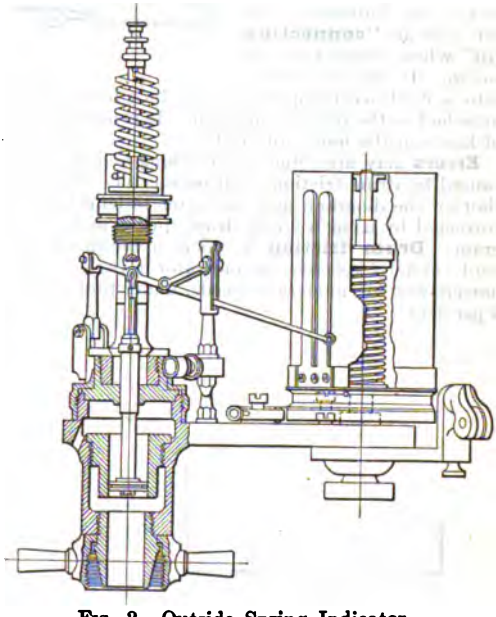

FIG. 2—Outside Spring Indicator.

Outside spring indicators, Fig. 2, originally introduced for internal combustion engine use, are now being generally adopted for steam engine use, particularly where high steam temperatures exist. In this type of indicator the temperature will not affect the elasticity of the spring and springs can be readily changed without removing other parts which are usually hot.

The essential requirements for a good indicator are (1) an accurately calibrated and elastic spring; (2) a light piston; (3) a small and light pencil mechanism and drum, and (4) a free and practically frictionless movement of the pencil on the paper.

Indicator springs. The **scale** or **number** of an indicator spring indicates the pressure in pounds per square inch that will produce a vertical movement of 1 in. at the pencil point. To find the proper spring to use, a safe rule in marine practice is to divide the boiler pressure, in pounds, by

the desired height of the diagram, expressed in inches. **Calibrations** of Indicator Springs that are good enough for rough checking in ordinary service may be made by comparing the vertical movement of the pencil with an accurate "test" steam gauge at about five "steps" of pressure above and three "steps" of pressure below the atmospheric line. For really accurate work, as on a **guarantee test**, comparison must be made with the indications of a mercury column or some special calibrating device.

Indicator cord should be of the very best quality. Steel wire may be used, but it has the disadvantage of weight and stiffness. **Wire**, however, should be used if the length is over 3 ft.

A hook, Fig. 3, should be attached to the free end of the indicator cord for ease in "**connecting up**" when a card is to be taken. It can be hooked into a cord with loop about 6 in. long in its end which can be permanently attached to the reducing motion. It is not practical to connect up by means of knots in the indicator cord.

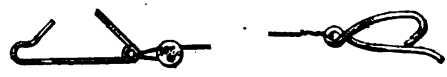

FIG. 3.

Errors may arise due to stretching of indicator cord; this stretch may be caused by drum friction or drum inertia. The former cause will operate to shorten the diagram, and the latter to lengthen it. **Drum inertia** may be corrected by using a stiffer drum spring, and, for high speeds, a shorter diagram. **Drum friction** is made a minimum by using proper quality of cord, and by keeping the indicator in perfect order. The error in indicated horsepower for moderate speeds, under proper conditions, should not exceed 3 per cent

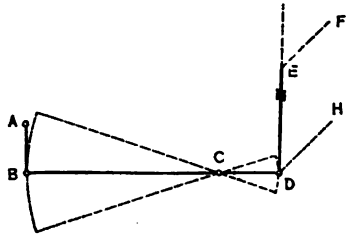

FIG. 4.

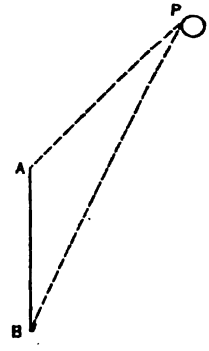

FIG. 5.

Reducing Motions. To obtain the conventional indicator diagram the drum of the indicator must be moved so as to give an exact proportional reproduction of the motion of the piston. A device which accomplishes this is called a perfect reducing motion. The most common form for marine indicators is composed of links, levers and bell cranks, Fig. 4. *A* is a pin attached to the crosshead; *AB* is a short link connecting the crosshead to a lever *BD* pivoted at *C*. The point *D* will then move in step and nearly in

constant proportion to the piston, and from *D* the motion for the drum may be taken, either by a cord direct, or from the end *E* of a rod *DE* moving as shown. In such cases the cord should run in continuation of the line *DE* and not off at an angle as *EF* or *DH*.

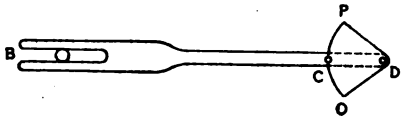

FIG. 6.

The reducing motion should be so adjusted that the **cord part should not undergo changes of angular direction**, or, at least, such changes should be made as small as possible. Thus in Fig. 5 suppose the point from which the motion is taken to move through a path *AB*, and the indicator guide pulley to be at *P*. Then at one extreme the cord will be represented by *PA*, and at the other by *PB*. Such a change in the angular direction of the cord relative to the line of motion *AB* will result in error, and should be avoided by bringing *P* in line with *A* and *B*.

Instead of taking motion from the crosshead by means of a short link as *AB*, Fig. 4, a lever *BD*, Fig. 6, may be employed, having a forked end and pivoted at *C* or *D*. A pin on the crosshead working in the forked end operates the lever, while the reduced motion is obtained from *D* or *C*.

Instead of attaching the cord direct to *C*, a sector of wood *POD* with centre at *D* may be attached to the arm and the string led off from the face of the sector. Such a sector may also be employed with the arrangement shown in Fig. 4. None of these motions are geometrically exact.

A form of **pantograph** consisting of jointed rods, as shown in Fig. 7, may be used where there is sufficient room. *A* is attached to the crosshead and *D* or *E* is the fixed pivot The other point *E* or *D* will provide a reduced motion for the indicator drum which is geometrically exact. Here again the string should be so led that its angularity will not vary. This gives geometrically exact reduction.

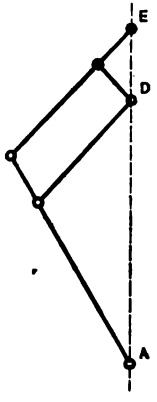

FIG. 7.

Reducing Wheels, which also give geometrically exact reduction, are coming into common use, Fig. 8. The large wheel receives the crosshead cord and the small wheel receives the drum cord. Ratio of diameters gives the reduction. Each reducing wheel is provided with a series of rings to alter the small wheel diameter so that various reductions are possible to suit the engine stroke. The indicator drum is started and stopped by turning the knurled nut on top of the small wheel. The large wheel has a spring in its centre to bring it back on the inward stroke.

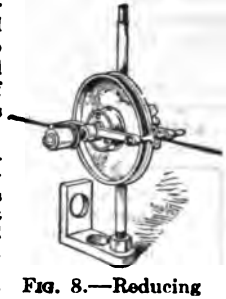

FIG. 8.—Reducing Wheel.

Testing Accuracy of Reducing Mechanism. Whatever type of reducing mechanism is used, its accuracy may be usually tested in the following manner: Lay off points on the engine guides at ¼, ½ and ¾ of a stroke. Connect the indicator as for taking diagrams. Put crosshead on dead centre, touch pencil to paper and make a vertical mark. In the same way make vertical marks when the crosshead reaches each successive point on the guides. If marks are exactly at fourths, longitudinally, on the card, the motion of the crosshead has been accurately reduced.

Preparation for Taking Indicator Card. The indicator should be adjusted and put in proper condition. See that all nuts and screws work,

that the piston works freely, that the necessary springs will fit, that the drum motion works well and that there is no lost motion in the pencil gear. The indicator should be attached as closely as possible to the cylinder; if, however, it is necessary to use pipes, they should be not less than five-eighths of an inch in diameter and as short and direct as possible.

After the indicator is connected it should be adjusted for working conditions. Try cocks and valves and especially the three-way cock. Open the valves at the end of the cylinder and blow steam through them to free them of dirt and water. Close the end valve and the three-way cock and pour some cylinder oil, not lubricating oil, into the elbow of the three-way cock, working the oil in with the handle.

Adjust the indicator cord so that it will have a vertical lead and see that the length is such that diagram will be in middle of card and not overlap the ends.

Put on the proper spring, the strength, or "number" of which may be calculated according to the formula already given or from the following table: For high pressures: H.P.—150; I.P.—80; L.P.—20; for low pressures: H.P.—150; I.P.—40; L.P.—10.

Oil the cylinder of the indicator. See that all nuts and screws are tight. Put a card in place and turn steam into the indicator (the drum not in motion) and adjust the card to get the proper height of diagram; the lowest point (or exhaust) should be about ½ in. above the bottom of the card. Take several sample cards to see that the indicator is properly adjusted and in good working order.

Take a diagram from each end of the cylinder, marking these "Sample Bottom" and "Sample Top." These cards serve merely for reference for those taken during actual test or trial.

Taking the Card. When the preparatory signal is given about two minutes before time to take cards, go over the indicator carefully. Put the card in place and turn on steam to warm up the indicator. Connect up the cord, be sure that valves and cocks are wide open and that the indicator cock is open to one end of the cylinder. On the executive signal, hold the pencil close to the card and allow the pencil to vibrate several times before taking the diagram. Then push the pencil motion up against the stop and allow it to pass over the diagram twice, which will give a good clear diagram. If the lines are double it is generally an indication that the card is defective.

Reverse three-way cock and take diagram from the other end of the cylinder. (To obtain the complete data for a full revolution of the shaft in double acting engines, two diagrams, one taken from either end of the stroke, are necessary.) Close the three-way and indicator cocks and take the atmospheric line. (In order to more correctly interpret the pressure lines, the line showing atmospheric pressure is drawn in its relative position.) Cards on all cylinders should be taken simultaneously.

Check over the instrument, re-oil if necessary, using cylinder oil, point and adjust pencil, place new card in clip. The indicator is now ready for the next card. When springs are changed, take sample cards and adjust diagram for height.

Additional Data Required. The following data should be filled in on the card and will be found convenient in making comparisons:

	Date. . . . .
Diagram from S S—	Engine. . . . .
Diam. of (H.P.) Cylinder.	
Length of Stroke.	Temperature.
R.P.M	Atmospheric.
Pressure of Steam in Boiler.	Injection.
Position of Throttle Valve.	Air Pump Discharge.
Vacuum in Inches.	Feed.
Temperature of Hot Well.	
Scale of Spring.	

The R.P.M., Steam Pressure, and Scale of Spring must always be noted.

Care of Indicator. After use it is usually most convenient to clean the indicator in place. Remove and clean the springs. Remove and clean piston, oil thoroughly, clean and oil cylinder and put piston in place. Wind cord around the drum and secure the end. Remove indicator and place in box.

To change a right hand to a left hand instrument and to adjust drum spring tension, consult manufacturers' directions.

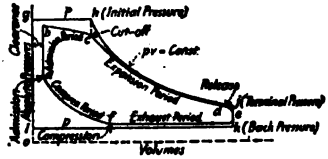

FIG. 9.—Typical Steam-Engine Indicator Diagram.

Among the reliable indicators on the market are the Crosby, the Thompson, the Tabor, the Manograph (used almost exclusively for the measurement of I.H.P. of internal combustion engines, and other engines operating at speeds exceeding 600 R.P.M.) and Hopkinson's.

A typical indicator diagram, *abcdef*, is shown in Fig. 9. Nomenclature as to valve action is shown at the points *a*, *c*, *d* and *f*. Nomenclature as to behavior of steam is designated on the intervening lines. The **idealized diagram** on which design is based is *ghjkl*, which differs from the actual diagram in showing no clearance or compression, in extending between the throttle pressure *P* and the atmospheric or condenser pressure *p*, in showing sharp corners at *h*, *j* and *k*, and in following the law pv = constant between *h* and *j*. Note the nomenclature adopted for pressures at *h*, *j* and *k*.

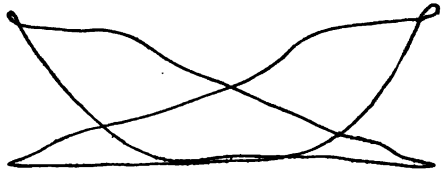

FIG. 10.

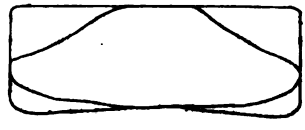

FIG. 11.

Deductions from the Card. In addition to giving the necessary data for calculating the I.H.P. of the engine, the cards may be expected to give information as to faulty design, adjustments, etc., some of which are shown in the following reproductions of cards:

Excessive angle of advance is shown in Fig. 10; all functions are "too early," cut-off, exhaust opening and closure. Steam lead and compression are excessive.

Insufficient angle of advance is shown in Fig. 11; all functions are "too late," which is a more serious defect than that shown in Fig. 10; steam lead is small or even negative; compression small.

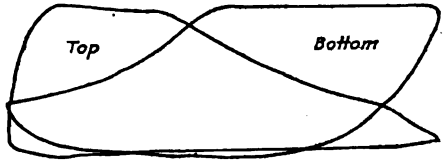

FIG. 12.

Valve stem too long is shown in Fig. 12; the top card shows at the top end of the stroke late and small steam opening, early cut-off, exhaust opens

early and full and closes late; the bottom card shows that exactly opposite conditions prevail. If the valve stem is too short, the defects are precisely the same, except they occur at opposite ends.

FIG. 13.

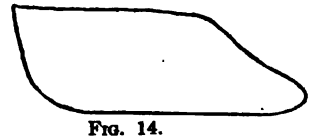

FIG. 14.

Excessive steam lap would give a card as shown in Fig. 13, indicating lead and steam opening late, early cut-off, port opening small and steam wire-drawn.

FIG. 15.

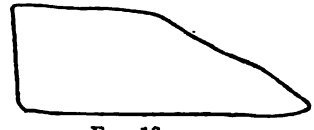

FIG. 16.

Insufficient steam lap is shown in Fig. 14; lead is large, opening is early, cut-off late and full port opening.

Excessive exhaust lap is shown in Fig. 15; compression is large, exhaust lead small, exhaust closes early and opens late.

Insufficient exhaust lap is shown in Fig. 16; compression is small, exhaust closes late and opens early.

Combined Indicator Cards. Cards taken from the engine cylinders are combined in such manner as to show the continuous expansion of the steam as it passes through the engine. To effect this all cards are laid down on a diagram to the same scale and pressure, Fig. 18. To combine the cards shown in Fig. 17 proceed as follows:

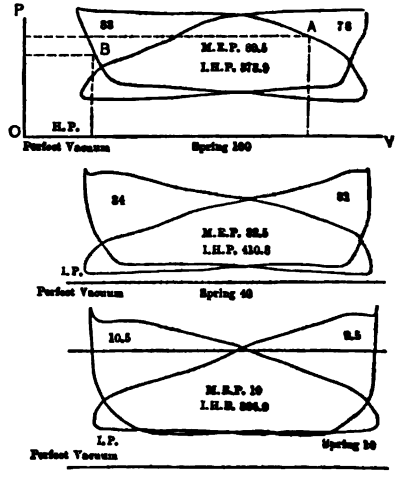

FIG. 17.—Readings From Port Engine of "Nevadan."

(1) Determine the clearance volume and volume swept by the piston for each cylinder, and on the *PV* diagram, Fig. 18, lay off lines *A*, *B* and *C* to represent to scale the clearance volumes of the H.P., I.P., and L.P. cylinders, respectively, and *E*, *F* and *G* to represent the total volumes of corresponding cylinders. The scale of volumes should be so chosen that the L.P. volume plus clearance will come within the desired limits on the combined diagram.

(2) To the new scale and with the clearance line of each cylinder as the zero of volume for the card of that cylinder plot cards on the PV diagram. The result is seen in Fig. 18.

(3) Extend the point of cut-off to the admission pressure line at K, and draw the hyperbola, PV = constant, through K. The hyperbolic expansion line represents the ideal expansion line.

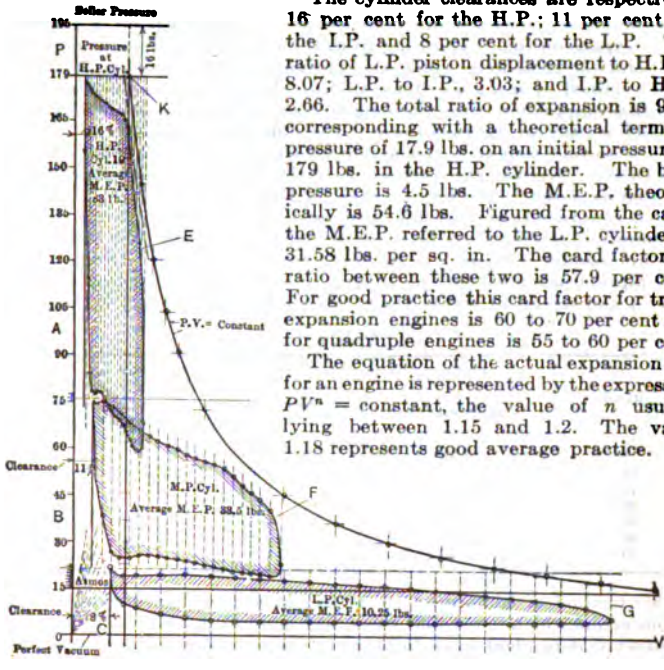

FIG. 18.—Combined Cards From Engines of the "Nevadan"—Mean of the Two Engines.

The cylinder clearances are respectively 16 per cent for the H.P.; 11 per cent for the I.P. and 8 per cent for the L.P. The ratio of L.P. piston displacement to H.P. is 8.07; L.P. to I.P., 3.03; and I.P. to H.P., 2.66. The total ratio of expansion is 9.96, corresponding with a theoretical terminal pressure of 17.9 lbs. on an initial pressure of 179 lbs. in the H.P. cylinder. The back pressure is 4.5 lbs. The M.E.P. theoretically is 54.6 lbs. Figured from the cards the M.E.P. referred to the L.P. cylinder is 31.58 lbs. per sq. in. The card factor, or ratio between these two is 57.9 per cent. For good practice this card factor for triple expansion engines is 60 to 70 per cent and for quadruple engines is 55 to 60 per cent.

The equation of the actual expansion line for an engine is represented by the expression PV^n = constant, the value of n usually lying between 1.15 and 1.2. The value 1.18 represents good average practice.

Effect of Varying Cut-off on Power Distribution. Referring to Fig. 18, cutting off earlier in the L.P. cylinder will result in raising the upper line of the L.P. and lower line of the I.P. cards, and thus in increasing the area of the former and decreasing that of the latter. In like manner cutting off earlier in the intermediate cylinder will result in raising the upper line of the I.P. and the lower line of the H.P. cards, and thus in increasing the area of the former and decreasing that of the latter. In like manner cutting off later in any cylinder beyond the H.P. will result in similar changes, but in the opposite direction. Thus a later cut-off in the I.P. cylinder will decrease the power developed in that cylinder, and increase the power developed in the H.P. cylinder. It thus results that a combination of changes such as a later cut-off in the I.P. cylinder and earlier cut-off in the L.P. will both tend to decrease

the power developed in the intermediate cylinder; while an earlier cut-off in the intermediate cylinder and a later cut-off in the low cylinder will both tend toward an increase of power developed in the intermediate cylinder.

Steam Consumption from Indicator Diagram. Where no other means are available for determining steam consumption this may be calculated from the indicator card as follows:

Referring to the H.P. card, Fig. 17, draw lines *OP* and *OV*, being lines of zero volume and zero abs. press. respectively. Select a point *A* near the end of the expansion line. Knowing from the card the pressure and volume at this point the total weight of *steam expanding in the cylinder* each stroke can be determined by using the steam tables. In a similar manner from a point *B* on the compression curve the weight of steam being *compressed in the cylinder* each stroke can be calculated. The difference between the first weight and the second weight is the weight of steam *passing through the cylinder* each stroke. Knowing the revolutions of the engine the amount of steam used per hour *as shown by the diagram* may be determined.

No account is taken in the above of condensation in the cylinder; the steam consumption as shown by the diagram must be corrected for this condensation by the use of Table 1 to get the actual water rate. Divide the steam consumption as shown by the diagram by the proper factor taken from Table 1 and the result will be the actual steam consumption of the engine.

Table 1.

Cut-off	Factor: Divide steam consumption per card by factor		
	Simple engines	Compound engines, H.P. cylinder	Triple expansion engines, H.P. cylinder
5	0.58		
10	0.66	0.74	
15	0.71	0.76	0.78
20	0.74	0.78	0.80
30	0.78	0.82	0.84
40	0.82	0.85	0.87
50	0.86	0.88	0.90

This method of computing steam consumption of an engine is liable to serious errors which in extreme cases may amount to as much as 50 per cent. Accuracy is dependent upon tightness of steam piston and packings. Similar engines may be compared by this method when operating under similar conditions but not when run under radically different conditions.

CALCULATING THE I.H.P.

The general formula for I.H.P. is $\text{I.H.P.} = \frac{P \times L \times A \times 2N}{33{,}000}$, where P = mean effective pressure in lbs. per sq. in., L = length of stroke in feet, A = area of piston in sq. in., and N = R.P.M.

The mean effective pressure (M.E.P.) is obtained from the indicator card by the planimeter or by the following method, see Fig. 19: Draw lines perpendicular to the atmospheric line *AB* and tangent to the card ends. Divide *AB* into 10 equal parts by ordinates perpendicular to *AB*. Midway between these ordinates draw dotted lines *O1*, *O2*, *O3*, etc., parallel to them. The mean heights of the areas formed by the ordinates are indicated by the enclosed lengths of the dotted lines midway between the ordinates.

The mean effective pressure of the whole diagram is found by measuring the dotted lines enclosed by the diagram with the scale corresponding to the spring used in taking the diagram. The sum of these mean pressures divided by 10, the number of divisions, will be the mean effective pressure sought in lbs. per sq. in. If necessary, the diagram may be divided into 20 equal divisions to insure a correct measurement of the pressures.

If the spring scale is not at hand mark off the height of the divisions on a strip of paper, one after the other continuously; the distance from the first to the last mark on the strip will represent the sum of all the measurements, which can be measured in inches with an ordinary rule. This quantity divided by the number of divisions in the diagram or diagrams and multiplied by the scale of the spring used will give the mean effective pressure. If there is a loop in the diagram subtract the area of the loop from that of the other part as it represents loss of power.

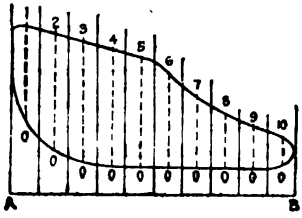

FIG. 19.

The quickest and most accurate method of measuring a diagram and finding the M.E.P. is by use of the **planimeter.** The planimeter gives the exact area of a diagram in square inches and decimal parts of an inch and avoids the possibility of error. To calculate the M.E.P. by this method divide the area by the length of the diagram, expressed in inches and decimals of an inch and multiply this value by the scale corresponding to the spring used; the result is the M.E.P.

The Planimeter is made in several different types all of which serve the same purpose, the measurement of irregular areas. Among the various planimeters are the Amsler, Conradi, Bristol-Durand and the Coffin, the last being the one in most common use on shipboard.

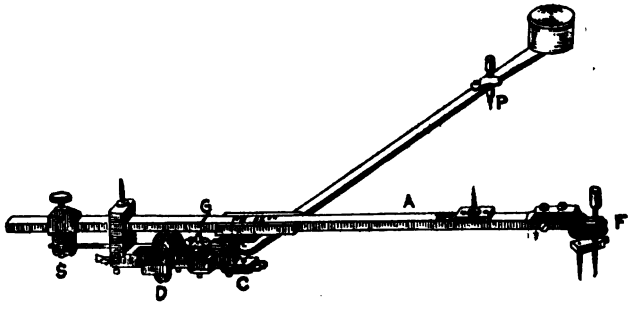

FIG. 20.—Amsler Polar Planimeter.

The Amsler Polar Planimeter, Fig. 20, consists of a radius rod *CP* fitted with a weighted needle point *P* which acts as the center about which the instrument turns. The opposite end of the radius rod is pivoted to a saddle which slides along the square rod *A* which in turn carries the tracing point *F*. This saddle also carries the recording wheel *D* with its spindle and worm which operates the recording disk *G*. The saddle is clamped to the tracing rod and can be accurately placed in position on the rod by the adjusting

screw *S*. The several marks on the tracing bar indicate the units in which the area may be recorded by the instrument, with the multiplier to be used in each case.

To Find Area. Place the saddle on tracing rod and bring it to mark corresponding to unit in which it is desired to measure area. The fixed center *P* is then placed somewhere *outside* the area to be traced. Bring tracing point *F* to a position on outline of area and prick starting point; read and record disk and vernier *D* and wheel *G*. Trace outline of area clockwise and when starting point is reached take readings and subtract therefrom the original readings. Multiply difference by constant on tracing bar and result will be area in desired units.

Zero Circle. In case an area to be measured is so large that it cannot be traced with fixed point *P* placed outside, this point may be placed inside the figure. In this case the result must be corrected by adding the value of the zero circle, which value is marked on the tracing bar for each saddle position.

The Coffin Planimeter, Fig. 21, may be considered a special form of the Amsler Planimeter in which the radius rod is of infinite length, the point *C* (Fig. 20) being constrained to move in a straight line. This instrument can be used to measure the area of any figure that can be placed in position on the board, but the size and shape of the board practically limits its use to the **measurement of indicator cards,** for which it is primarily designed.

This instrument consists essentially of a rod having on one end a tracing point *O*, and on the other end a pin which slides in a groove in the plate *I*, the weight *Q* keeping the pin in the groove. This rod also carries the graduated recording wheel *W* which is provided with a vernier. The rod is thus supported on three points, namely, the tracing point *O*, the pin at *Q* and the flange of wheel *W*. Attached to the board are the stationary clip *C* and the sliding clip *K*.

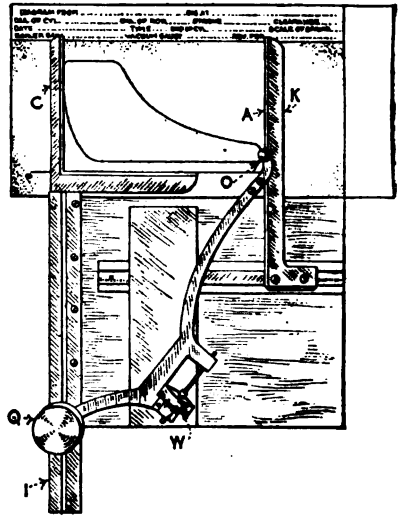

FIG. 21.—Coffin Planimeter.

To obtain M.E.P. slide indicator card under clips *C* and *K* so that atmospheric line is parallel to clip *C*, and so that left hand edge of diagram touches vertical limb of *C*. Slide clip *K* until it just touches right hand edge of diagram. Place planimeter with tracing point at *O* where clip *K* touches diagram. Puncture diagram at this point and set recording wheel at zero. Starting at *O* trace outline of diagram in clockwise direction until *O* is again reached; at this point the wheel reading equals card area in sq. in. Slide tracing point along *K* until wheel reads zero (say at *A*). *OA* is mean height of card and when scaled with the indicator spring scale the result is M.E.P. of card.

Care must be taken that card is horizontal and that clips *C* and *K* just touch diagram edges. The instrument mechanically divides the area traced by the distance between clips so it is essential that this distance be the accurate card length. When measuring areas the instrument reads in sq. in. to $\frac{1}{50}$ sq. in.

Calculating I.H.P. Having obtained in the foregoing manner the value of M.E.P., and knowing the values of L, A and N, the I.H.P. of the cylinder is found by the following formula:

$$\text{Cyl. I.H.P.} = \frac{\text{P(head end)} \times \text{L} \times \text{A} \times \text{N}}{33{,}000} + \frac{\text{P(crank end)} \times \text{L} \times (\text{A} - \text{r}) \times \text{N}}{33{,}000}$$

where r = the area of the piston rod in sq. in.

If M.E.P. and R.P.M. are each equal to unity in the preceding expressions, the result is the I.H.P. per pound M.E.P. at one R.P.M. These factors are called the **cylinder constants**, the cylinder constants for the two ends being respectively $\frac{\text{A} \times \text{L}}{33{,}000}$ and $\frac{(\text{A} - \text{r}) \times \text{L}}{33{,}000}$. The cylinder constants are among the data furnished by the shipbuilder.

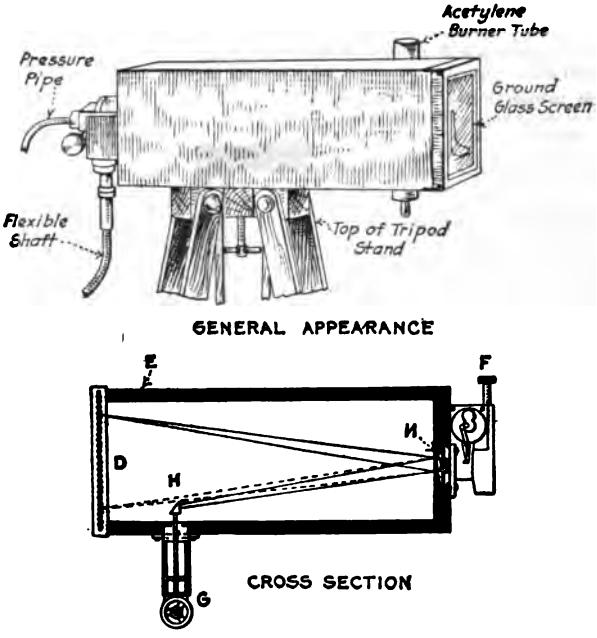

FIG. 22.—Hospitalier-Carpentier Manograph.

If the valves of an engine are well adjusted and the M.E.P. for the two ends of a cylinder are approximately the same, an approximate calculation of the I.H.P. of the cylinder may be made by multiplying the average M.E.P. of the two ends by the R.P.M. and the sum of the cylinder constants.

The I.H.P. of a multiple expansion engine is the sum of the I.H.P. of all cylinders.

The Manograph for indicating internal combustion engines and high-speed steam engines overcomes the inherent difficulties of the piston type indicator for this work; the piston type is unsuitable because of the inertia of its moving parts at high speeds and the tendency of springs to break when used on internal combustion engines.

The Hospitalier-Carpentier Manograph, Fig. 22, consists of a light-tight box mounted on a tripod. Within the box a mirror *N* is so mounted that it is capable of rotation about two axes at right angles to each other. An acetylene burner *G* on one side of the box, shining through a diaphragm, reflects a beam of light through the prism *H* to the mirror *N*. The beam is again reflected from the mirror *N* onto the screen or plate *D*.

The mirror *N* is given two motions, one in proportion to the piston motion and the other at right angles to the first in proportion to the pressure on the piston at any simultaneous piston position. As the mirror is moved the beam of light will follow a path which is compounded from two rectangular coördinates, one proportional to the piston position and the other proportional to the simultaneous piston pressure. In other words the beam will trace a card on the screen similar to a card obtained by an ordinary indicator. To obtain a permanent record, a photographic plate may be substituted for the ground glass screen.

Motion proportional to engine pressure is given the oscillating mirror as follows: The mirror is mounted on springs, Fig. 23, which tend to keep it parallel to the screen. The tube *T* communicates with the engine cylinder, permitting cylinder pressure to act on the diaphragm *M*. The diaphragm is connected with the mirror *N* by pin *A* offset from the mirror centre. The mirror is rotated by this pin an amount proportional to the pressure on the diaphragm, which is the cylinder pressure. Tube *T* can communicate with the various cylinders of a multicylinder engine by means of a multiway cock.

Motion proportional to piston travel is given the mirror *N* as follows: The flexible shaft, Fig. 22, which rotates with the engine shaft, connects the crank shaft centre to *L*, Fig. 23. By means of the gear *L* and a pin *B*, which is 90 degrees on the mirror from pin *A*, motion proportional to the piston travel is imparted to the mirror.

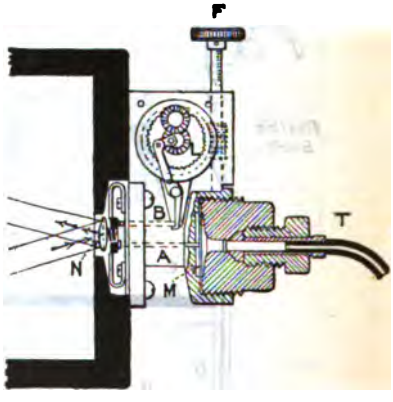

FIG. 23.—Mirror Mechanism, Hospitalier-Carpentier Manograph.

The angular motion of the mirror is very small. Thumb screw *F* is used to establish synchronism between the engine crank and the small manograph crank.

TORSION METERS

When power is transmitted through an elastic shaft, within its elastic limit, the shaft is twisted through an angle, the size of which depends upon the size of shaft and the power transmitted. In any given shaft the amount of twist and the applied moment of rotation are in a fixed proportion. The shaft is calibrated to obtain this fixed proportion and the torsion meter furnishes the means of measuring the amount of distortion of the shaft while

running. From these and from the revolutions of the shaft the power can be calculated.

Torsion meters may be of (1) the **mechanical type**, best represented by the Gary-Cummings instrument, (2) the **mirror type** as in the Hopkinson-Thring, or (3) the **electro-mechanical type** such as the Denny-Johnson instrument.

Most torsion meters are attached to the outside of the shaft by means of suitable sleeves or clamps. The Gary-Cummings meter differs fro n the others in that it is placed inside the shaft, only the card holder and the portions carrying the recording needles being outside the shaft. This permits of the torsion meter proper being left in place at all times which assures that readings in service will agree with the data obtained when the shaft is calibrated. It cannot be used on solid shaft installations unless one section is bored to receive it.

Shaft Calibration. To find S.H.P. from torsion meter readings the torsion meter constant for the shaft involved must be known. This constant varies with each shaft due to fluctuations in the inherent strength of shafting material. It is usually among the engineering data furnished by the shipbuilder and is determined before shafting is installed in the following manner:

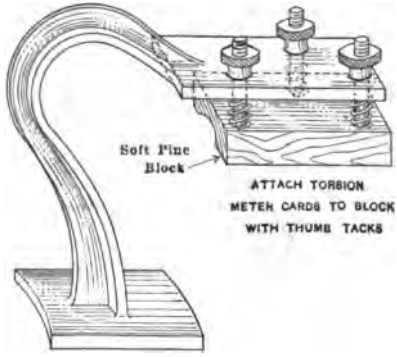

FIG. 24.

The section of shaft to which the torsion meter is to be attached is placed on two suitable supports, one end being securely fastened to a heavy fixed girder. The opposite end of the shaft is supported by a bearing which allows it to turn in an angular direction with as little friction as possible. To this free end of the shaft is attached another girder for applying the test loads. Care should be taken that the turning forces are balanced so that there is no eccentric twist in the shaft.

The usual method is to apply known weights at one end of the beam for a downward pull, and a turn-buckle or jack in connection with a weighing scale, at the other end, to supply an equal upward thrust. Suitable corrections must be made for the weight pans, turn-buckle or jack, and attachments. The points at which the turn-buckle and scale pan act upon the beam should be knife edges carefully measured from and on the same line with the axis of the shaft. For convenience these distances are usually made equal.

Fig. 24 shows a convenient card holding device for obtaining meter readings when calibrating shafting with Gary-Cummings Torsion meters.

A ring should be secured around shaft to guide the card holder as shown in Fig. 25, which gives the arrangement of calibrating apparatus for Gary-Cummings Torsion meters. About six cards should be attached to the block which should be adjusted by means of the thumb nuts so that both meter recording points mark a fine line on card when the holder is moved around shaft in direction of rotation.

In making the calibration test a zero reading should be taken. A known moment is applied to the shaft which is then rapped sharply with a lead hammer to remove any "set" that may have been caused by binding in its bearing, and a record taken. Weights are then added in increments and the turn-buckle adjusted each time for an equal up-

ward thrust. The shaft should be thoroughly rapped each time with lead hammers before readings caused by the torsional deflections are taken. These operations are continued until desired load has been reached.

To check this, similar records are taken as the load is removed step by step. From these results, with proper corrections for zero readings of the instrument, the calibration curve for the meter is easily plotted. Torsion meter readings are plotted as abscissæ and the torque, which is the equivalent load applied at 1-ft. radius, as ordinates. If the test has been properly conducted the resulting curve will be a straight line, indicating that the angular twist in the shaft is directly proportional to the load applied.

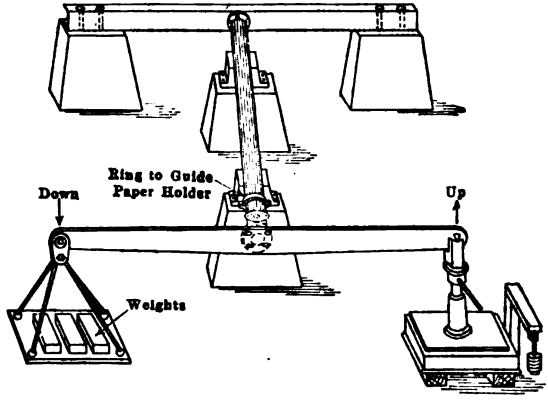

FIG. 25.

The torque per 1 in. of meter reading is calculated from the above. This will be found practically the same for all the loads applied to the shaft. Theoretically it should be exactly the same. The average of these gives a figure on which to base the horsepower constant for that meter in that particular shaft.

Algebraically, the calculations are as follows:

$$S.H.P. = \frac{2 \times \pi \times W \times R \times R.P.M.}{33{,}000} = \frac{2 \times \pi \times M \times R.P.M.}{33{,}000} =$$

$$\frac{2 \times \pi \times Q \times T \times R.P.M.}{33{,}000} \quad \text{Assume that} \quad K = \frac{2\pi \times Q}{33{,}000} \text{ then, } \mathbf{S.H.P. = K \times T \times R.P.M.,}$$

Where R = radius on which load is applied in feet, W = weight of load in pounds, $R.P.M.$ = revolutions per minute, M = torque or moment in foot-pounds, Q = foot-pounds for unit torsion meter reading, T = torsion meter reading in inches, and K = torsion meter constant.

The complete calibration data sheet of the U. S. S. New Mexico is shown in Fig. 26.

Unless the torsion meter readings are not directly proportional to twist of shaft, which is the case with some special torsion meters not generally used, it is not necessary to apply the full load but this is recommended in order to reduce the error in calibrating. By taking seven or eight points up and down the error is averaged. If the exact torque

U.S.S. NEW MEXICO

SHAFT CALIBRATION DATA.

NUMBER	DOWNWARD PULL	UPWARD PULL	TOTAL PULL	TOTAL TORQUE	SHAFT NO. 1 METER READING	SHAFT NO. 1 CORRECTED METER READING	SHAFT NO. 1 TORQUE PER INCH METER READING	SHAFT NO. 2 METER READING	SHAFT NO. 2 CORRECTED METER READING	SHAFT NO. 2 TORQUE PER INCH METER READING	SHAFT NO. 3 METER READING	SHAFT NO. 3 CORRECTED METER READING	SHAFT NO. 3 TORQUE PER INCH METER READING	SHAFT NO. 4 METER READING	SHAFT NO. 4 CORRECTED METER READING	SHAFT NO. 4 TORQUE PER INCH METER READING
1	0	0	0	0	.265	0	-0	.065	0	0	.150	0	0	.080	0	0
2	2500	2490	4990	49900	.785	.520	95920	.600	.535	93270	.630	.540	91840	.590	.540	92410
3	5000	4983	9983	99830	1.30	1.065	95530	1.145	1.080	92440	1.235	1.085	90910	1.120	1.085	93900
4	7500	7435	14967	149670	1.850	1.585	94430	1.680	1.615	92670	1.780	1.630	91820	1.660	1.630	93540
5	5000	4983	9983	99830	1.30	1.065	95530	1.145	1.080	92440	1.240	1.090	91590	1.120	1.085	93900
6	2500	2490	4990	49900	.795	.525	95080	.610	.545	91900	.710	.560	89110	.590	.540	92410
7	0	0	0	0	.270	0	0	.067	0	0	.150	0	0	.050	0	0
AVERAGE							95300			92540			91330			92990

SHAFT CALIBRATED TO MAXIMUM CAPACITY OF CALIBRATING APPARATUS

S.H.P. = SHAFT HORSE POWER

R.P.M. = REVOLUTIONS PER MINUTE.

M = MOMENT IN FOOT LBS

Q = TORQUE PER INCH TORSION METER READING

T = TORSION METER READING IN INCHES

K = TORSION METER CONSTANT

$$S.H.P. = \frac{2\pi \times M \times R.P.M.}{33000} = \frac{2\pi \times Q}{33000} \times T \times R.P.M. = K \times T \times R.P.M.$$

$$K = \frac{6.2832 \times Q}{33000}$$

NO. 1 SHAFT (STARBOARD OUTBOARD)

$$K = \frac{6.2832 \times 95300}{33000} = 18.15 \qquad K = 18.15 \quad \text{NO 1 SHAFT}$$

NO 2 SHAFT (STARBOARD INBOARD)

$$K = \frac{6.2832 \times 92540}{33000} = .1762 \qquad K = 17.62 \quad \text{NO 2 SHAFT}$$

NO 3 SHAFT (PORT - INBOARD)

$$K = \frac{6.2832 \times 91330}{33000} = 17.40 \qquad K = 17.40 \quad \text{NO 3 SHAFT}$$

NO 4 SHAFT (PORT OUTBOARD)

$$K = \frac{6.2832 \times 92990}{33000} = 17.71 \qquad K = 17.71 \quad \text{NO 4 SHAFT}$$

CALIBRATED UNDER SUPERVISION OF
CUMMINGS SHIP INSTRUMENT WORKS
AT NAVY YARD NEW YORK - FEB. 1917

APPROVED H. M. Cary 1917

FIG. 26.

corresponding to any given torsion meter reading were accurately known, the constant could be readily calculated.

The theoretical constant may be calculated, if for any reason a shaft cannot be calibrated, by the formula $K = 0.3196 \times \frac{D^4 - d^4}{l \times t'}$ where K = torsion meter constant, t' = units torsion meter reading per degree twist, l = length in inches of the calibrated portion of the shaft, D = outside diam. of shaft in inches, d = diam. of bore of shaft in inches.

It is advisable to measure the shaft if it is available instead of relying on drawings for dimensions. In determining shaft length for use in the above formula, flanges, fillets, etc., must be reduced to shaft diameter and considered.

To calculate the theoretical constant for Gary Cummings Torsion Meters use

$$K = 0.3196 \times \frac{D^4 - d^4}{l_1 \times t'}$$

Where D and d have the same meaning as above, t = torsion meter reading per degree of twist as furnished by the manufacturers and l_1 is obtained from the formula

$$l_1 = l_2 + 1.60r + \left(l_3 \times \frac{D^4}{D_1{}^4}\right), \text{ see Fig. 27.}$$

This formula for the value of l_1 is rendered necessary by the fact that the Gary Cummings Torsion Meter has its fixed end in a flange of the shaft, Fig. 27. If the radii, r, are not all equal use an average value. Take all dimensions, Fig. 27, in inches.

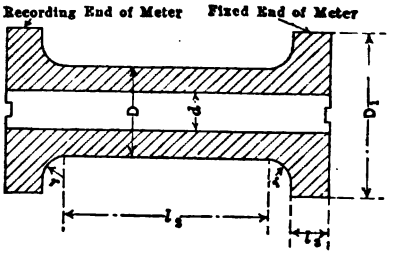

FIG. 27.

Gary-Cummings Torsion Meter. The three characteristics of the Gary-Cummings Torsion Meter are (1) extreme **accuracy,** (2) unusual **simplicity** of construction and (3) **lightness** per unit length of shaft. The instrument is largely housed within the shaft and contains no **gears** whatever. A contributory feature to its accuracy is the fact that it measures the torsion in one complete section of the shaft, making necessary but one multiplying device. This gives considerable advantage over meters which attempt to measure the angular twist in a short section and which require numerous multiplying devices, all of which are possible sources of error.

The instrument, Fig. 28, is primarily a rigid steel tube AB, extending for one complete shaft section, fastened rigidly at one end but free to move at the other. The **fixed end** of this tube is firmly attached to an arm C which rests in a radial slot in the flange of the shaft where it is securely fastened. Along the tube are spider collars D which fit loosely but are held from moving along the tube by long cotter pins placed through the tube on either side of the collar. These spiders eliminate all possibility of **whipping action.** The **free end** of the steel tube is supported and centered in a ball bearing so that it is free to rotate with respect to the shaft, and will therefore transmit any angular displacement between the tube and the shaft to an arm which is

secured across the end and extends radially through another slot in the shaft flange in which it is free to move angularly. That is, any angular displacement between the two flanges on the shaft can be noted by the movement of this arm with respect to the flange.

An **umbrella** *F*, equipped with two recording points, *H*, is fastened securely to the outside of the shaft flange at the free end of the tube in such a manner that it can rotate in a plane at right angles to the shaft. Any movement between the arm and the flange is transmitted through a sliding ball joint to the umbrella, causing it to turn an amount which is directly proportionate to the twist in the shaft. As the shaft revolves two points on the periphery of the umbrella record on an especially prepared card placed in the holder *G* at the side of the shaft, these points being adjusted so as to cause a small negative reading when no power is being transmitted.

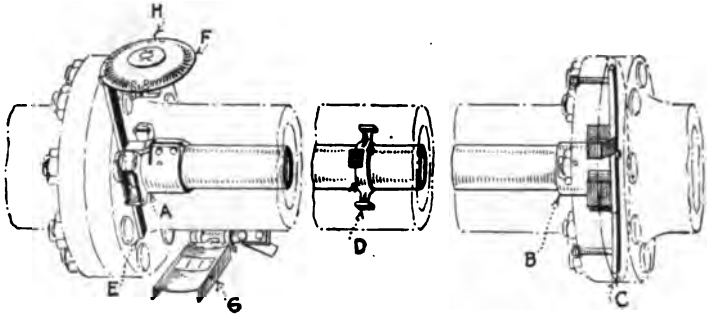

FIG. 28.—Gary-Cummings Torsion Meter.

When the shaft is transmitting power the linear distance in inches between the two marks recorded on the card, minus the zero reading, is directly proportional to the twist in the shaft. From this measurement the actual torque transmitted by the shaft can be readily obtained from the calibration data. **The S.H.P.** is calculated from the corrected meter reading, the meter constant and the R.P.M.

The meter tube itself is not subject, or cannot possibly be subjected to any twisting action, for it is the shaft that twists about the meter tube and it is this twist or difference in displacement between the shaft and the tube that is measured.

Method of Recording Twist of Shaft. A V-shaped or butterfly arm extends from the radial piece to the sliding ball point which moves the umbrella. This butterfly piece is pivoted to the radial arm in order to provide for the very slight but positive movement of this arm in a plane vertical to the shaft axis. The ball joint is fastened to the supporting axis of the umbrella so that any angular movement of the radial arm attached to the free end of the meter tube causes a proportionate revolution of the umbrella.

The recording points *H* are set on opposite sides of the outer edge of the umbrella. When the shaft rotates these recording points move in a plane perpendicular to the shaft and as the shaft is twisted the plane of one recording point moves aft and the other forward in proportion to the amount of twist in the shaft. The needles record on a card the exact distance between these planes, the holder *G* of the card being supported directly from the hull or bearing foundation, Fig. 29.

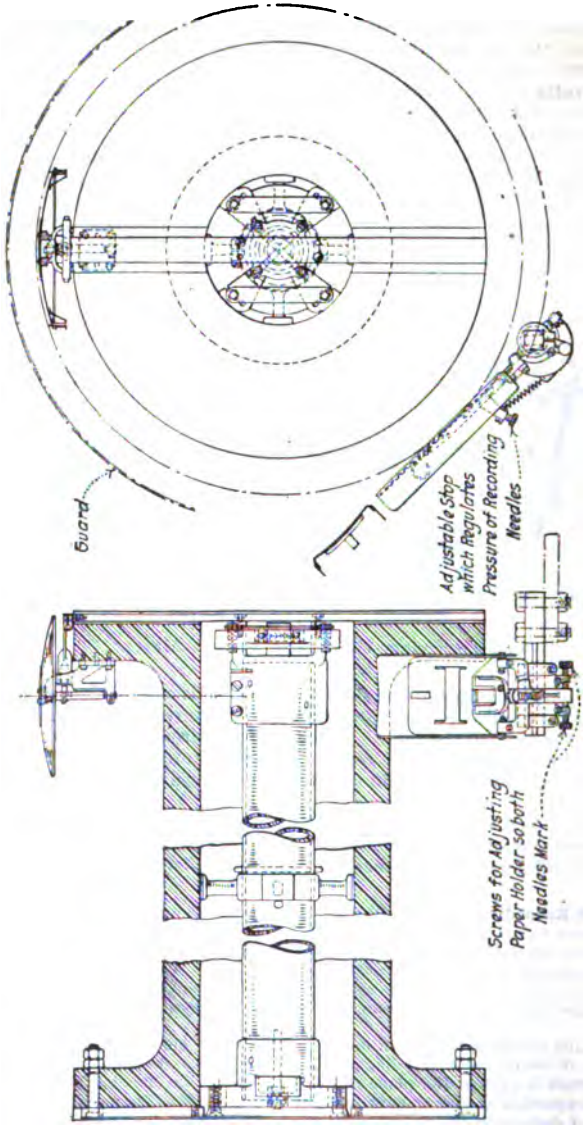
Guard
Adjustable Stop which Regulates Pressure of Recording Needles
Screws for Adjusting Paper-Holder so both Needles Mark

FIG. 29.—Section and End View of Gary-Cummings Torsion Meter.

Taking Readings. Tighten all screws and inspect torsion meter. To get torsion meter zero reading drag the shaft as follows: With about one-third speed on ship shut off all steam to turbine, including auxiliary exhaust, permitting shaft to revolve partly by weigh of the ship and partly by its own and the rotor inertia; take cards at frequent intervals as the shaft slows; the **minimum reading thus obtained is the zero reading.**

Place about a dozen cards in the holder, adjusting the stop so that the recording needles on the umbrella just make a mark without tearing the card. The fore-finger of one hand placed horizontally back of the card will bulge it forward sufficiently to make the needles record while the other hand moves the holder forward. This method avoids all possibility of the needles striking metal and becoming broken and does not require so fine an adjustment of the stop screw. Should one needle mark before the other, adjust this by the side thumb screws, Fig. 29, rather than attempt to remedy it by pushing the card with the fingers

At the same time that the card is taken, the R.P.M. should be noted. Mark on each card the shaft from which it is taken, the date, the time and the R.P.M.

Calculating S.H.P. The linear distance in inches between the two parallel marks on the cards, less the zero reading (in rare cases this may be plus because the planes of the needles cross as the shaft twists) is the correct torsion meter reading.

$S.H.P. = K \times T \times R.P.M.$, where T = corrected torsion meter readings, $R.P.M.$ = rev. per. min. and K = meter constant = $\frac{2\pi Q}{33{,}000}$, where Q = ft.-lb. for 1 in. of torsion meter reading as found from calibration data.

A typical Gary-Cummings torsion meter card is shown in Fig. 30, the vertical marks being the records made by the recording needles as the shaft revolved. In this case they are 2.12 in. linear measurement apart, which is the meter reading in this instance.

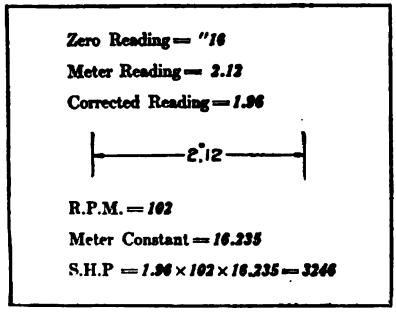

FIG. 30.—Gary-Cummings Indicator Card.

Hopkinson-Thring Torsion Meter. The principle of this apparatus is a differential one and consists in the observation of the twist between two points on the shaft by means of two beams of light projected onto a scale from fixed and movable mirrors. The beam projected on the scale by the fixed mirror is taken as a zero point while the beam projected by the movable mirror indicates the amount of torque on the shaft. Both mirrors revolve with the shaft but even at moderate speeds the reflections appear as continuous lines of light across the scale.

The apparatus consists of a torsion meter proper and a scale box, the diagrammatic arrangements of which are shown in elevation and plan in Fig. 31. The torsion meter consists of a collar clamped to the shaft, which collar has a flange projecting at right angles to the shaft with an extension around the shaft. A sleeve provided with a similar flange and extension at one end is clamped at its further end to the shaft, so that its flange is close to that on the collar, while its extension overlaps that of the collar on which it is supported to keep it concentric. The collar and sleeve are rigid so that

when the shaft is twisted by the transmission of power, the flange on the sleeve will move relatively to that on the collar, the movement being equal to that between the two parts of the shaft on which these fittings are clamped. The degree of movement is made apparent by one or more systems of torque mirrors, mounted between the two flanges, which reflect a beam of light projected from a lamp onto a graduated ground glass scale.

Each system of torque mirrors consists of a mounting pivoted top and bottom to one or other of the flanges and carrying two mirrors placed back

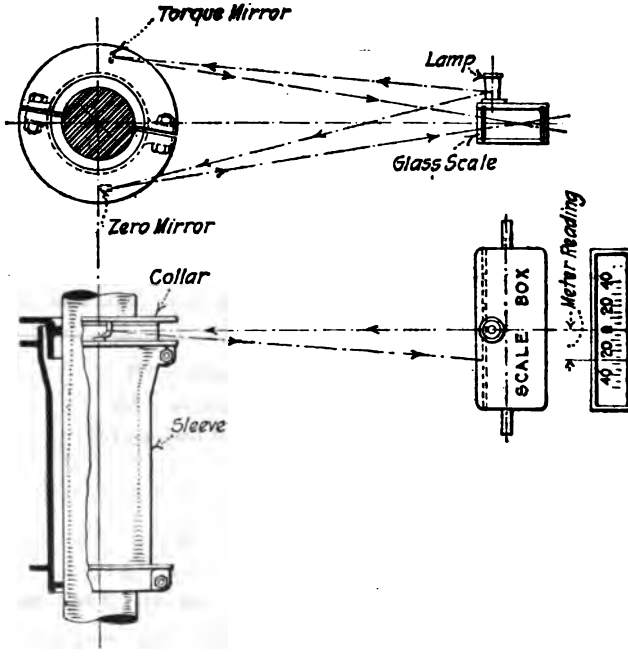

FIG. 31.—Hopkinson-Thring Torsion Meter.

to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn the torque mirror causing the beam of light to move along the scale, the deflection produced being directly proportional to the torque applied to the shaft. Hence, knowing the R.P.M. and the modulus of the elasticity of the shaft, the S.H.P. can be readily calculated.

With this torsion meter, as with all others, the shaft should be calibrated to determine the constant of the instrument. It may be calculated within 2 or 3 per cent, if the shaft section within the instrument is uniform, but to insure accuracy, without which all power tests are only of relative value, the shaft should be calibrated.

The Denny-Edgecombe Torsion Meter is another mechanical type meter that is carried wholly on the outside of the shaft. It consists of a stump and sleeve, made in halves, the stump being secured rigidly to the shaft. The sleeve is secured to the shaft only at the end most remote from the stump, the end nearest the stump being fitted with roller bearings which are adjustable and roll upon the spigot of the stump. At this end of the sleeve there are two arms corresponding to similar arms on the stump. As the shaft twists the sleeve remains rigid and there is a transverse displacement of its arms relative to those of the stump corresponding to the twist of the shaft between the two sets of securing bolts.

One of the arms has a short rack of involute teeth upon its periphery. The other carries an enclosed multiplying gear which meshes with the rack and transmits the displacement to the indicator which multiplies the motion and indicates the amount of torque. A recorder may be attached to the front of the indicator for the purpose of making a permanent diagram.

SECTION 15

TESTS, TRIALS AND INSPECTION OF MACHINERY, CLASSIFICATION SOCIETIES

BY

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CONTENTS

TESTS, TRIALS, INSPECTIONS

BY F. W. STERLING

	PAGE
Tests for Machinery before same is placed on board	142
Piping before same is placed on board	143
Machinery after same is placed on board	1431
Machinery, preparatory to Trials	1433
Trials of Machinery	1434
Dock Trials	1435
Contract Acceptance Trials	1435
Commissioning, Post Repair and Laying Up Trials	1448

CLASSIFICATION SOCIETIES

BY K. D. WILLIAMS

Names and Addresses of Societies	1450
Steamboat Inspection Service	1450
Publications	1451
Legal Status	1451
Bureau of Standards	1452
Publications	1452
Legal Status	1452
American Bureau of Shipping	1452
Publications	1453
Legal Status	1453
American Society for Testing Materials	1453
Purpose	1454
Publications	1454
AmericanSociety of Mechanical Engineers	1454
Publications	1454
Legal Status	1454
Bureau of Explosives	1455
American Institute of Electrical Engineers	1455
Publications	1456
Legal Status	1456
Master Car Builder's Association	1456
Purpose	1456
American Railway Engineering Association	1457
Purpose	1457
Publications	1457
Legal Status	1457
American Institute of Weights and Measures	1457
Purpose	1457
Publications	1458
American Society of Naval Engineers	1458
Purpose	1458
Publication	1458

TESTS, TRIALS AND INSPECTION OF MACHINERY

BY

FRANK W. STERLING

TESTS

Practically all parts of machinery and piping subject to pressure are fitted with **relief valves** set to operate at pressures considerably below the safe working pressures. All such parts for new vessels should be tested during manufacture or erection, and on vessels already in service parts after being repaired should be tested before being placed in operation.

Table 1. Tests of Boilers and Machinery Before Same is Placed on Board

Tests to be by Water Pressure

Parts to be tested	Test pressure per square inch above atmosphere
Reciprocating engine parts:	
H.P. cylinder and connections	1.5 boiler pressure
H.P. cylinder valve chest	1.5 boiler pressure
I.P. cylinder and connections	0.68 boiler pressure
I.P. cylinder valve chest	0.68 boiler pressure
L.P. cylinder and connections	0.34 boiler pressure
L.P. cylinder valve chest	0.34 boiler pressure
Turbine engine parts:	
H.P. turbine	1.5 working pressure
L.P. turbine	1.5 working pressure
H.P. reverse turbine	1.5 working pressure
H.P. cruising turbine	1.5 working pressure
L.P. cruising turbine	1.5 working pressure
Main boilers	1.5 working pressure
Cylinders and all parts of auxiliary machinery subject to boiler pressure	1.5 boiler pressure
Fittings and connections subject to boiler pressure	1.5 boiler pressure
Auxiliary machinery:	
Circulating pump casings	30 lbs.
Condenser shells and water chambers	30 lbs.
Distiller heads, shells, and tubes	50 lbs.
Evaporator shells	75 lbs.
Evaporator steam heads and tubes;	
First effect	Boiler pressure
Second effect	75 lbs.
Feed-water heater heads, when on discharge side of pumps	2.0 boiler pressure
Feed-water heater shells	75 lbs.
Air cylinders of air pumps	30 lbs.
Fuel-oil heater heads and tubes	600 lbs.
Fuel-oil heater shells	1.5 boiler pressure
Pumps—pump end, including cylinder, valve chests and air vessels:	
Distiller, circulating	150 lbs.
Distiller, fresh water	100 lbs.
Dynamo-room hot-well	100 lbs.
Evaporator brine	100 lbs.
Evaporator feed	100 lbs.
Feed	2.0 boiler pressure
Fire and bilge	225 lbs.
Forced lubrication	100 lbs.
Fuel-oil booster	150 lbs.
Fuel-oil service	600 lbs.
Oil cooler, circulating	100 lbs.
Oil drain	100 lbs.
Reserve feed water	100 lbs.
Tanks, air compressor (depending on service)	150–225 lbs.
Tanks, feed, filter, hot-well, distiller test, boiler compound, lubricating oil, waste, store, etc	Filled

Table 2. Tests of Piping Before Same is Placed on Board

Tests to be by Water Pressure

Pipes to be tested	Test pressure
Air pump, main and dynamo, suction and discharge	50 lbs.
Bilge discharge	150 lbs.
Bilge suction	15 lbs.
Bleeder	1.5 boiler pressure
Blow, boiler	1.5 boiler pressure
Blow, evaporator	100 lbs.
Circulating pumps, main and dynamo, suction and discharge	100 lbs.
Distiller, circulating	150 lbs.
Distiller, fresh-water	100 lbs.
Drains to traps, high pressure	1.5 boiler pressure
Drains, separators	1.5 boiler pressure
Drainage suction	100 lbs.
Escape	30 lbs.
Exhaust, auxiliary	50 lbs.
Exhaust, branch	50 lbs.
Exhaust, dynamo	50 lbs.
Exhaust, to atmosphere	50 lbs.
Exhaust, to feed heater	50 lbs.
Exhaust, main	30 lbs.
Feed, discharge	1.5 boiler pressure
Feed, evaporator	100 lbs.
Feed, suction	100 lbs.
Fire-main supply connections	225 lbs.
Fire main to stern tubes	225 lbs.
Fireroom hydrants	100 lbs.
Flushing system connections	100 lbs.
Forced lubrication	60 lbs
Fresh water from ship's side	100 lbs.
Fresh-water pumping system	100 lbs.
Indicator	1.5 boiler pressure
Oil-fuel suction	50 lbs.
Oil-fuel discharge to burners	600 lbs.
Overboard discharge	150 lbs.
Overflow from feed tanks	30 lbs.
Pneumatic main (depending on service)	150–225 lbs.
Reserve feed-tank connections	100 lbs.
Sea suctions	100 lbs.
Steam, auxiliary	1.5 boiler pressure
Steam, branch	1.5 boiler pressure
Steam, galley	100 lbs.
Steam, dynamo	1.5 boiler pressure
Steam, main	1.5 boiler pressure
Steam-heating-system supply	150 lbs.
Steam, from evaporators	75 lbs.
Steam radiators	150 lbs.
Steam, to first-effect evaporators	1.5 boiler pressure
Steam, to second-effect evaporators	75 lbs.
Steam, to sea valve	1.5 boiler pressure
Vapor	30 lbs.
Whistle and siren	1.5 boiler pressure

Tables 1 to 3 inclusive give the **machinery tests required on new vessels** built for the U. S. Navy. Although these tests increase the initial cost of the machinery of a vessel, they do serve to insure first-class material and workmanship and an initial condition of the machinery which in service will more than compensate the owner for the increased first cost.

Turbine cylinders may be tested in sections, but in all cases the test pressure should be at least 1.5 times the highest working pressure to which any part of the section will be subjected. No water test should be applied after turbine cylinders have been annealed, except the test for air leaks as specified under 7, p. 1432.

Tests of boilers and machinery before same is placed on board are given in Table 1.

Tests of piping before same is placed on board are given in Table 2.

Tests of valves, manifolds, and castings for pipes before being placed on board are given in Table 3.

In addition to the tests specified in Tables 1 and 3, **steel castings for steam** should be tested under steam, before being placed on board, to 25 per cent in excess of working pressure, sufficient time being allowed to thoroughly heat the casting before being inspected. Porous and other defective castings may be detected by this test that would successfully pass the specified water pressure tests, which are for strength.

Table 3. Tests of Valves, Manifolds and Castings for Pipes Before being Placed on Board

Tests to be by Water Pressure

Valves, manifolds, and castings	Test pressure
Feed valves and castings	1.5 boiler pressure
Steam valves	1.5 boiler pressure
Exhaust valves	100 lbs.
Fuel-oil discharge	600 lbs.
Sea, suction and outboard delivery gate valves, valves open	100 lbs.
Sea suction and outboard delivery gate valves, valves closed	50 lbs.
Water valves, low pressure	100 lbs.
Composition castings for feed	2 boiler pressure
Steel and composition castings for steam	1.5 boiler pressure
Composition castings for low pressure	100 lbs.

Tests to be Made on Machinery, after Same has been Placed on Board

1. **Boilers.** Close boiler against draft of air through furnaces and break all pipe joints next to valves on boiler to permit tightness of valves to be proved. Fill boiler with fresh water of approximately the same temperature as the surrounding air, taking care that boiler is completely full before closing air valve. Pump boiler to 1.5 times boiler pressure and lock pump throttle valve. If the boiler has been previously fully and permanently assembled and tested in accordance with Table 1, this pressure may be reduced to 1.25 times boiler pressure.

Note time, gage pressure and temperature of fireroom every half hour and examine boiler for leaks every hour for 24 hours. **A drop of more than 10 per cent** of test pressure **in 24 hours** should be considered unsatisfactory. If loss through leakage is excessive and such leakage exists at tube joints and drum seams, tubes must be rerolled and seam calked. If loss is high and very slight leakage exists at tube joints, drum seams, and gaskets, valves must be reground.

If **small tube leakage** develops and drops form very slowly, tubes should not be disturbed as such leaks will take up under steam and will cause less damage to the tube and tube-sheets than will excessive rolling.

2. **Boilers** should be tested under steam to 1.25 times boiler pressure and all parts should be made tight before being clothed and lagged.

3. **Main and Auxiliary Feed Systems** as a whole, after installation, should be tested by water pressure from and including pressure parts of the pumps up to the valves on the boilers to 2 times boiler pressure.

4. **Condensers** should be tested, after installation, by water pressure to 15 lbs. above atmosphere

5. **Oil-fuel System** between service pump and boilers should be tested by oil pressure, after installation, to 600 lbs. above atmosphere.

6. **All Parts** after being secured on board should be tested under working conditions to working pressures.

7. **Main Turbines and Condensers for Air Leaks.** After the main and cruising turbines and condensers are installed in the vessel and all pipe connections of every description that will be subject to pressure below atmosphere have been made, including connections to gages, air pumps, turbine drains, etc., an air test for air leaks should be made which will consist of filling the condensers, turbines, exhaust trunks, and connections subject to the vacuum of the condenser with air under a **pressure of 10 lbs.** per sq. in. at the highest part of the system, **using soap suds** around all seams to detect leaks, and all leaks should be made tight. This test should be carried out preferably after the dock trial and before trials in free route

8. **No Covering or Lagging** should be on cylinders, condensers, or pipes during the foregoing tests.

9. **Circulating Pumps** should be tested on board the vessel by working under condenser conditions and by discharging water from the bilge.

10. **A Four-hour Capacity Test of Evaporating and Distilling Plant** should be made under service conditions, using seawater feed and maintaining a density of $\frac{5}{32}$" in the evaporators. The distilled water should contain not more than 2 grains of chlorine per gallon.

11. **Refrigerating Machines** should be tested for 12 hours after getting up to working pressure with circulating water at a temperature of about 70 deg. F. This test is designed to demonstrate the ability of the plant to reduce the temperatures in the cold-storage compartments to the specified point and to make ice. If the plant is designed for ice making, as well as for refrigeration, a batch of ice should be made each four hours during the test.

12. **Steering Gear.** The steering gear should be tested by running under each main equipment from hardover to hardover for a continuous period of four hours with the vessel at rest afloat. The rate of operation for this test should be not less than 20 complete cycles per hour. A careful examination of all bearings should be made during this run to see that there is no undue heating and that all parts are functioning properly and without undue noise. For torpedo boat destroyers and other vessels of special light construction, the duration of this test should be two hours instead of four.

The steering gear should be operated from each of the various stations throughout the ship a sufficient number of times to insure proper working; the graduations on the steering standards and elsewhere should be verified; the agreement of movement of rudder with direction of movement of steering wheels or master controllers should be verified; the stops for engine and rudder and the rudder angle indicator should be verified. The number of turns of steering wheels from hardover to hardover and the force required to turn them should be noted. For tests underway see p. 1442.

13. **Telemotor piping and telemotor cylinders should** be tested in the ship to a pressure of 500 lbs. per sq. in., and must be absolutely tight under this pressure. All **rams, piping,** and **fittings** used in connection with electrohydraulic steering gears, should be given a hydrostatic test to double the maximum working pressure. They should withstand this test without showing objectionable leakage.

14. **The Anchor Engine** should be tested on or before the official trial of the vessel by hoisting and lowering two bower anchors simultaneously, in 30 fathoms of water or in the greatest depth of water obtainable in the vicinity of the building yard, continuously at the approximate rate of 6 fathoms of chain per minute. If the depth of water is less than 30 fathoms, weights shall be attached to the anchors to compensate for the lesser depth, the weight, however, not to exceed the weight of the anchor. The operation of the windlass and any heating of thrust and worm bearings should be noted. The windlass brake and locking device should be tested.

In the case of steam windlasses the duration of the test should be one hour.

In the case of electric windlasses the test should be divided into three divisions, viz., the wildcats being labeled "A," "B," and "C," the test shall be run continuously with all motors operating together as follows: "A" and "B"—one-half hour; "B" and "C"—one-half hour; "C" and "B"—one-half hour. For further tests in deep water see p. 1448.

15. The tests of evaporating and distilling plant, of refrigerating machines, of steering and anchor gear, and the bilge test of circulating pumps should be made before trials in free route.

Tests of Machinery, Preparatory to Machinery Trials

Boiler Tests Before Trials. Prior to a machinery trial, each boiler that has been longer than six months without having had full pressure of steam raised in it should be subjected to a **test by cold water** pressure and to a **test by steam** pressure. The water pressure applied should be 25 per cent in excess of the authorized safety-valve setting. Steam pressure of not less than 95 per cent of the authorized safety-valve setting should be raised and maintained for at least one hour, after which the boiler, with the water retained in it, should be allowed to cool off. Throughout these tests, and afterward when cold, the boiler should be carefully and thoroughly examined for indications of any defects or weakness, particularly as to the tightness of the shells, drums, joints, tubes, nipples, manhole and handhole plates, and other pressure parts.

Inspection of Boilers Before Raising Steam for Tests. Previous to raising steam for testing the condition of boilers, care should be taken to see that the **dry pipes** are clear and their openings not clogged, that the **internal feed pipes** are well secured, and that the **zincs** are efficiently connected and in proper condition. The following parts, with all gear connected to them, should be examined and actually worked to see that they are in proper condition: (1) Main and auxiliary stop valves; (2) safety and sentinel valves; (3) surface and bottom blow valves, pipes, and sea valves; (4) steam and water gages, cocks, and pipes; (5) feed check and stop valves and pipes; (6) drain cocks and pipes; (7) all other boiler fittings connected with the safety of operation and management.

Preparation for Tests under Steam. While making preparations for tests under steam, **all valves** that are to be used should be moved by hand and the **engines should be jacked over,** in both ahead and astern directions. In the case of reciprocating propelling engines, the **piston and valve clearances** should be taken. The **valve settings** should be checked if these have not been already established with certainty.

The following parts should be examined and worked as far as possible: Piston rings, springs, followers, and their fastenings; all drain cocks and traps; reversing gear, hand and steam, and other starting gear; main air, circulating, feed and hot-well pumps; feed heaters and grease extractors. All couplings,

nuts, keys, etc., connecting the various working parts, and all important holding down bolts and other fastenings of the machinery should be examined and carefully sounded. Condensers and tubes should be tested for tightness.

TRIALS OF MACHINERY

Object of Trials. Machinery trials of vessels should be held for the purpose of ascertaining the capabilities of the engines, boilers, and appurtenances, either in connection with the contract requirements for new vessels, or to determine their efficiency under service conditions, the extent of repairs necessary, the sufficiency of repairs that have been made, or the most economical rates of performance under various conditions of service.

Trials required in the U. S. Naval Service, in the case of **new vessels,** are (1) **preliminary acceptance trial,** which is held prior to the delivery of a new vessel built under contract and (2) **final acceptance trial,** which is held within six months or other specified time from the preliminary trial, to ascertain if there has appeared any defect, weakness, failure, or deterioration for which the contractor is responsible.

Builder's Trial. Before reporting a vessel ready for the preliminary acceptance trial, it is customary for the builders to give new vessels what is known as a "builder's trial," in order to insure themselves that they are ready for the official trial. This trial is optional with the builders.

Rules for Trials. The following general rules should be observed during all full-power trials, and during other machinery trials to which they may be applicable and consistent with the conditions imposed.

1. The engines should be worked so as to use all the steam made by the boilers with the pressure maintained as near the designed working pressure as practical, and the steam pressures, vacuum, number of revolutions, and other variables should be kept as nearly uniform as possible.

2. The speed of the engines should be gradually increased to the maximum attainable under the conditions imposed. Before starting a full-power trial on coal-burning vessels, the engines should be worked at about three-quarters power for a sufficient time to allow the boilers to be brought to proper steaming condition after forced draft has been started.

3. No full-speed trial should take place in less than forty fathoms of water.

4. Care must be taken to obtain correct weight or measure of the fuel used, and on coal-burning vessels that the fires are in the same condition at the end of the trial as at the beginning. The kind and quality of the fuel used should be specifically stated in the report of the trial.

5. Coal should not be hand picked but should be used as it runs from the bunkers.

6. Salt water should not be fed into the boilers under any circumstances. To this end, an ample reserve supply of fresh water should be provided in advance of the trial, and the evaporators run to make up extra feed if necessary. If the fresh feed supply fails, the trial should be discontinued rather than use salt feed

7. Should it be found desirable to continue a four-hour full-power trial beyond the four hours from the beginning, the observations should be continued until the trial is finished. Then those four consecutive hours should be taken which are to be considered as the trial period, and the difference of the counter readings at the beginning and end of this period divided by 240 will give the mean revolutions per minute for the whole period for each screw. The mean of the revolutions for all the screws will be the mean for the trial. The foregoing assumes that during the trial there has been no material variation from the average speed.

Dock Trials

Merchant Vessels. It is desirable, as well as being the usual practice among builders, to conduct a 6 hours' dock trial of machinery. Previous to starting this trial the main engines are gradually warmed up and run at slow or moderate speeds until satisfactory assurance is had that no serious difficulty will interfere with a trial of the engines at or as near the greatest speed of revolutions it is possible to maintain with the vessel moored alongside the dock.

When undergoing dock trial, all data available should be taken at half-hour intervals. Especial care is taken to note the steam pressures, vacuum, lubricating oil temperatures, condition of bearings, and general working condition of all machinery parts. On reciprocating engines, indicator cards are also taken in both ahead and astern gear for computing horsepower and for checking valve settings. A half hour before conclusion of this 6-hour trial the engines should be reversed and run astern for 10 to 15 minutes at the maximum speed obtainable, again reversed and run ahead for the remainder of the trial.

Examinations After Dock Trial. (*a*) In the case of **turbine engined ships,** driven through **reduction gears.** Gear should be carefully examined, removing all inspection plates in the gear casings that may be provided for this purpose. (*b*) In the case of **reciprocating engined ships.** Cylinders and valve chest covers should be removed and examination made of the inside of the cylinders and the condition of piston rings.

Naval Vessels. In modern war vessels, which are much more highly powered in proportion to their size than are merchant vessels of the usual type, **dock trials** as above described are not possible in the case of completed vessels. In lieu of such trials it is usual, after getting steam on the boilers to run each and every steam driven unit separately at or above the maximum speed at which it will run in service when under full power. In the case of geared turbine driven installations, the main turbines are operated with one of the line shaft couplings disconnected, the object being to finally test the turbines and gears for dynamic balance.

Contract Acceptance Trials

The General Purpose of a contract acceptance trial is to determine any or all of the following data: (1) Highest speed or power that can be maintained for a specified interval; (2) fuel or water consumption at various speeds; (3) general strength and condition of vessel and machinery; (4) design data for use in future design.

General Procedure. The first operation, after bringing the vessel to contract displacement, is standardization, described on page 1436. This is followed by runs of four or more hours each at various speeds or powers, including full power, during which runs fuel, water and lubricating oil consumption are measured and all possible steaming data is recorded. The displacement at the *middle* of each trial should be the contract displacement. A backing test is conducted and the steering and anchor engines are tested.

General Observations. In addition to the engineering data recorded on the trials, the following observations should be made: Working of the main engines or turbines, bearings, reduction gears and pinions, lubricating oil system, condensers, state of vacuum, pumps and all other auxiliaries, boilers, burners, fireroom draft, ventilation of machinery space, steering qualities,

steadiness and seaworthiness of the vessel, whether the vessel and machinery are well built and in accordance with contract plans and specifications and the state of completion of same.

Standardization is conducted to obtain relations between speed, revolutions and power. The vessel is run over a course of known length at various predetermined speeds. The average R.P.M. and I.H.P. or S.H.P. corresponding to these speeds are observed and curves are plotted showing the various desired relations. These curves are used for determining performance in free route. Standardization is carried out as follows:

The vessel is weighted so as to bring her to the draft and trim corresponding to the contract displacement. Trial is over a measured course, preferably a nautical mile measured by ranges ashore; the **measured mile at Rockland, Me.**, or that at the **Delaware Breakwater** are the best trial ranges on the Atlantic Coast, and the one at **Santa Barbara Channel, Cal.**, is an excellent course on the West Coast.

A series of runs are made as follows: Five consecutive runs, alternating in direction over the course, should be made at the highest speed attainable under full power and at as nearly the same number of revolutions as possible to determine the highest point. At least three consecutive runs, alternating in direction and at as nearly the same number of revolutions as possible, should be made for each one of the other points at the reduced speeds. Each series of runs at the same speed should be uninterrupted and should be performed in sequence and while the tide, if any, is running in one direction. If it should be found necessary to throw out any run, a sufficient number of additional runs should be made at that speed to produce at least three consecutive runs, alternating in direction over the course.

The **runs over the measured course** should be made back and forth over the same water; between runs the vessel should be taken well away from the measured course, so as to insure the attainment of the required speed on the next run, and should be straightened on the course while at least one mile from the first range.

At **full power, when standardizing**, the engines and boilers should be worked to the utmost extent of their capabilities, not only when running the measured course but throughout the intervals between runs. The engine speed should be so regulated that the steam pressure in the boilers will be maintained at a maximum until the whole number of consecutive runs at the highest speed is completed.

At **reduced power, when standardizing**, a sufficient number of boilers should be used to insure a steady steam pressure while working at the required revolutions. Pressure should be maintained steady and revolutions should be kept as nearly constant as possible, both on and off the courses, throughout a series of runs at the same speed.

The following **data for each run** should be recorded: (1) elapsed time on course; (2) total revolutions of each engine while on course; (3) gage steam pressure at main engines; (4) mean effective pressures referred to L.P. pistons for reciprocating engines; (5) I.H.P. or S.H.P. of main propelling engines (from cards taken during runs; see p. 1403 for measuring power). The following data should also be recorded as pertinent to the trial: Trim, mean draft and displacement; condition of vessel's bottom and date of last docking; state of sea, and direction and force of wind.

Deductions from Standardization Runs are made as shown in Table 1, and **curves of R.P.M. vs. speed and power** are constructed as shown in Fig. 1. The propeller slip ratio, at any speed, may be found as follows:

Slip ratio $= \dfrac{p \times R - 101.3V}{p \times R}$ where p = propeller pitch in feet, R = revolutions per minute and V speed of ship in knots.

Trials of U.S.S. Wadsworth. The procedure and information obtained from any preliminary trial may be illustrated from the following description

Table 1. U.S.S. Wadsworth

Standardisation Trial Data—June 21, 1915

No. of run	Time on course		Speed in knots	R.P.M.			S.H.P.		
	Min.	Sec.		Star. engine	Port engine	Mean	Star. engine	Port engine	Total
1	7	15.8	8.26	101.32	101.66	101.49	79	72	151
2	8	12.0	7.32	102.35	102.61	102.48	79	87	166
3	6	59.5	8.58	104.30	102.71	103.51	81	65	146
Mean of group			7.87	.....	.....	102.49	...	...	157
4	5	25.1	11.07	151.08	151.05	151.07	277	245	522
5	4	44.3	12.66	156.34	154.82	155.58	331	240	571
6	5	23.5	11.13	150.15	149.98	150.07	244	233	477
Mean of group			11.88	.....	.....	153.08	...	...	535
7	3	35.4	16.71	211.97	211.54	211.76	658	702	1,360
8	3	49.8	15.67	210.29	210.02	210.16	727	682	1,409
9	3	35.8	16.68	211.06	211.09	211.08	670	700	1,370
Mean of group			16.18	.....	.....	210.79	...	...	1,387
10	2	57.8	20.25	272.54	272.62	272.58	1,520	1,578	3,098
11	2	52.0	20.93	271.15	271.80	271.48	1,493	1,592	3,085
12	2	56.3	20.42	272.97	272.99	272.98	1,503	1,638	3,141
Mean of group			20.63	.....	.....	272.13	....	....	3,102
13	2	27.1	24.47	337.23	337.53	337.38	3,237	3,335	6,572
14	2	29.3	24.11	337.39	337.27	337.33	3,203	3,285	6,488
15	2	26.9	24.51	337.67	337.72	337.70	3,170	3,289	6,459
Mean of group			24.30	.....	.....	337.44	....	....	6,502
16	2	13.7	26.93	388.09	387.57	387.83	5,187	5,343	10,530
17	2	11.2	27.44	389.46	389.62	389.54	5,195	5,388	10,583
18	2	13.2	27.03	388.00	388.36	388.18	5,121	5,425	10,546
Mean of group			27.21	.....	.....	389.02	....	....	10,561
19	1	58.4	30.41	449.14	449.02	449.08	7,608	7,764	15,372
20	2	00.0	30.00	447.52	448.22	447.87	7,330	7,812	15,142
21	1	57.9	30.53	450.12	451.24	450.68	7,657	7,739	15,396
Mean of group			30.24	.....	.....	448.88	....	....	15,263
22	1	55.5	31.17	485.22	485.35	485.29	8,904	9,115	18,019
23	1	52.4	32.03	486.30	486.17	486.24	8,408	9,232	17,640
24	1	55.6	31.14	484.90	484.43	484.67	9,310	9,059	18,369
25	1	51.6	32.26	486.81	486.89	486.85	8,668	9,114	17,782
26	1	55.7	31.11	487.29	487.33	487.31	9,254	9,186	18,440
Mean of group			31.64	.....	.....	486.02	....	....	18,005

of the trials of the U.S.S. Wadsworth, written by *Mr. Henderson B. Gregory*, and published in the *Proc. A.S.N.E., July*, 1915. The Wadsworth was built by the *Bath Iron Works, Ltd.* She is a twin-screw vessel, driven by Parson's turbines, fitted with reduction gear, and designed for a speed of 30 knots, at a displacement of about 1050 tons, with the turbines developing 17,500 S.H.P.

"The contract required the following trials for each vessel:

"(*a*) A progressive trial over a measured-mile course at Rockland, Me., for standardizing the screws, extending from maximum speed (at least 30 knots) down to a speed of 8 knots. (Method is described on p. 1436.)

"(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 30 knots.

Table 2. Trial Data—U.S.S. Wadsworth

	4-hr. full-speed trial (*b*)	4-hr. 25-knot trial (*c*)	4-hr. 20-knot trial (*d*)	4-hr. 16-knot trial (*e*)	4-hr. 12-knot trial (*f*)
Date of trial	June 22, 1915	June 22, 1915	June 23, 1915	June 23, 1915	June 24, 1915
Speed in knots	30.685	24.99	20.086	16.004	11.991
Draught, mean on trial, feet and inches	9–1	9–1⅝	9–2¼	9–1¼	9–1¼
Displacement, corresponding, tons	1,034.2	1,042	1,049	1,043.7	1,044.8
Slip of propellers, per cent, starboard-port	21.76 | 21.28	16.04 | 15.92	10.62 | 10.34	9.86 | 9.67	8.95 | 8.73
Slip of propellers, per cent, mean	21.52	15.98	10.48	9.77	8.84
Number of boilers used	4	4	2	1	1
Number of burners used (12 per boiler)	36.375	16	8	5.25	2.625
Heating surface, square feet	21,500	21,500	10,750	5,375	5,375
Heating surface, square feet per S.H.P.	1.335	2.887	3.669	3.743	9.755
Cooling surface, square feet (main cond.) per S.H.P.	0.671	.451	3.689	7.527	19.617
Pressures (average):	S. eng. | P. eng.	S. eng. | P. eng.	S. eng. | P. eng.	S. eng. | P. eng.	S. eng. | P. eng.
Main steam at boilers, pounds gage	250.1	248.2	202	150	151.25
Main steam, eng. room, pounds gage	240.88 | 240.06	242.13 | 242.00	194.75 | 194.25	140.00 | 140.25	141.25 | 140.00
H.P. ahead chest, pounds absolute	225.38 | 215.06	205.63 | 215.00	89.38 | 107.75	54.50 | 64.88	34.00 | 37.50
H.P. main belt, pounds absolute	184.94 | 205.69	73.25 | 88.13	32.75 | 41.00	17.50 | 20.25	15.00 | 15.00
H.P. 8th expansion, pounds absolute	85.31 | 95.63	32.50 | 39.00	13.63 | 15.00	8.75 | 9.56	6.13 | 5.38
L.P. chest, pounds absolute	23.73 | 27.30	9.50 | 13.50	5.30 | 6.00	4.50 | 4.50	2.39 | 2.91
Vacuum, main cond., in. of mercury	28.48 | 28.94	29.26 | 28.98	29.26 | 29.05	28.90 | 29.03	28.46 | 28.65
Vacuum, aux. cond., in. of mercury	Last hr. 21.60	Last hr. 18.00	Not used	Not used	Not used
Barometer, inches of mercury	29.53	29.12	29.46	29.76	29.55
Aux. steam, pounds gage	241.60	243.75	194.50	141.12	140.63
Aux. exhaust, pounds gage	10.76	5.13	4.57	9.56	5.10
Gland steam, pounds gage	0.88 | 0.25	0.30 | 0.19	0.19 | 0.25	0.49 | 0.25	0.41 | 0.25
Main feed, at pumps, pounds gage	345.00	327.82	353.13	250.00	240.00
Forced lubrication, pounds gage	36.94	33.63	30.05	33.25	28.50
Oil at main thrust, pounds gage	6.38 | 7.91	5.80 | 7.11	5.55 | 7.80	6.46 | 8.76	7.01 | 8.15
Oil at reduction gear (H.P. pinion), pounds gage	6.87 | 4.05	6.44 | 3.49	6.43 | 4.20	7.11 | 4.94	7.01 | 4.88
Oil at reduction gear (H.P. pinion), pounds gage	7.01 | 5.68	6.44 | 4.81	6.06 | 4.91	6.89 | 5.98	6.90 | 5.55
Fuel oil, pounds gage	116.40	103.76	84.0	99.38	107.5
Air pressure in firerooms, in. of water	6.02	3.14	2.5	2.55	2.5
Revolutions or double strokes per minute (average):					
Main shafts, starboard-port	458.85 | 460.44	349.50 | 349.72	263.57 | 263.82	208.44 | 203.44	154.61 | 154.56
Main shafts, mean	459.65	349.61	263.70	208.44	154.59
Main air pumps	19.81 | 18.94	14.38 | 16.88	(1) 27.00	(1) 21.75	(1) 21.63

Aux. cond. air pump	Last hr. only 61.80	Last hr. only 43.00	Not used	Not used	Not used
Aux. cond. circulating pump	1,134	1,121.66	Not used	Not used	Not used
Main feed pumps	(2) 24.82	(2) 14.07	(1) 14.00	(1) 8.76	(1) 9.00
Forced lub. pumps	(2) 59.60	(1) 75.00	(1) 52.13	(1) 43.50	(1) 32.38
Forced draft blowers	(4) 1,335	(4) 886.25	(2) 774.38	(2) 812.82	(2) 778.76
Fuel oil service pumps	(3) 23.82	(2) 17.52	(1) 19.13	(1) 15.88	(1) 9.50
Shaft horsepower (average):					
Starboard shaft	8,066	3,729	1,534	797	292
Port shaft	8,034	3,719	1,395	639	259
Total	16,100	7,448	2,930	1,436	551
Temperatures, deg. F. (average):					
Main air pump discharge	69.19	63.37	59.88	62.88	69.88
Main feed tank	71.38	64.13	61.13	64.13	65.63
Main injection	54.81	54.57	52.38	52.57	52.07
Main outboard delivery	67.50 | 68.25	60.75 | 60.88	56.50 | 56.25	56.25 | 55.50	53.63 | 53.13
Outside air	61.75	61.13	62	61.63	59.75
Engine room	88.25 | 67.38	79.88 | 66.00	78.75 | 60.38	81.50 | 69.13	76.25 | 71.75
Firerooms	95.75	92.63	95.38	93.00	84.88
Feed water	186.25	199.75	213.50	230.25	218.75
Oil drains, H.P. turbines, for'd bearings	125.81 | 119.06	115.38 | 106.13	103.50 | 98.38	98.00 | 91.88	93.38 | 87.13
Oil drains, H.P. turbines, after bearings	126.69 | 116.25	110.50 | 102.63	105.75 | 94.88	96.50 | 90.13	90.00 | 83.38
Oil drains, L.P. turbines, forward bearings	112.63 | 112.19	99.75 | 99.38	90.25 | 92.63	89.25 | 87.00	84.38 | 81.63
Oil drains, L.P. turbines, after bearings	112.19 | 114.38	99.00 | 103.50	91.75 | 94.88	88.63 | 90.00	82.75 | 83.88
Main thrust	110.31 | 107.25	96.38 | 91.38	87.50 | 84.88	83.63 | 82.29	74.25 | 74.38
Reduction gear pinion bearings, H.P.	112.75 | 112.44	98.75 | 111.00	89.75 | 98.75	84.25 | 92.13	80.00 | 85.00
Reduction gear pinion bearings, L.P.	114.56 | 118.38	100.75 | 104.63	91.25 | 93.88	86.25 | 89.25	80.03 | 83.13
Fuel oil to heaters	69.03	70.94	70.25	72.38	71.63
Fuel oil to burners	151.35	153.82	164.13	162.25	162.50
Smoke pipes	664.69	465.31	408.58	442.50	443.75
Water consumption:	First 3 hours	First 3 hours			
Pounds per hour, measured	199,467	92,547.67	44,985.25	30,449.25	19,793.75
Pounds per hour, reserve feed	2,207	2,263	291.5	None	None
Pounds per hour, total evaporated	201,674	94,810.67	45,276.75	30,449.25	19,793.75
Pounds per hour, per S.H.P.	12.533§	12.723†	15.453	21.204	35.923
Pounds per hour, evaporated per sq. ft. of H.S.	9.38	4.41	4.212	5.665	3.683
Pounds per hour, evaporated per lb. of oil	13.126°	13.85°	14.533°	14.694°	14.167°
Pounds per knot run	6,574.968▲	3,795.919▪	2,254.145	1,902.602	1,650.717
Fuel oil consumption:					
B.t.u. per pound	19,658.5	19,658.5	19,658.5	19,658.5	19,658.5
Specific gravity at 60 deg. F.	0.8726	0.8726	0.8726	0.8726	0.8726
Pounds per hour*	15,364.09°	6,845.41°	3,115.40°	2,072.16°	1,397.21°

Table 2. Trial Data—U.S.S. Wadsworth—(*continued*)

	4-hr. full-speed trial (*b*)	4-hr. 25-knot trial (*c*)	4-hr. 20-knot trial (*d*)	4-hr. 16-knot trial (*e*)	4-hr. 12-knot trial (*f*)
Pounds per knot run	500.90[▲]	274.07[○□]	155.10[○]	129.48[○]	116.52[○]
Pounds per knot at	30 kts. 473[○]	25 kts. 275[○]	20 kts. 153.5[○]	16 kts. 129.48[○]	12 kts. 116.52[○]
Pounds per knot guaranteed at	641[○]	403[○]	245[○]	233[○]	203[○]
Pounds per knot below guarantee	168[○]	128[○]	91.5[○]	103.52[○]	86.48[○]
Pounds per hour per S.H.P.	0.955[○]§	0.919[○]†	1.063[○]	1.443[○]	2.536[○]
Pounds per hour per sq. ft. of H.S.	0.715[○]	0.318[4○]	0.29[○]	0.386[○]	0.26[○]
Special data taken in for'd engine room: (All readings inches of mercury, absolute)					
L.P. turbine exhaust	2.18	1.33	1.14	1.29	1.55
L.P. turbine ahead dummy	2.13	1.40	1.04	1.18	1.63
L.P. turbine astern dummy	2.11	1.39	1.04	1.16	1.61
Main condenser, top	1.71	1.30	1.04	1.15	1.63
Main condenser, bottom	1.49	1.26	1.05	1.15	1.64
Air pump suction	2.19	2.28	2.14	2..0	3.46

NOTES.—Figures in brackets () indicate number of auxiliaries in operation. * The pounds per hour, at 19,658.5 B.t.u. per pound of oil, for the various trials were; (*b*) first 3 hours, 15,240.19; (*c*) first 3 hours, 6790.21; (*d*) 3090.28; (*e*) 2055.45; and (*f*) 1385.94. ○ On basis of 19,500 B.t.u. per pound of oil. § On basis of 16,091 S.H.P., average for first 3 hours. † On basis of 7452 S.H.P., average for first 3 hours. ▲ On basis of 30.673 knots, average for first 3 hours. □ On basis of 24.977 knots, average for first 3 hours.

Table 8. Supplementary Trial Data—U.S.S. Wadsworth

	Last hour of	
	4-hour full-speed trial	4-hour 25-knot trial
Shaft horsepower	16,129.00	7,435.00
Speed in knots	30.72	25.03
Oil burned, pounds	15,446.68	6,658.55
Oil burned, pounds	[a]15,572.23	[a]6,712.68
Oil per knot run, pounds	[a]506.94	[a]268.19
Water per hour, total consumption from first 3 hours of trial, pounds	199,467.00	92,548.00
Main turbines and augmenter only, pounds	180,468.00	78,537.00
Auxiliaries and losses, pounds	[b]18,999.00	[b]14,011.00
Main turbines and augmenter only per S.H.P., pounds	11.19	10.56

[a] On basis of 19,500 B.t.u. per pound of oil.
[b] Difference between total consumption, first 3 hours and consumption main turbines and augmenter, last hour.

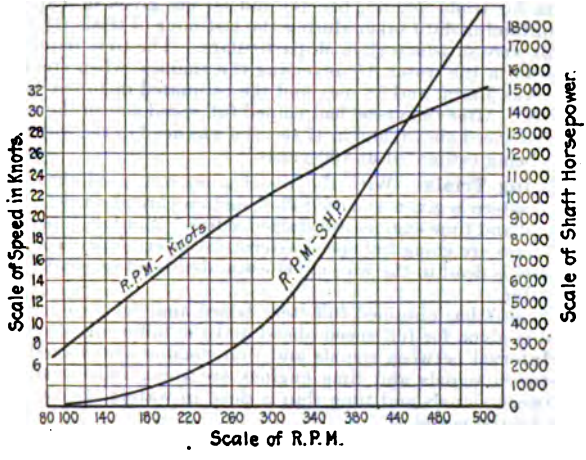

FIG. 1.

"Four fuel-oil and water-consumption trials, designated (*c*), (*d*), (*e*) and (*f*), of four hours' duration in the open sea in deep water, at an average uniform speed of 25, 20, 16 and 12 knots, respectively, as nearly as possible. The trial to be conducted as nearly as possible to service conditions.

"(*g*) Trials to determine the ability of the vessel to back satisfactorily at full speed and at cruising speed, and one trial to determine the time required to bring the vessel dead in the water, and the distance head reached, when steaming ahead at 30 knots.

"Fuel Oil Consumption Guarantee. The contractors guaranteed that the consumption of fuel oil per knot run, for all purposes, including that necessary for all auxiliaries in use on trial, shall not exceed 641 pounds at 30 knots, 403 pounds at 25 knots, 245 pounds at 20 knots, 233 pounds at 16 knots and

nds at 12 knots, the consumption of fuel oil at these speeds to be determined by a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*), (*e*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

"The trials were conducted June 21 to 24, 1915, inclusive. The contract speed was exceeded without effort, and the fuel-oil consumption guarantees were improved upon beyond expectation. Twenty-six runs were made on the standardization trial (*a*), as noted in Table 1, which gives the data used in plotting the curves, Fig. 1.

"In view of the decided changes in the turbine installation from past practice, it was considered both of interest and desirable to get a line on the steam consumption of the main turbines only. To accomplish this it was agreed to divide the two high-speed trials, (*b*) and (*c*), into two parts each, of three hours' and one hour's duration. During the first three hours the total water consumption, including main turbines, auxiliaries and drains, was measured, and by agreement the official fuel-oil consumption was based on this period. On the last hour of the trial the water consumption of the main turbine only was measured, plus the steam used in the vacuum augmenters, which obviously could not be separated from the former. The auxiliary exhaust and drains were led to the auxiliary condenser and not measured.

"The data for trials (*b*), (*c*), (*d*), (*e*) and (*f*) are given in Tables 2 and 3, the latter containing data taken during the last hour of trials (*b*) and (*c*).

"Trials (*g*) were satisfactory in all particulars. The time required to bring the vessel dead in the water, by reversing the engines, when steaming ahead at 30 knots, was 3 min. and 38 sec., and the estimated distance head reached was 500 yards. After the vessel had gained full speed astern the engines were reversed to full speed ahead, and in 47 sec. she was dead in the water, the sternboard made being estimated at 150 yards."

Maneuvering Trials. With the vessel going full speed ahead, the signal for full speed astern is given and the following times are observed: (1) interval between signal and time engines are started astern; (2) interval between signal and time engines are going full speed astern; (3) interval between signals and time the vessel is dead in the water; (4) estimated head reached before vessel is dead in water.

After the vessel has obtained full stern board and the revolutions are constant, signal is given for full speed ahead and the following observations are made: (1) Interval between signals and time engines are started ahead; (2) interval between signals and time engines are going full speed ahead; (3) interval between signals and time ship is dead in water; (4) estimated stern reach made by the vessel.

Backing Trial. The vessel is backed in both directions with full open throttle over the measured mile or other standardization course. Bearing temperatures should be carefully observed during this trial.

The Steering Engine should be tested during the preliminary trials by throwing the rudder from hardover to hardover with the vessel going at full speed ahead. The engine should be capable of accomplishing this in 20 seconds. This test should be performed with each combination of main gear fitted.

Each main equipment should also be tested during the preliminary trials with the vessel going astern at the speed obtained with the maximum backing power the machinery develops, as follows:

(*a*) The rudder being amidships, it shall be first brought to 15 deg. starboard and allowed to remain there one or two minutes.

(*b*) The rudder shall then be brought to 25 deg. port and allowed to remain there one or two minutes.

NEWPORT NEWS SHIPBUILDING & DRY DOCK CO. — FIRE ROOM LOG of S.S. _____ No. _____ On _____ Trial. Date _____ Sheet No. _____

Boilers: Type _____ No. _____ Des. Press. _____ D. _____ L. _____ H.S. _____ Furnaces _____ Type Draft _____ No. Blowers _____ Diam. _____ Tip _____ Eng. _____ Oil _____

Fuel Oil Plant: _____ Burners per Boiler _____ Service Pump _____ Supply Pump _____ Heaters _____

Time	Boiler Pressures						Burners in Use							Blowers		Temperatures						Pressures				Oil Strokes Pumps		Oil Consumption				Smoke	Remarks
						Avg.							Total	R.P.M.	Air Press	Boiler Room	Stack	Oil Tanks Bet. Tank Oil Heat.	Oil to Burners	Oil to Boilers Oil Heat. Strainers	Mean to Heater	F.O. Suction	F.O. Service Tank	Feed Pump Gals.	Diff.	Meter Gals.	Diff.				

FIG. 2.

NEWPORT NEWS SHIPBUILDING & DRY DOCK CO.—ENGINE ROOM LOG of S.S. _____ No. _____ On _____ Trial _____ Date _____ Sheet No. _____

Left Dock at _____ Draft: Fwd: _____ Aft: _____ Mean: _____ Displacement: _____ Mean Draft: of Trip _____ Course: _____

Returned to Dock at _____ " " _____ " " _____ " " _____ " Displacement of Trip _____ Weather _____

Boilers: No. _____ Type: _____ Sizes: _____ Pressure: _____ Total H.S. _____ Total G.S. _____ H.S. _____ G.S. _____ Draft: _____ Fuel: _____

Engines _____ Diam. Piston Rod _____ Cooling Surface _____ Air Pump: _____ Feed Pumps _____ Propeller Diam. _____ Pitch: _____ Dev. Area: _____

Engine	Time	Card No.	Counter	R.P.M. from Gov.	R.P.M. by Str. Counter	Main Steam	H.P. Check	Receivers 1st	Receivers 2nd	Receivers 3rd	Vacuum	Linked Up H.P.	Linked Up I.P.	Linked Up P	Linked Up L.P.	M.E.P. H.P.	M.E.P. I.P.	M.E.P. P	M.E.P. L.P.	I.H.P. H.P.	I.H.P. I.P.	I.H.P. P.	I.H.P. L.P.	I.H.P. Total Each Engine	I.H.P. Total Both Engines	Ave. M.E.P. Referred to L.P.	Temperatures Eng. Room	Temperatures Exhaust	Temperatures Eng. Room	Temperatures Main Jet	Temperatures Cond. Outlet	Temperatures Air Pump Outlet	Temperatures Main Feed	Temperatures Feed	H.P. & L.P. Main Circ.	Double Stroke Pumps Main Feed	Double Stroke Pumps Aux. Feed	Aux. Exhaust to Feed Heater	Remarks
Stable Readings																																						

Engine Constants: H.P. _____ I.P. _____ P. _____ L.P. _____

FIG. 3.

Table 4. Trial Data—Naval Auxiliaries

	Panama canal colliers		U. S. fleet colliers		
	Ulysses	Achilles	Cyclops	Vulcan	Mars
Date of trial	April 14 and 15, 1915	June 8 and 9, 1915	Oct. 26, 27 and 28, 1910	Sept. 15 and 16, 1909	Aug. 18 and 19, 1909
Duration, hours	24	24	48	24	24
Speed in knots	14.605	14.281	14.61	12.82	12.65
Draught, mean on trial, feet and inches	27–5¾	27–8½	27–3 11/16	24–8 1/16	24–8¾
Displacement, corresponding, tons	19,336	19,322	19,095	12,264	11,292
Pressures (average):	St'b'd | Port	St'b'd | Port	St'b'd | Port	St'b'd | Port	St'b'd | Port
Main steam, at boilers, pounds gage	200.0	194.5	192	191	...
Main steam, at engines, pounds gage	194.1 | 194.9	194.6 | 196.0	...	182.6 | 182.6	179
Main steam, at 1st rec'r, lbs. absolute	82.3 | 82.9	70.6 | 75.0	.... | 67.7	70.8 | 76.0	77.6 | 80.0
Main steam, at 2d rec'r, lbs. absolute	30.1 | 29.7	24.8 | 24.3	24.6 | 24.0	25.45 | 26.66	26.7 | 26.1
Vacuum, main condenser, inches of mercury	25.7 | 25.9	25.9 | 26.0	26.4 | 27.7	26.8 | 25.65	27.18 | 26.85
Barometer, inches of mercury	30.37	29.83	....	....	....
Main feed, at pumps, pounds gage	230.8	212.2	....	....	....
Auxiliary exhaust, pounds gage	11.3	5.81	12.5	....	....
Air pressure in ash pits, in. of water	0.5	0.258	0.41	....	2.0
Mean effective pressures, lbs. (average):					
H.P. cylinder	87.50 | 85.70	82.00 | 85.25	85.90 | 90.12	86.30 | 87.50	87.30 | 84.50
I.P. cylinder	29.60 | 32.40	27.02 | 29.32	30.79 | 31.77	27.50 | 27.30	28.80 | 30.70
L.P. cylinder	13.10 | 14.90	10.08 | 12.00	12.38 | 10.49	11.65 | 11.35	10.30 | 11.10
Referred pressure	34.93 | 37.57	33.48 | 30.32	34.27 | 33.26	33.53 | 33.41	32.87 | 37.28
Revolutions or double strokes per minute (average):					
Starboard engine	96.62	93.02	92.109	92.59	91.99
Port engine	96.30	92.58	92.336	91.76	94.69
Mean	96.46	92.80	92.223	92.18	93.34
Main circulating pumps	102.2 | 100.4	131.42 | 133.71	159 | 156	199 | 202	158 | 203
Main feed pumps	11.5	10.75	22.8	12.5	7.59
Forced draft blowers	363.5 | 389.8	370	289 | 267	485.0	522.0
Indicated horsepower (average):					
H.P. cylinder	1,185 | 1,144	1,062 | 1,098	1,106 | 1,160	631 | 633	645 | 627
I.P. cylinder	1,138 | 1,245	1,002 | 1,078	1,127 | 1,163	596 | 592	632 | 690
L.P. cylinder	1,184 | 1,575	1,028 | 1,220	1,244 | 1,077	652 | 632	581 | 643
Total	3,707 | 3,964	3,092 | 3,396	3,477 | 3,400	1,879 | 1,857	1,858 | 1,960
Total, both engines	7,671	6,488	6,877	3,736	3,818

Temperatures, deg. F. (average):					
Main injection	44.9	53.08	.....	67.2	....
Overboard discharge	119.7 | 118.2	118.5 | 116.13	.....	112.2 | 110.5	....
Feed tank	125.6	139.30	.....	.....	....
Feed water	192.6	202.54	.....	208.5	....
Engine room	70.5	75.79	70.90	.....	....
Firerooms	76.5	75.0	79.90	86.0	....
Outside air	52.3	65.07	58.00	....	....
Smoke pipe	739.2	661.04	337.00	520	....
Coal consumption data:					
Kind	New River	New River	.....	Pocahontas	Pocahontas
Quality	Excellent	Excellent	.....	Run of mine	Run of mine
B.t.u. per pound	13,577	14,897	.....	.....	.....
Pounds per knot run	791.784°	678.034°	681.314	492.434	561.502
Pounds per hour	11,564	9,683	9,954	6,313	7,103
Pounds per hour per I.H.P., main engines	1.507	1.492	1.447	1.69	1.86
Pounds per hour per square foot of grate surface	25.641	21.471	22.12	26.90	30.26
Pounds per hour per square foot of heating surface	0.589	0.493	0.514	0.619	0.696
Miscellaneous data:					
Slip of propellers, per cent, mean	9.85 | 9.32	8.77 | 7.88	10.7 | 10.95	.....	.....
Slip of propellers, per cent, mean, Starboard and port	9.59	8.33	10.825	.....	.....
Grate surface, square feet	451	451	450	234.68	234.68
Heating surface, square feet	19,635	19,635	19,379	10,200	10,200
I.H.P. per square foot of grate	17.009	14.385	15.282	15.92	16.28
Sq. ft. of heating surface per I.H.P.*	2.56	2.028	2.818	1.73	2.67
Sq. ft. of cooling surface (main cond.), per I.H.P.*	1.382	1.634	1.455	1.65	1.62
Knots per ton of coal	2.829	3.304	3.288	4.549	3.989
Main engines, type	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.
Main engines, diameter of cylinders, in.	27½ × 46 × 76	27½ × 46 × 76	27½ × 46 × 76	22 × 37½ × 60	22 × 37½ × 60
Main engines, stroke, feet	4	4	4	3½	3½

* I.H.P. main engines only. °At 14 knots. Guarantee 850 lbs.: Ulysses 703.422 lbs., Achilles 633.92 lbs.
Less than guarantee: Ulysses 146.578 lbs., Achilles 216.08 lbs.

Table 4. Trial Data—Naval Auxiliaries.—(*continued*)

	U. S. fleet colliers				
	Hector	Proteus	Nereus	Orion	Jason
Date of trial	Oct. 14 and 15, 1909	June 27 and 28, 1913	Sept. 3 and 4, 1913	July 10, 11 and 12, 1912	June 18 and 19, 1913
Duration, hours	24	18	18	48	24
Speed in knots	12.87	14.675	14.575	14.467	14.322
Draught, mean on trial, feet and inches	24–11⅝	27–2¾	27–3¾	27–0⅝	27–0½
Displacement, corresponding, tons	11,162	19,058	18,838	18,800	18,862
Pressures (average):	St'b'd | Port	St'b'd | Port	St'b'd | Port	St'b'd | Port	St'b'd | Port
Main steam, at boilers, pounds gage	196	202.3	197.5	195	199.6
Main steam, at engines, pounds gage	..... |	189.5	192.3	192	194.9
Main steam, at 1st rec'r, lbs. absolute	85.00 | 84.45	72.6 | 73.20	85.30 | 79.70	...	82.10 | 84.10
Main steam, at 2d rec'r, lbs. absolute	28.97 | 29.47	23.2 | 23.40	22.80 | 23.72	...	25.30 | 26.80
Vacuum, main condenser, inches of mercury	25.64 | 26.05	26.3 | 26.02	26.44 | 26.52	...	25.84 | 26.14
Barometer, inches of mercury	...	30.02	30.07	...	30.19
Main feed, at pumps, pounds gage	...	.....	.....	...	.....
Auxiliary exhaust, pounds gage	...	.....	.....	...	.....
Air pressure in ash pits, in. of water	2.0	0.89	0.985	1.0	1.334
Mean effective pressures, lbs. (average):					
H.P. cylinder	86.50 | 80.33	97.20 | 94.30	81.60 | 86.10	...	85.50 | 82.30
I.P. cylinder	31.95 | 30.33	30.40 | 33.75	36.70 | 37.10	...	33.90 | 34.70
L.P. cylinder	12.45 | 12.58	10.98 | 12.20	9.80 | 11.10	...	9.70 | 10.70
Referred pressure	36.17 | 34.79	33.14 | 35.16	32.24 | 34.22	...	32.48 | 33.39
Revolutions or double strokes per minute (average):					
Starboard engine	92.78	100.86	98.82	94.76	96.04
Port engine	94.57	102.14	99.27	95.24	94.35
Mean	93.675	101.50	99.07	95.00	95.20
Main circulating pumps	206.6 | 209.4	207 | 205	140.1 | 140.2	.....	121.87 | 120.37
Main feed pumps	9.48	11.5	13.85	.....	11.35
Forced draft blowers	494.00	295 | 263	335.3 | 325.1	.....	485.8 | 484.48
Indicated horsepower (average):					
H.P. cylinder	619 | 586	1,235 | 1,210	1,020 | 1,080	.....	1,091 | 1,055
I.P. cylinder	679 | 653	1,100 | 1,230	1,308 | 1,326	.....	1,284 | 1,323
L.P. cylinder	682 | 702	1,138 | 1,294	1,017 | 1,152	.....	1,002 | 1,123
Total	1,980 | 1,941	3,473 | 3,734	3,345 | 3,558	3,393 | 3,550	3,377 | 3,501
Total, both engines	3,921	7,207	6,904	6,943	6,878

Temperatures, deg. F. (average):					
Main injection	65.62	75.5	75.3	....	63.1
Main overboard discharge	116.58 | 111.83	115.0 | 106.2	103.9 | 107.2	....	108.3 | 110.8
Main feed tank	.....	124.30	119.9	....	113.5
Feed water	202.03	153.30	170.1	....	174.8
Engine room	86.20	100.90	88.4	....	87.0
Firerooms	.....	100.35	100.5	....	94.88
Outside air	15.45	80.00	74.0		66.4
Smoke pipe	550	.....	....	....	.....
Coal consumption data:					
Kind	Pocahontas	New River	New River	Pocahontas	Pocahontas
Quality	Run of mine	Picked	Run of mine	Fair	Poor
B.t.u. per pound	.....	15,163	14,841	14,388	12,688
Pounds per knot run	505.944	746.494	727.199	724.131	914.305
Pounds per hour	6,511.5	10,954.8	10,598.93	10,476	13,094.67
Pounds per hour per I.H.P., main engines	1.66	1.52	1.535	1.509	1.904
Pounds per hour per square foot of grate surface	27.746	25.47	24.649	23.809	29.761
Pounds per hour per square foot of heating surface	0.638	0.592	0.573	0.554	0.692
Miscellaneous data:					
Slip of propellers, per cent, mean	.....	10.19 | 10.64	8.97 | 9.38	14.056 | 14.488	16.05 | 14.55
Slip of propellers, per cent, mean, starboard and port	.....	10.42	9.18	14.272	15.3
Grate surface, square feet	234.68	430	430	440	440
Heating surface, square feet	10,200	18,492	18,492	18,921	18,921
I.H.P. per square foot of grate	16.708	16.76	16.056	15.78	15.632
Sq. ft. of heating surface per I.H.P.*	2.601	2.55	2.678	2.725	2.751
Sq. ft. of cooling surface (main cond.), per I.H.P.*	1.579	1.25	1.304	1.536	1.55
Knots per ton of coal	4.427	3.001	3.08	3.093	2.45
Main engines, type	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.	3-cyl., 3-exp., vert.
Main engines, diam. of cylinders, in	22 × 37½ × 60	26 × 43¼ × 74	26 × 43¼ × 74	27 × 46 × 76	27 × 46 × 76
Main engines, stroke, feet	3½	4	4	4	4

* I.H.P. main engines only. ° At 14 knots. Guarantee 850 lbs.: Ulysses 703.422 lbs., Achilles 633.92 lbs. Less than guarantee: Ulysses 146.578 lbs., Achilles 216.08 lbs.

(*c*) The rudder shall then be brought to 30 deg. starboard and allowed to remain there one or two minutes.

(*d*) The rudder shall then be brought to 35 deg. port and allowed to remain there one or two minutes.

(*e*) The rudder shall finally be brought to 35 deg. starboard and allowed to remain there one or two minutes.

(*f*) The rudder shall then be brought to amidship position.. For steering gear tests before trial see p. 1432.

Anchor Engine Trials. U. S. Navy specifications require that: "With the vessel at sea in a depth of water exceeding 60 fathoms, one of the anchors shall be backed out until the 60-fathom shackle is at the water's edge and the anchor clear of the bottom. The anchor shall then be hove in at a speed of at least 6 fathoms per min. and the results noted. Two of the anchors shall then be backed out simultaneously, one to 60 fathoms and the other 55 fathoms, so that two shackles will not be on the wildcat at the same time. The anchors shall then be hove up together at 6 fathoms per min. and the results noted. The single-anchor test shall be made twice, using a different windlass each time (also a different motor in case of electric windlasses). The two-anchor test shall also be made twice, using different pairs of windlasses for the two tests (using both motors in case of electric windlasses). The deep-sea tests shall be carried out under such conditions as will not cause fouling of the anchors and chains. All useful data shall be taken during both series of tests, such as speed of hoisting and lowering, temperature of bearings and worm gearing, etc. For electric windlasses there should be included the horsepower developed by the motors, and speed of motors. If there is only one wildcat the windlass test shall be conducted by hoisting and lowering one anchor instead of two." For tests of anchor engines when placed on board see p. 1432.

Trials for Naval Auxiliary Vessels are not carried out in such an elaborate manner as described herein. The trial generally consists of a combined full-power and fuel-oil consumption run of 24 or 48 hours duration. Table 4 is the result of tabulating data of the trials of 10 Naval Colliers, Proc. A. S. N. E., August, 1915.

Trials for Merchant Vessels generally consist of trials similar to those conducted on Naval Auxiliary Vessels, although generally of shorter duration. The paucity of merchant vessel trial data is regrettable, especially to the designing engineer; the only remedy for this situation is to so extend merchant vessel trials as to include the observation of more complete steaming data than is the present practice. This procedure would undoubtedly result in more economical designs; economy of machinery operation might, all other factors being equal, become the controlling factor in the financial return from a vessel's service. Figs. 2 and 3 show log sheets used on merchant vessel trials by the **Newport News Shipbuilding and Dry Dock Co.**

Commissioning, Post Repair and Laying Up Trials

Trials required in the U. S. Naval Service in the case of **old vessels** are (1) **commissioning trial,** which is held on a vessel that has been out of commission, in reserve, or otherwise not in active service; (2) **post repair trial,** which is held after extensive repair to machinery to prove the efficiency and sufficiency of such repairs; and (3) **laying up trial,** which is held before a vessel is placed out of active service to demonstrate the actual condition of machinery and to determine the extent of repairs and alterations necessary to improve its efficiency or to restore it to a condition fit for further service.

Commissioning Trial. The object of this trial is to determine the exact condition of the machinery of a vessel upon going into commission as to whether the machinery has received proper care and attention while out of commission; what power can be developed and maintained by the engines and boilers after extensive repairs or alterations have been made and to test the efficiency and sufficiency of such repairs and alterations; what speed the ship should be expected to attain at full power under service conditions during her commission. A standardization trial (see p. 1436) should be held prior to other trials and the general procedure and observations are the same as in acceptance trials, p. 1435.

A Post Repair Trial is made whenever the machinery of a vessel, while she has been continued in commission, has undergone extensive overhauling, repair or alteration materially affecting the power or capabilities of the vessel or machinery. The object of the trial is to ascertain if the work has been completely and efficiently performed and to determine the power and capabilities of machinery when extensive changes have been made. The procedure and observations depend upon the character of the work but are in general the same as in acceptance trials.

A Laying Up Trial is held before a ship proceeds to a ship yard for extensive overhauling, repair or alteration. A full power run is made to demonstrate the actual condition of the machinery and the character of the defects, and to afford a means of judging the full extent of repairs and alterations necessary to improve its efficiency and to restore it to a condition fit for further service.

SOCIETIES, ASSOCIATIONS AND COMMISSIONS PROMULGATING SPECIFICATIONS FOR ENGINEERING MATERIALS

BY

KARL D. WILLIAMS

There are a number of societies, associations and commissions which have various functions utilized by Marine Engineers. In the course of planning and executing Marine Engineering work it is often necessary to consult and conform to standards, rules and regulations promulgated and enforced by many of the above. For the benefit of such Marine Engineers the following has been compiled as an index when special information or specifications are required.

NAMES AND ADDRESSES OF SOCIETIES

(*a*) Steamboat-inspection Service, Department of Commerce, Washington, D. C.

(*b*) Bureau of Standards, Department of Commerce, Washington, D. C.

(*c*) American Bureau of Shipping, 66 Beaver Street, New York, John W. Coutillion, Sec'y and Treas.

(*d*) American Society for Testing Materials, C. L. Warwick, Asst. Sec'y, 1315 Spruce St., Phila., Pa.

(*e*) American Society of Mechanical Engineers, W. E. Bullocks, Asst Sec'y, 29 West Thirty-ninth St., New York, N. Y.

(*f*) Bureau of Explosives, Underwood Building, 30 Vesey St., New York, N. Y., J. E. Fairbanks Sec'y and Treas.

(*g*) American Institute of Electrical Engineers, 33 Vesey St., New York, N. Y., F. L. Hutchinson, Sec'y.

(*h*) Master Car Builders' Association, 746 Transportation Bldg., Chicago, Ill., V. R. Hawthorne, Acting Sec'y.

(*i*) American Railway Engineering Association, Room 1011, 910 Michigan Avenue, Chicago, Ill., E. H. Fritch, Sec'y.

(*j*) American Institute of Weights and Measures, 20 Vesey St., New York, N. Y., Frederick A. Halsey, Commissioner and Secretary.

(*k*) American Society of Naval Engineers, Navy Department, Washington, D. C., Frank W. Sterling, Sec'y and Treas.

STEAMBOAT-INSPECTION SERVICE

The Steamboat-inspection Service is charged with the duty of inspecting vessels, the licensing of the officers and crew of vessels and the administration of the laws relative to such vessels and their officers for the protection of life and property.

The Supervising Inspector General and the Supervising Inspectors constitute a Board that meets annually at Washington, and establishes regulations for carrying out the provisions of the Steamboat-inspection laws.

Publications

1. **Laws Governing the Steamboat-inspection Service**
2. **General Rules and Regulations,** prescribed by the Board of Supervising Inspectors:
   (*a*) Ocean and Coastwise.
   (*b*) Great Lakes.
   (*c*) Lakes other than the Great Lakes, Bays and Sounds.
   (*d*) Rivers.
3. **Pilot Rules:**
   (*a*) Inland Waters of the Atlantic and Pacific Coasts and the Coast of the Gulf of Mexico.
   (*b*) Western Rivers and the Red River of the North.
   (*c*) Great Lakes.

Legal Status

The act of Congress approved July 7, 1838, which provided for the inspection of the hulls and boilers of steam vessels, was the first legislation upon the important question of "the better security of lives of passengers on board of vessels propelled in whole or in part by steam." By this act the inspectors were appointed by the district judge of the United States for the district in which the vessel was inspected, and were paid by the steamer the sum of $5.00 each for each inspection. This act provided for lifeboats, signal lights, fire pumps and hose, and also for iron rods or chains in place of rope for steering gear. The boilers were inspected every six months and the hulls every twelve months, and it was also required that steamers employ "a competent number of experienced and skillful engineers." Certificates of inspection were required "to be posted up and kept in some conspicuous part of the boat for the information of the public."

This act was modified by the act approved March 3, 1843, which provided for additional steering apparatus, but exempted certain vessels from the requirements of the previous act. The act of Congress approved March 3, 1849, provided for signal lights for all vessels.

The act of Congress approved August 30, 1852, was really the establishment of the present Steamboat-inspection Service, and was known as the "steamboat act." This act amended the act of 1838, and provided for the appointment and payment of nine supervising inspectors, and for local inspectors at various ports. The act also provided for lifeboats, life-preservers, and other life-saving equipment, and for the licensing of engineers and pilots for passenger steamers, and for the stamping of boiler plate, and is really the foundation upon which the whole superstructure of the Steamboat-inspection Service has been raised.

This act was modified and amended in minor particulars by the various acts of March 3, 1853, April 29, 1864, June 8, 1864, March 3, 1865, and July 25, 1866.

The act of Congress approved February 28, 1871, really promulgated the present law upon the organization and administration of the Steamboat-inspection Service, and although amended and changed from time to time it is, in fact, the basis of the present practice. This act provided for the appointment of a Supervising Inspector General and created the Board of Supervising Inspectors, designating also the time and place of meeting of said Board and defining the duties thereof. It provided for the inspection, testing, and stamping of material entering into the construction of marine boilers, a provision which has proven of the highest value and importance in the prevention of explosion and loss of life.

The bill upon which this legislation was based was prepared with much care, and its enactment was, without doubt, the wisest legislation ever devised upon these lines. It has served well the purposes for which it was designed and has fully justified the highest expectations entertained by those who supported it.

Legal Authority to Establish Standards or Conduct Inspections

Under the provisions of Section 4405, R. S., the Supervising Inspectors and the Supervising Inspector General are required to assemble as a Board once a year in the City of Washington, District of Columbia, on the third Wednesday in January, and at such other times as the Secretary of Commerce shall prescribe, for joint consultation, etc., and that Board is required to establish all necessary regulations required to carry out in the most effective manner the provisions of Title 52.

Under the authority of Sections 4417 and 4418, R. S., the hulls and equipment and boilers of vessels with appurtenances are required to be inspected by the Boards of Local Inspectors.

Section 4438, R. S., gives Boards of Local Inspectors the authority to license and classify officers of vessels.

Section 4450, R. S., gives the Boards of Local Inspectors authority to investigate all acts of incompetency or misconduct committed by any licensed officer while acting under the authority of his license.

BUREAU OF STANDARDS

The National Bureau of Standards was established by Congress to have the custody of standards, comparison of standards, preparation of standards, and the solution of problems which arise in connection with standards.

The standards with which the Bureau is authorized to deal may be conveniently classed as follows: Standards of measurements, standards of values of constants, standards of quality, standards of mechanical performance, and standards of practice.

Publications

The Bureau of Standards publishes four series of publications:

(*a*) **Bulletins** which comprise the separate scientific papers covering researches of the Bureau within its field.

(*b*) **Technologic papers.**

(*c*) **Circulars** covering information of a technical or administrative character.

(*d*) **Miscellaneous publications.**

Legal Status

The Organic Act of March 3, 1901, established the Bureau of Standards by stating that "the office of standard weights and measures shall hereafter be known as the National Bureau of Standards," and the same act expanded its functions to accord with the modern needs for standardization.

Successive acts such as the Legislative Appropriation Act, the Sundry Civil Act, and others have provided the necessary authority and funds for specific investigations.

Legal authority to establish standards and specifications as noted above was established by an act of Congress and, supplemented by Section 3 of the Organic Act, authorized the Bureau to exercise its functions for the federal, state, or municipal, governments, societies, institutions, firms, corporations, or individuals.

AMERICAN BUREAU OF SHIPPING

Quoting from the Constitution and By-Laws:

"The **objects** of the Bureau are as shown by the charter, and for the purpose of carrying out these objects the Bureau is to provide for shipowners, shipbuilders, underwriters, shippers, and all interested in maritime commerce, a faithful and accurate classification and registry of mercantile shipping, and to aid and develop the Merchant Marine of the United States of America."

The Bureau maintains a **corps of inspectors** whose duty is to inspect material both in the process of manufacture and fabrication and to inspect completed vessels and machinery; classification certificates are issued on the findings of these inspections.

Publications

The Bureau publishes annually a "**Record of American and Foreign Shipping.**" This "Record" for the year 1918 has more than 1100 pages, giving the following information:

List of Offices, where situated, etc.

List of Officers, Managers, Committees, Counsel, etc.

List of Surveyors and Ports. In this connection it may be said that there are Surveyors throughout the United States and in Foreign Ports all over the world.

List of Subscribers to the Record.

List of Vessels, including signal letters, name, rigging, nation, hailing port, registered dimensions, tonnage, when and by whom built, owners, engine and boilers, and memoranda as to classification, surveys, damage surveys, etc.

List of Owners and vessles owned by same.

List of Shipbuilders, Dry Docks and Marine Railways and Constructors of Marine Machinery and boilers throughout the world.

The Bureau also publishes semi-monthly or as often as needed, supplements to the "Record," giving particulars of new vessels and changes to old ones.

The Committees referred to above under "List of Officers" are: Executive, Finance and Audit, Naval Architecture, Engineering and Classification.

Legal Status

The Bureau was incorporated under the laws of the State of New York, April 22, 1862, under the Title of "American Shipmasters' Association," and on September 26, 1898, the title was changed by legal authority to "American Bureau of Shipping."

The Bureau under its charter, Constitution and By-Laws examines the plans of vessels submitted for classification, approving same or making suggestions toward complying with the Rules governing the construction; and finally gives a certificate of class and of machinery, and when required assigns the least proper freeboard that the vessel may not be overloaded.

AMERICAN SOCIETY FOR TESTING MATERIALS

The International Association for Testing Materials was formed at a conference of workers in experimental engineering held in Munich in 1882. Larger and more important meetings have subsequently been held and the membership now extends to thirty-one countries.

It was decided at the Stockholm Congress (1897) to encourage the consolidation of the membership in various countries into separate national organisations. In pursuance of this action the American members met in Philadelphia on June 16, 1898, and organized under the name of the "American Section of the International Association for Testing Materials."

In 1902 the Association obtained a charter and was incorporated under the laws of the State of Pennsylvania under the name of the "American Society for Testing Materials." The American Society for Testing Materials is represented on the International Council by one member. The membership of the Society in 1918 was 2261.

Purpose

One of the most important functions of the Society as stated in its charter, is the **standardization of specifications and the methods of testing.** In pursuance of this activity the Society has adopted 128 standards covering ferrous and non-ferrous metals, cement, lime, gypsum and clay products, and certain miscellaneous materials such as preservative coatings, lubricants, road materials, rubber materials, rubber products, textile materials, etc.

The Society has also published 49 tentative standards which is the term applied to a proposed standard which is printed in the Proceedings for one or more years with a view of eliciting criticism before final action is taken toward their adoption as standards.

Publications

"A.S.T.M. Standards" are published triennially; the next publication will appear in 1921. From 1910 to 1916 the standards were published annually; from 1916 to June, 1918, when publication was put on the present basis, they were published biennially.

In addition to the publications of its standards the Society issued annually **Proceedings** of about 1200 pages in two parts; Part I contains the committee reports and tentative standards, and Part II the technical papers presented at the annual meeting.

Each of the standards and tentative standards of the Society are available in separate pamphlet form. Any of the above publications may be purchased upon application to the Secretary.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

The American Society of Mechanical Engineers was founded in 1880 "to promote the arts and sciences connected with Engineering and Mechanical Construction." The Society endeavors to accomplish this object through its meetings, general and section, held periodically through the United States; its publications—The Journal and Transactions—which are distributed free to its members, to libraries in exchange and to others by subscription; its large numbers of committees, working to develop the Society's usefulness in the mechanical engineering profession; and its library, located in New York City, but available to engineers throughout the country through the medium of the Library Service Bureau, a special library mail service.

Publications

The Journal of the Society is published monthly. Features of the Journal are addresses, papers and discussions presented at general meetings of the Society, addresses, papers or discussions presented at Local Sections, and reviews of technical books by experts selected by the Publication Committee, etc.

The Transactions of the Society is published about July of each year and contains the more important addresses, papers and reports that have been presented at local section meetings during the preceding year.

The Year Book of the Society is published in February of each year and contains a list of its members, the Charter, Constitution, By-Laws and Rules of the Society, etc.

Legal Status

The Society was organized as a corporation under the laws of the State of New York, April 7, 1880, its office being located in the City of New York.

C 56 of its constitution reads as follows: "The Society shall not approve

any standard or formula, or approve any engineering or commercial enterprise. It shall not allow its imprint or name to be used in any commercial work or business."

It maintains a **"Committee on Standardization"** to formulate and revise standards of principal methods of procedure in Engineering practice. There are upward of fifty standards covering various engineering materials already created and these have been widely adopted and have become the basis of extensive manufactures.

BUREAU OF EXPLOSIVES

In April, 1905, *Mr. James McCrea*, later President of the Pennsylvania Railroad, advocated before the American Railway Association the appointment of a committee to prepare regulations to govern the safe transportation of explosives.

This committee was appointed and with the aid of civilian experts in explosives, and consulting representatives of the Army and Bureau of Ordnance, Navy Department, prepared regulations, and the same were approved by the Association in October, 1905. No marked change resulted in the transportation of explosives until the Association in October, 1906, decided that some central agent of all roads was necessary for uniform enforcement of the regulations and a constitution and by-laws for the **Bureau for the Safe Transportation of Explosives and Other Dangerous Articles** were adopted.

In the following year the office of the **Chief Inspector** was opened in New York City. Seventy-eight companies at that time were members. The present membership comprises 244 railway companies.

At the present time the personnel of the Bureau consists of the Chief Inspector and office force, 23 Local Inspectors, 2 Special Agents, 3 Assistant Local Inspectors, a Chemist, an Assistant Chemist, and a Laboratory Assistant. The duties of the Inspectors are to investigate conditions at manufacturing plants, etc., and make reports to the designated operating officials in their districts and to the Chief Inspector.

The Federal law of May 30, 1908, now codified by the Act of March 4, 1909, as Sections 232 to 236, requires the Interstate Commerce Commission to formulate and the carriers to enforce reasonable regulations to promote safety.

The Bureau of Explosives submitted to the Interstate Commerce Commission a set of regulations for their consideration and with but a few changes these were adopted and were prescribed to take effect October 1, 1908.

The Bureau has prepared specifications for containers for the transportation of gases both liquefied and non-liquefied, and files the history and records of their manufacture and test as well as data on containers for the transportation of other dangerous articles.

The cost of maintaining the Bureau is shared by its members, an assessment being levied each six months by the Executive Committee.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Quoting an extract from its constitution the purpose of the Institute is stated as follows:

"Its **objects** shall be the advancement of the theory and practice of Electrical Engineering and of the allied Arts and Sciences and the maintenance of a high professional standing among its members. Among the means to this end shall be the holding of meetings for the reading and discussion of professional papers and the publication of such papers, discussions and communications as may seem expedient."

Publications

The Proceedings are issued at the beginning of each calendar month and have two principal divisions: The first part contains matters of transient interest, such as announcements, election of members, personal notes, reports of the meetings of Sections and Branches, and general information pertaining to the Institute. Technical papers and other matters of an engineering character are contained in the second part.

The Transactions are published in one or more bound volumes each year, and consist of technical papers and discussions, reports, and other matter, previously published in the monthly Proceedings, deemed worthy of permanent record.

Legal Status

The Institute was incorporated on March 16, 1896, under the Membership Corporations Law, Section 5, of the laws of the State of New York.

The Institute does not conduct inspections of any kind. It has established electrical standards, relating to the rating and testing of electrical equipment, but not including specifications of design and dimensions. These standards have been revised from year to year as a result of changes and developments in the electrical art. The Institute is also affiliated with the International Electrotechnical Commission and its standards conform to those adopted by the Commission. These standards, known as the **Standardization Rules of the A.I.E.E.**, are universally accepted as official by engineers throughout the country. The Rules were not formulated under any legal authority, but as one of the proper functions of the national organization representing the electrical profession.

MASTER CAR BUILDERS' ASSOCIATION

The Master Car Builders' Association was originally composed of master car builders and foremen of railway car shops, and was organized at Altoona, Pennsylvania, September, 1867. In June, 1881, an amendment to the constitution was proposed, and in October, 1882, this amendment creating a new class of members called "representative members" was adopted. This amendment permitted each railroad to be represented in direct proportion to the number of cars it owned. The number of cars represented in the Association is approximately 2,800,000. Its membership extends to England, France, Russia, India, Australia, Japan, China, Argentina, Chile, Brazil, Cuba and the Philippines.

Purpose

Quoting from information furnished to the Railroad Administration by the Association's late secretary, Mr. Joseph W. Taylor, the purpose of the Association is as follows:

"The objects of the Association are the advancement of knowledge concerning the construction, maintenance and service of railroad cars and the parts thereof, by investigations through committees and discussions in conventions; to provide an organization through which the members and the companies they represent may agree upon such joint action as may be required to bring about uniformity and interchangeability in the parts of railroad cars; to improve their construction and to adjust the mutual interest growing out of their interchange and repair, but the action of the Association shall have a recommendatory character, and shall not be binding upon any of its members or the companies represented in it."

This Association was responsible for the adoption of automatic couplers for cars in 1887 and the automatic air brake in 1888. Through their efforts also standards for 62 different parts of cars have been adopted.

The standardisation of parts of cars and method of loading same is of the greatest value when it is considered how often cars are shifted from one road to another and many repairs are necessarily performed while a great distance from their home shops.

AMERICAN RAILWAY ENGINEERING ASSOCIATION

Purpose

The object of the American Railway Engineering Association is the advancement of knowledge pertaining to the scientific and economic location, construction, operation and maintenance of railways.

Publications

The American Railway Engineering Association publishes:

The Monthly Bulletin, issued ten times per year (May and June omitted).

The Annual Proceedings, issued annually.

The Manual for Railway Engineering and Maintenance of Way Work, issued at intervals of from three to five years. Supplements thereto are issued annually.

Legal Status

The American Railway Engineering Association is incorporated under the laws of the State of Illinois "not for profit."

Legal authority to establish standards or conduct inspections: The recommendations of the Association are not binding upon the members of companies with which it is connected, but the standards are recommendatory.

The Association does not make "inspections" but has made and is continually making experimental tests of various kinds, the results of which are published for the information and benefit of the members of the organisation and railroad companies generally. Such tests cover:

The effect of moving loads on railroad bridges.
Tests on columns (bridge).
Tests of rail joints.
Tests of water columns.
Strength of bridge timbers, etc.

The Master Car Builders' Association and the American Railway Engineering Association are utilised to obtain standards for rolling stock and tracking in Marine Shore Establishments.

AMERICAN INSTITUTE OF WEIGHTS AND MEASURES

Purpose

Quoting from the constitution of the above Institute adopted June 28, 1916, and amended February 17 and 19, 1917, its **object is:**

"(*a*) The maintenance and improvement of our present (English) system of weights and measures, for the good of our commerce and industry and the well-being of our country.

"(*b*) The education of the people with respect to the importance of our weights and measures, through the dissemination of correct information with respect to them and to the danger inherent in changes of our basic standards of measurement.

"(c) The improvement of old and the development of additional standards as they may be needed by reason of new conditions in commerce, industry, science, and engineering.

"(d) The promotion of wise legislation for the conservation of our basic English units of weight and measure, and opposition to hasty and ill-considered legislation involving changes from our fundamental English standards."

Publications

The Institute publishes a **quarterly report of activities issued** to members alone. Two Reports on The Metric System in Export Trade and on the Weights and Measures of Latin America have been issued, as well as five bulletins on the following subjects:

Bulletin No. 1.—"The Six Metric Myths."

Bulletin No. 2.—"Endorsements that Count."

Bulletin No. 3.—"The American Metric Association and the Metric System in Munitions of War."

Bulletin No. 4.—"The Bushel Standard."

Bulletin No. 5.—"Extracts from Reports of British Parliamentary Committees Relating to the Metric System."

The Legal Status of the Institute is that of an unincorporated Society.

AMERICAN SOCIETY OF NAVAL ENGINEERS

This Society was established in 1888 when it was apparent that Marine Engineering should be accorded fuller recognition and that steps should be taken to augment Marine Engineering efficiency and promote engineering prestige. The paucity of Marine Engineering literature at this period determined the founders of this Society to issue a quarterly JOURNAL to cover this field. The founders of this Society were all Naval Engineers. The early roster of the Society contained such eminent names as Isherwood, Westinghouse, Melville, Griffin, McFarland, Manning, McKean, Baird, Mattice, Towne, Theiss, Bieg, Norton and Barton. The present Engineer-In-Chief of the Navy, Rear Admiral R. S. Griffin, U. S. Navy, was the first Secretary.

Purpose

Quoting from the by-laws of the Society the purpose of its formation is "to promote a knowledge of Marine Engineering and Naval Architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed essential to the advancement of the science."

Publication

The **Journal** of the A.S.N.E. is published quarterly. It is composed of two parts, the first being original papers in the Marine Engineering field, and the second comprising a compilation of notes of current Marine Engineering papers of the quarter.

INDEX

A. S. M. E. boiler code, 391
Absorption of gases by water, 185
Acceleration, accelerations,
  composition and resolution of, 30
  definition of, 27
  of gravity values, 14
Acetylene, 163
Achilles, U. S. S., trials, 1444
Acidity (boiler corrosion), 466
Adamson Ring furnaces, 415
Adiabatic expansion of gases, 202, 203
  of steam, 529
  of vapors, 223
Admiralty formula, powering ships, 1025
  metal, 60
After body ship lines, 1029
  burning, 948
Air banks, oil fuel burning, 302, 315
  boiler corrosion due to, 466
  chambers, fuel oil piping, 302, 315
  compressed, discharge of, 1199
    mean effective pressure, 1201
    temperature of, final, 1201
  compression of, 1200
    theory of, 206
    work of, 206, 1201
  compressors, 1201
    dimensions of, 1202
    installation of, 1203
    lubrication of, 1205, 1392
    operation and maintenance, 1204
    piping for, 1203
    testing, 1203
    types of, 1201
  control registers (*see Registers, Air Control*)
  discharge of, coeff. of, 1199
  ducts, ventilation, 1186–1189
  ejectors, 994
  engine cycles, 204
  excess, in combustion, 280, 287
    losses due to, 280
  flow through orifices, 233
    velocity of, 1166
  heat conduction of, 188
  horse power, 1166
  leakage, condenser, 974
  moist, properties of (table) 224, 225
  moisture in, 223
  pressure required for Mexican oils, 325
    static, 1165
    total or impact, 1165
    velocity, 1165
  pump (*see Pump, Air*)
  quantity required for ventilation, 1182
  required for combustion, 480
  specific heat of, mean, 1199
    weight of, 1198
  temperature on shipboard, 1182
  thermal conductivity of, 191
  velocity due to pressure, 1165
    in ventilating main, 1184
    of supply to ship, 1182
  volume per person on shipboard, 1182
  weight of (table) 1198
Ajax plastic bronze, 74
Alkalinity of boiler water, 473
Allen dense air refrigerating machine, 288
Allis-Chalmers bearing, 662
  blading, 662
  coupling, 610
Alloy, alloys,
  aluminum, 63, 67
  anti-friction (*see Bearing Metals*)
  bearing-metal (*see Bearing Metals*)
  copper-nickel, 66
  copper-tin, properties of (table), 54
  copper-zinc, properties of (table), 57
  for die casting, 77
  for metallic packing, 78
  fusible (*see Fusible Alloys*)
  melting points of (table), 55
  nickel-copper, 66
  non-ferrous, 41–78
  -steel forgings, 135, 142
  steels, 119
  strength of, at high temperatures, 68
  white metal, 78
Almy boilers, 365–367
Alquist reduction gear, 747
Aluminum, 52
  alloys, 63, 67
  brass, 63
  bronze, 63
  -copper alloy, 67
  extruded, 53
  grades of, 53
  impurities in, 53
  marine use for, 53
  pipe, 1260
  properties of, 52
  U. S. Navy specifications, 53
  -zinc alloys, 67
American Bureau of Shipping, 390, 1152
  Institute of Electrical Engineers, 1455
    of Weights and Measures, 1457
  process, malleable iron, 117
  Railway Engineering Ass'n., 1457
  Society for Testing Materials, 1453
    of Mechanical Engineers, 1454
    of Naval Engineers, 1458
Ammonia, 218, 220
  Mollier diagram for, 219
  properties of (table) 218
  refrigeration produced by, 229, 230
  solubility in water, 186
  superheated, equation for, 219
    properties of (table), 220
  vapor, 218

Amorphous graphite, 1385
metal, 98
Analysis of coal, 241
of gases of combustion, 481
of Sea Water, 466
Analytical method of balancing recipro. eng., 921
Anchor engine (*see Windlasses*), 1230–1234
test of, 1433, 1448
Anti-friction alloys, 68, 78 (*see Bearing Metals*)
Antimony, analysis of, 51
-lead-tin bearing alloys, 76
Aqua ammonia, 186
Arbitration bar (cast iron testing), 152
Area equivalents (conversion table), 15
Armature heat transmission from, 188
Arsenic bronze, 73
Asbestos board, 299
Ash and fire doors, 453
ejector, 1194
handling machinery, 1192
hoe, 452
hoist, electric, 1193
ram type, 1194
steam type, 1192
in coal, extraneous, 242
inherent, 242
Aspinwall governor, 912
Astern turbines, 562, 572
Atomization of oil fuel, 281
for marine purposes, 281
Atomizer, atomizers, 281
angle of spray in, 283
Bureau S. E. type 1, 285
capacity-pressure curves for various, 283–286
Compressed air, 282
for marine purposes, 281
increasing capacity of, 301
increasing number of, 301
oil cone in, 283
Peabody, 286
pressure-capacity curves for various, 283–286
pressure or mechanical, 282, 285
proportions of oil pressure in, 284
Schutte & Koerting, 286
steam, 281
Attached pumps, 906
Augmentera, vacuum, 1000
Parsons, 1000
Radojet, 1000
Austenite, 85
Axial clearance, turbine, 554

Babbitt metal, metals, 75
genuine (hard and soft), 75
German, 72
lead-tin-antimony, 76
S. A. E. specifications, 75
Souther, 72
tin-base, 75
U. S. Navy specifications, 75
Babcock & Wilcox boiler, 338, 367–375
oil burning, test of, 500–505
water testing kit, 473
Babcock's formula (steam flow), 236
Backing power of recipro. eng., 839
trials, 1442
Baffles, boiler, 377, 441
condenser, 982
evaporator, 1085
Balance piston, reciprocating engines, 8
turbine, 632
Balancing, machine, Akimoff, 640
Carwin, 644
Cleary, 641
Vibration Specialty Co., 640
reciprocating engines, 920
analytical method, 921
graphical method, 926
Barlow's formula, pipe strength, 1262
Barometric readings, corr. for gravity, 9
Barrel (turbines), 551
Baume'—specific gravity tables, 203–2
Bearing, bearings,
centrifugal pumps, 1121
cooling, 1387
die cast, 76
heat of, proper, 1387
metals, 68–78 (*see also Brasses, Bronze*
Babbits, 75
bronze, wear of, 73
classification of, 68
copper-base, 72
copper-tin, 72
copper-tin-lead, 73
copper-tin-lead-zinc, 74
lead-base, 76
lead-tin-antimony, 76
lead-tin-antimony-copper, 77
properties of, 72
S. A. E. specifications for, 75
tin-base, 75
U. S. Navy specifications, 75
wear of bronze, 73
zinc base, 77
pressure of, 1387
reciprocating eng., 864, 940
rubbing speed of, 1383
turbine,
Allis-Chalmers, 662
De Laval, 761
friction in, 563
gear and pinion, 636
General Electric, 635, 638, 748, 75
Kerr, thrust, 638, 764
Kingsbury, thrust, 637
main, 635
Newport News, 777
oil for, 638
admitted at point of least pres
of, 661
removal of hot, from, 661
Parsons, 636
pressure, safe, 636
Poole, 769
Terry, 789
thrust, 637
warm, 636
Westinghouse, 635, 739
Bed and bed plate, recipro. eng., 850
Bell metal, 56, 72
Belleck china fixtures, 1373
Bending, moment, 33
Bends, pipe, 680 (*see Pipe Bends*)
Benedict metal, 66
Bessemer converter, 90
baby, 91
bottom and side blow, 91
Bi-rotor centrifugal pumps, 1119

Bismuth solder (fusible alloy), 78
Bituminous coal, 244
Black heart malleable iron, 117
Blade, blades, turbine,
Allis-Chalmers, 622
back curve of, 615
construction, 616
Curtis, 546
De Laval, 618, 761
edges, thickness of impulse, 613
end thrust due to, 545
exit angle of, 545
friction loss in impulse, 544
gaging, 551, 570, 588
General Electric, 617, 748, 753
impulse, 543, 736
design of, 611
Kerr, 764
leakage, 533, 552, 619
lengths of, 554, 569, 619
Ljungstrom, 779
losses in, 561, 562
materials, 622
Parsons, 548
degree of reaction in, 549
Poole, 767
reaction, 620
roots, forms of, 616
Rosary construction of, 622
sections, propeller, 1049
semi-wing, 551, 620
speed of Parsons, 567
Terry, 793
thickness, propeller, 1055
tip clearance, propeller, 1033
broad, fan and narrow, propeller, 1034, 1050
vibrations in, 613
Westinghouse, 736
widths of, 555
wing, 551, 620
Blast furnace, 80
products, 81
Blister bar, 94
Block coefficient, ship, 1029
Blow and feed, evaporator, 1094, 1096
holes in ingots, 97
-off valves, boiler, 449
Body plan of ships, 1022
Boiler, boilers,
Adamson Ring furnace for, 415
Almy, 365–367
ash and fire doors, 453
Babcock & Wilcox, 338, 367–375
baffles, 377, 441
blow-down valve, 449, 1301
blow-off piping, 1254
valves, 449
bottom blow, 379, 449, 1301
braces, bars for, 321
bracing, 126
brickwork, 438
bumped heads of, 406
care of idle, 464
casings, 438
leaks in, 292
caulking of, 405
Coil, 341
combustion chamber, 431
girders, 389, 420
common combustion chamber type, 385
compound (corrosion), 468
Boiler, construction of, 390
Scotch, 426
corrosion, exterior, 464
interior, 466
design, draft, effect of, on, 344
fuel, effect of, on, 344
furnace, oil burning, 297
general, of, 343
number and size of units, effect of, on, 357
Scotch, of, 385
size and number of units, effect of, on, 357
space and weight, effect of, on, 358
steam output, effect of, on, 354
pressure, effect of, on, 352
weight and spaces, effect of, on, 358
designation of, 339
development of, 333
diagonal ligaments, 422
dished heads of, 406
double end, 340, 385, 386
drum pads, 449
dry-back, 339
dry pipes, 440
efficiency, 488
Express, 297, 341
externally fired and internally fired, 340
factor of safety in, 391
Faron's, 337
feed connections, 379
pipes, 440, 1254
pumps, 1121
water regulators, automatic, 382
fire-box, 341
fire tube, 339
fittings, external, 442, 443
internal, 440
flash, 341
flat stayed surfaces, 388, 410
flue, 340
foreign, built, 519
foundations, 438
furnace, furnaces, 415–433, 519
Adamson Ring type, 415
Bowling Hoop type, 514
Brown's cambered section, 416
Deighton type, 416
design, oil fuel, 297
Farnley type, 416
Fox type, 416
Holmes type, 416
horse-collar form, 416
Morrison type, 416
mountings, 453
number of, 387
Purves type, 416
spacing of, 388
tube roofed, 297
gage glasses and fittings, 445
cocks, 447
steam, 447
grate surface, 355, 376, 386
Gunboat, 341
handholes and manholes of, 408
haystack, 333
heat balance of, 389
heating surface, 355, 376, 386
horsepower of, 354
hydraulic tests of, 392
lagging, 439
leg, 341

Boiler, lobster back, 341, 342
locomotive, 334, 341
MacGregor Laird, 335
manholes and handholes, 408
materials of, insp. requirements, 514
mountings, 440
oil, removal of, from, 465
pipe, 342
plates, steel, 124, 516
plugs, fusible, 449
Porcupine, 341
requirements, ideal, of, 343
retarders, 377
return tubular, 340
riveted joints in, 393
shells, 388, 408, 426
rivets, finished and rods for, 125, 394, 402–405
room instruments, oil fuel burning, 315
flue gas analysis, 318
gage, draft, 315
differential, 315
pressure, 316
manometer, 315
meters, oil, 318
Bowser, 321
Empire, 318
steam flow, 322
venturi, 322
pyrometers, 316
optical, 316, 317
potentiometer type of, 316
Ringelmann's scale, 318
Smoke chart, 318
thermocouples, base metal, 316
precious metal, 316
thermometers, 316
nitrogen, 316
rules and regulations governing construction of, 390
safety, 342
valves, 447, 1324, 1326
scale, removal of, from, 487
Scotch (def.), 341
area through tubes of, 387
furnaces of, number of, 387
spacing of, 388
length of, 386
steam space for, 387
tubes for, 387
sectional, 342
separate combustion chamber type, 386
surface blow valve, 1301
setting, 438
shell, 341
shells, riveted, of cylindrical, 388, 408, 426
single end, 340, 385
sizes, 358–375
Almy, 365–367
Babcock & Wilcox, 367–375
Scotch, 359–364
water tube, 365–375
Smith's, 336
stay bolts, 432
stays, 389, 413
bars for, 521
steam outlet of, 379
Stevens, 333
Stimers, 334

Boiler, stokers, mechanical, for marine, 391
surface blow, 379, 449
test after installation, 1431
before installation, 1429
superheaters, 461
machinery trials, 1433
tests, 488–514
try cocks, 447
tube, tubes, 123, 387, 434, 522
beading and flaring, 437
expanding, 436
holes, 436
joints, 436
lapwelded, 522
plugs, 437
sheets, back, 431
combustion chamber, 421
front, 428
shell, 422, 427
sizes and weights, 435
tubular and tubulous, 340
wagon, 333
water testing, 469–474
tube, 339
weights, 358–375
Almy, 365–367
Babcock & Wilcox, 367–375
Scotch, 359–364
water tube, 365–375
welded drums of, 406
Boiling points of liquids, 185, 208
Bolinder Semi-Diesel engine, 963
Bolt, bolts,
and nuts, U. S. Standard, 161
materials for, 127–129
safe loads for U. S. Standard, 161
U. S. Standard, dimensions of, 161
weight of, 162
Bomb Calorimeter, 256
Booster pump, oil fuel, 302
Bosh, blast furnace, 80
Bottom blow, boiler, 379, 449
Bower-Barff process, protecting iron and steel, 156
Bowling Hoop furnace, 415
Bowser oil filter, 672
Braces, boiler, bars for, 321
Brass, brasses
alloy castings, Naval requirements, 57
aluminum, 63
and bronzes, 53–66
brazing, 56
cast, 56
for electrical appliances, 59
marine uses for, 57
cold working, 60
definition of, 56
free cutting, 60
heat treatment of, 61
hot working, 58
industrial, 56
leaded, 55
modified hot working, 59
pipe, 1254, 1259
plates, 59
red, 64
rich low, 58
rods, weight of (table), 61
commercial, for, 59
seamless pipe, 1259, 1260
shapes, 59

Brass, sheets, weight of (table), 44
spring, 60
tubing, 59
U. S. Navy specifications for, 56–59
wire, weight of (table), 61
wrought, 58
yellow, analyses of, 56
Brazing, fluxes for, 78
metal, 58, 65, 78
solder, 56, 78
spelter, 78
Breadth of ships, 1013
Brickwork, boiler, 438
oil burning, damage to, 329
Bridge gages, 650
Bridge walls, supports, 455–460
British Admiralty Coal, 243
British Board of Trade, 390
British Corporation, 391
British thermal unit, B.t.u., (def.), 180
mechanical equivalent of, 196
Bronze, bronzes
Ajax plastic (bearing metal), 74
aluminum, 63
arsenic, 73
castings, 55
cyprus (bearing metal), 72
Demo (bearing metal), 72
heat treatment, effect of, 56
journal, 55
manganese, 62
melting points of various, 55
Parsons (bearing metal), 74
phosphor, 65, 72
plastic (bearing metal), 72
Plumbic (bearing metal), 72
strength at various temperatures, 56
Tobin, 59, 71
Vanadium, 64
Brown's cambered section furnace, 416
Buckets, turbine (*see Blade, Blades*)
Sturtevant, 784
Terry, 788
Bulkhead flanges, 1278
Bull rings, 858
Bumped heads, boiler, 405
Bunker oil, "A," "B," and "C," 272
Bureau Veritas, 391
of Explosives, 1455
of Standards, 1452
Burners, oil fuel (*See Atomizers*)
Bursting tests, piping, 1263
Busheled Scrap, 89

Calcium chloride solutions, 184
Calking boiler, 405
tools, 405
Calorific value of coal, 242
of fuel (*See Heating Value*)
Calorimeter, 490
bomb, 256
tests, 490
Calorizing, 154, 381
Capacity equivalents, (conversion table), 16
turbine, limiting factors of, 563
Capstans, 1242
Carbon dioxide, 229
deposits, 1399
fixed in coal, 241
Carbon gland (turbine), 556, 755, 761, 764
on atomizing tips, 325
on tubes, furnace sides and bottom, 325
properties of (table), 222
refrigeration produced by, 229, 231
Carbonic acid (*See Carbon Dioxide*)
Card factor, 832, 833
Cargo oil pumps, 1149
winches, 1234
Carnot cycle, 204, 827
engine, ideal, 197
Carwen balancing machine, 644
Case carburization, 109
hardening, 94
heat treatment after, 111
Casing, casings,
boiler, 438
turbine, 625, 735, 739, 747, 753, 761, 774, 786, 789
gland, 556, 633, 755, 761, 764
centrifugal pump, 1119
Cast Iron
aluminum, influence of, 116
A. S. T. M. specifications for, 150
chemical composition, influence of, 114
chilled, 116
chromium, influence of, 116
contraction of, 117
electrical apparatus, for, 116
grades, 111
gray, 86, 111
growth of, at red heat, 116
gun, 116
malleable, 117
manganese, influence of, 116
melting processes for, 118
microstructure of, 112
mixture making for, 119
Navy specification for, 148–150
nickel, influence of, 116
oxygen, influence of, 115
phosphorous, influence of, 115
pipe, 1260
properties of, relation to microstructure of, 112
Scotch, 114
shrinkage of, 117
silvery, 114
strength of, 118
sulphur, influence of, 115
vanadium, influence of, 116
white, 111
Castings,
alloy, 57
aluminum, 68
brass, commercial, 58
brass, Naval, 56
iron, 149
malleable, 150
monel metal, 67
phosphor bronze, 65
steel, 131, 138–140, 146
vanadium bronze, 64
Castor oil, 1383
Catalizer, 109
Cavitation (propellers), 1043–1045
Celite boiler lagging, 439
Celsius thermometer scale, 177
Cementation, partial, 109
Cementation process, 94

Disc, discs, turbine,
design of, 593
General Electric, 747
Haerle's curves, strength of, 601
Kerr, 763
losses in, 561
Materials, 593
Newport News, 774
Poole, 767
stresses in, 594
Weaver, S. H., on, 596
Discharge coeff. of compressed air, 1199
Dished heads, boiler, 406
Distilled water, test for salt in, 1093
Distiller, distillers, 1096–1102
accessories for, 1097
and evaporators, 1079–1102
installation methods, 1090–1093, 1097
materials of, 1097
service of, 1096
size required, 1097
space occupied by, 1097
test before installation, 1429
type, types, 1097–1102
commercial 1097
straight tube,
cast iron shell, 1095
copper or brass shell, 1099
unit coil,
cast iron shell, 1098
steel shell, 1095
Distilling plant test after installation, 1432
Distribution stations, oil fuel, 253
Doors, ash and fire, boiler, 453
Double annealing, 101
Draft (*see also Fans, Centrifugal*)
balanced, 347
forced, 345, 347, 1164
blowers location for, 347
closed ashpit system of, 347
fireroom system of, 347
deflectors for, 347
Ellis and Eaves system of, 352
Howden system of, 349, 352
induced, 348, 1164
steam jets for, 348
influence of wind on, 344
intensity of, 344, 345
measured, 1166
at uptake, 345
in furnace, 345
natural, 344
of ships, 1014
Peabody formula for, 344
required at different rates of combustion, 346
ventilators for, 348
size and location of, 348
Drains, turbine cylinder, 625, 681, 1004
Westinghouse, 629
Drawings, electrical, 1364
Drop forgings, U. S. Navy specifications, 136
Drum losses, turbine, 561
rotors, design of, 609
Dry pipes, boiler, 440
Ducts, air, 1186–1189
Du Long's formula, heat value of coal, 242
Dummy piston, 559, 631, 735
rings, 624
Duncan, U. S. S., engine room, 795
Duplex steel process, 94
Dynamic balance, turbine rotors, 640
Dynamometer, torsion, 713

E H. P., 1403
from indicator card, 1412
from model tests, 1026
Economical speed of ships, 1014
Efficiency, boiler, 488
maximum, oil burning, 324
of centrifugal fans, 1167
of pumps, 1151
per cent. of boilers, 488
ratio, turbine, 534, 535
stage, turbine, 534
wheel, turbine, 563
Ejector, air, 994
ash (*see Ash Ejector*), 1194
Electric,
conductors, protection of, 1365
lead covered, steel armored, 1365
types of, 1366
twin, cable, 1367
conduits, 1365
current, boiler corrosion, 475
drive (ship propulsion)
general notes on, 705–707
S. S. Mjölner, 703
S. S. Wulsty Castle, 704
U. S. S. Jupiter, 693
U. S. S. New Mexico, 694–701
U. S. S. Tennessee, 701
distribution, 1366
door switches, 1373
feeders, 1366
fire alarms, 1377
fittings and fixtures, 1372
furnaces for steel, 95, 119
generating sets, 1367
heaters, 1181
illuminating methods, 1374
installation, 1363
drawings, 1364
methods, 1363
requirements, classification Societies, 1363
junction box, 1372
lamps, 1374
lighting system, 1371
motors and controllers, 1376
moulding, wooden, 1365
power system, 1375
running lights, 1373
searchlights, 1375
signalling systems, 1377
steel, chemistry of, 96
switchboards, 1368–1371
generator and feeder sections, 1371
telegraph, engine, 1377
radio-, 1378
telephones, 1377
wire sizes, 1364
Electrical apparatus, oil cargo vessels, 275
Electrolysis, (boiler corrosion), 467
Electroplating, 154
End thrust (turbines), 631
Energy, conservation of, 196
defined, 27
degradation of, 196
equivalents (table), 18
transformed of, 196
Enfoliation in steel, 110

Engine, engines,
constant volume, 946
Diesel, 945–962 (*see Diesel Engines*)
explosion, 945
internal combustion, 945–963
reciprocating, (*see Reciprocating Engines*)
room layouts, turbine drive, 794–804
Semi-Diesel, 945–962
surface-ignition, 945
Entrophy, 196
of water and steam (tables), 209
-temperature diagram, 215, 530
Epicassit process, protection of iron and steel, 154
Equilibrium, (def.) 32
general condition for, 33
diagram, iron-carbon, 84
Equivalents, tables of, 14–19
Ericsson cycle, 204
U. S. S., engine room, 806
Etching process, protection of iron and steel, 157
Ethyl chloride, 221, 222, 229
Eutectic, 83
Eutectoid, 85
Evaporator, evaporators, 1079–1096
accessories, 1085
and distillers, 1079–1102
baffles, 1085
blow and feed regulation, 1094
-down, 1096
brine and feed pump, combined, 1094
capacity, maximum, 1080
vs. volumes of, 1081
vs. weight of, 1081
design, controlled by, 1085
double effect, 1080, 1092
feed, 1096
and blow regulation, 1094
and brine pump, combined, 1094
heaters, 1098–1102
installation, 1092, 1100
sizes, 1099
space, 1100
types, 1099, 1102
weight, 1100
installation methods, 1086, 1090–1093
materials for, 1081
galvanizing, 1081
headers, steam, 1081
heating surface, 1081
shells, 1081
operation of, 1094
orifice method of operation, 1096
piping arrangements, 1093
plants, 1079, 1086
capacity and tests of, 1079
overload capacity, 1079
regenerative compressor fitted to, 1091, 1093
service of, 1079
single effect, 1080, 1090, 1091
space occupied by, 1080
test before installation 1429
types, commercial, 1080
straight tube, horizontal, 1082
submerged bronze shell, 1087
cast iron shell, 1086
partially, 1089
steel shell, 1085
U-tube, vertical 1083
weights of, 1080
Exhaust system of ventilation, 1183
Expansion, adiabatic, of gases, 203
bends, pipe, 1267
coefficients of, 178, 179; of metals, 41
joints, 681 (*see Pipe Bends*)
of bodies by heat, 178
of gases (table), 203
of pipes, 1266 (*see also Pipe Expansion*)
of saturated vapors, 223
of steam, actual, in turbines, 533
adiabatic, 529
over-, in nozzles, 540
under-, in nozzles, 540
polythropic, of gases (table), 203
tables for fuel oil, 266, 267
Explosion engines, 945
Express boiler, 341
Extinguishers, Foamite fire, 321
Extruded aluminum, 53

Factor, reheat, 535
Fans, centrifugal,
characteristic curves, 1172
classified, 1163
dampers, balanced, for, 1165
diffusor plates, 1164
efficiency of, 1167
manometric, 1168
volumetric, 1168
equivalent orifice, 1167
forced draft, 1164
blowers, 1164
heating and ventilating, 1164
impellers, types of, 1170
induced draft, 1164
laws, 1172
load, 1171
operation in parallel, 1175
orifice, equivalent, 1167
output, limits on, 1171
performance, expressed by, 1163
effect of speed, density, temp. on, 1169
resistance in parallel, 1166
in series, 1166, 1167
speed, effect on performance, 1169
temperature, effect on performance, 1169
ventilating and heating, 1164
Faron's boiler, 337
Feed and blow-down, evaporator, 1094, 1096
connections, boilers, 379, 441
pumps, boiler, 1121
system, oil leaks into, 329
tests after installation, 1431
Feed water heaters, 384, 1103–1110
accessories for, 1103
design of, 1108
economy due to, 1103
evaporator, 1098–1102
friction drop through, 1108
heat transfer rates of, 1110
installations, 1104
dead end, 1107, 1008
direct flow (thoroughfare type), 1104, 1107
materials of, 1103

Feed water heaters, operation of, 110
retarders, 1110
service of, 1103
size of, 1107
test, 1429
types of, 1103–1106
bow-tube, 1105
closed, 1103
open, 1103
straight-tube, 1106
unit coil, cast iron shell, 1104
steel shell, 1105
Feed water regulators, automatic, 382
Vigilant, 383
Fenton's alloy (bearing metal), 77
Ferrite, 84
Ferro-alloys, 119
compositions of, 120
Ferro-manganese, 91
Ferro-silicon, 114
Ferrules, condenser, 976
Filters, lubricating oil, 671
Fire alarms, 1377
doors, boiler, 453
extinguishers, oil fuel burning, 327
point, fuel oil, 267
lub. oil, 1393, 1395
tools, boiler, 453
Fires, cleaning boiler, 479
on oil burning vessels, 326
Firing and fuel, 477
time, 478
Firing boilers, 477
axioms for, 478
time, 478
Flame, color of, in oil fuel burning, 287
Flange metal, 58
Flanged fittings (*see Pipe Fittings*; *Pipe Flanges*)
Flareback, oil fuel burning, 326
to prevent, 326
Flash point, fuel oil, 268
lubricating oil, 1391, 1393, 1395
Floor plates, 132
Flow of gases, 232
through orifices, 232
of steam, in pipes, 236
through nozzles, 234, 235
Flue gas analysis, 482–487, 513
oil fuel burning, 318
sampling, 482
Fluids, heat conductance of, 187, 188
Flux, blast furnace, 80
brazing, 78
soldering, 78
Foamite, Firefoam extinguisher, 327
Force, forces, centrifugal, 38
centripetal, 38
composition and resolution of, 32
defined, 27
equivalents, (conversion table), 14
parallel, 32
Forced feed lubricating oils, 1388
Forgings for hulls, engines and ordnance, 133, 135–138
machinery, 133
Föttinger hydraulic reduction gear, 707–709
Fox furnace, 416
Foundations, boiler, 438
Fractions, decimal value of, 11
Freeboard of ships determined by, 1014
Freezing mixtures, 183
points of liquids, 184
of salt solutions, 184, 185
Friction, (lubrication), 1381
drop, feed heaters, 1108
fluid, (viscosity), 1381, 1394
losses in impulse blades, 544
in turbine bearings, 563
steam, in turbines, 534
Fuel and firing, 177
-injection, Diesel engines, air spraying, 950
mechanical spraying, 949
solid injection, 950
-oil, (see also Oil Fuel) 250–276, 281–329
-oils for Diesel engines, 963
chemical properties, 963
general physical requirements, 963
Fullager Diesel engine, 961
Funnel area, 378
Furnace, boiler, 415–443, 519
electric, for steel, 95, 119
insulating materials for, 439
insulating of, 297, 439
oil-burning boiler, design, 297
types of, 297
volume, oil fuel burning, 299
Fusible alloys, 78
melting points of, 183
plugs, 449
Fusion, heat of, 185

Gage, gages, bridge, 650
dial spring, 968
differential, 315
draft, 316
oil pressure, turbine, 663
pressure, 316
absolute, 968
Gaging blades, 551, 570, 588
per cent. of, 551
Gallon, Imperial (def.), 14
U. S. (def.), 14
Gangue, 80
Gary-Cummings torsion meter, 1420
Gas, gases
constants, values of, 201
laws of, 200
mixtures, 201
perfect, 200
characteristic equation of, 195
ideal cycles with, 204
properties of, 201
solubility in water, 185, 186
specific heats of, 200
weights and volumes of, 200
thermodynamic equations of, 202
Gaskets, exhaust pipe, 681
oil pipe, 679
steam pipe, 680
Gasoline, commercial, 277, 278
inspection of, 277
specifications for motor, 277
specific gravity of, 277
test of, 277
value of, as fuel, 316
Gauge, gauges (*see Gage, Gages*)
Gear, reduction (*see Reduction Gear*)

Geared turbines, 528
General Electric bearings, turbine, 635, 638
blading, 617
coupling, shaft, 610
governor, emergency, 654
shroud rings, 619
turbines, 745–755
electric drive, for, 751
high pressure, 750
low pressure, 750
small, 753
trials of, 498
valve, emergency, 657
maneuvering, 659
Generating sets, 1367
connections to, steam, 1370
switch boards, 1371
Generator, U. S. S. New Mexico, 697
German silver, 66
Girod electric furnace, 96
Gland, carbon, 556, 755, 789
diaphragm, 557
piping, turbine, 682
shaft, clearance, turbine, 555
water sealed, 558, 736, 744
Governors, centrifugal pumps, 1128, 1129
pump (see Pump Governors)
turbine, 653–660
Aspinwall, 912
De Laval, 757, 762
electric drive, for, 655, 739
General Electric, 654, 745
Ljungstrom, 781
New Mexico, U. S. S., 696
Newport News, 778
overspeed, 655, 764, 786
Sturtevant, 786
Terry, 789, 794
Westinghouse, 653, 736, 739, 740, 744
Grain growth of low carbon steel, 103
Graphical method of balancing recipro. eng., 926
Graphite, 1385
Grashof's formula, mean temp. diff., 973
steam flow, 233
Grate bar, grate bars, 455–460
air space in, 458
bearers, 453
double and quadruple, 458
Scotch boilers, 457
surface, boiler, 355, 377, 486
Gravity, accleration of (table), 14
centers of, 33–36
feed, lub. oils, 1388, 1390
lubricating system, turbine, 663
Gray cast iron series, 86
Greases, 1384
lubricating value of, 1385
Guide vanes, steam flow through, coeffs. for, 235
Gun iron, 116
compositions and properties, 54
Gyration, radius of (def.), 36

Haerle's curves (turbine discs), 601
Hand wheels, valve, 1301, 1323
Heat, 177–237
available (turbines), 531
balance, boiler, 489, 503
Heat, conductance of fluids, 187, 188
conduction, conversion table for, 19
through pipe coverings, 193
through steam pipes, 193
content—entrophy diagram, 200
conversion table for, 18
-engine cycles, 204
expansion by, 178
insulation, application of, 1358
commercial types, 1358
sizes, 1358
conductivity of, 190
economic thickness of, 1356
losses through, 1350
savings from, 1350
specifications, 1359
latent, 186
losses from bare metal surfaces, 1348
from flat iron surfaces, 1349
from iron pipe, 1349
through 85 per cent. magnesia covering, 1351
mechanical equivalent of, 195, 196
of fusion (table), 185
of vaporization (table), 195
resistance of fluids, 187
specific (*see Specific Heat*)
thermal conductivity, 189
transmission, 186
between plate separated fluids, 191
between steam and boiling liquids, 192
by convection and conduction, 187
by radiation, 194
coefficient of, 192
conversion tables for, 19
from steam to air, 193
from steam to water, 188
in condensers, 192, 973
transfer, feed heaters, 1110
treatment of carbon steel, 101
of carbon steel castings, 102
of machinery steel, 101
of rolled and forged objects, 101
units (def.), 180
value of coal, calometric determination of, 242
Du Long's formula for, 242
oil fuels, 256
Heaters, electric, 1181
feed water (*see Feed Water Heaters*)
oil fuel, 302, 307–309
capacity of, 307
charcoal fired, 308
Cramp type, 308
Reilly multicoil type, 309
Schutte & Koerting type, 309
Heating, 1177
and ventilation, 1177, 1189
combined, 1189
fans, 1164
Siroccosone system, 1191
osone production in, 1192
power consumption for, 1192
temperature control in, 1192
Thermofan system, 1189
exhaust operation, 1191
performance curves for, 1191
plenum operation, 1189
circuits, 1180

Heating, manifolds, 1180
Navy specifications, 1180
Newport News practice, 1178
piping, 1178
systems, 1181
through wooden decks, 1178
radiator, 1177, 1180
pipes, 1180
surface, oil-burning boiler, 299
radiator, 1177
Thermo-tank, 1180
Hector, U. S. S. trials, 1446
Hele Shaw pump, 1214
Hematite, 79
Hempel flue gas analysis apparatus, 483
Heroult electric furnace, 95
Hirn's analysis of steam engine, 227
Hopkinson-Thring torsion meter, 1423
Horse power (def.), 1403
effective and indicated, 1403
-hour, B.t.u. equivalent of, 196
measuring, 1403
of boilers, 354
shaft, 1403
reduced to I. H. P., 1404
Hot-air engines, cycles of, 204
Hoyt metal, 47
Hubs, propeller, 1064–1076
Huddling chamber, safety valve, 1324
Hull appendages, 1034
coefficients, 1029
Humidity, measurement of, 223
relative, 224
Hydraulic reduction gear, 707–709
Hydrokineters, 441
Hydrometer, Tagliabue, 262

I. H. P., 1403
from indicator card, 1412
from manograph, 1416
of Diesel engines, 1416
Ice machine oil, 1392, 1396
Idalia tests with superheat, 513
Illumination, methods of, 1374
Impellers, fan, 1170
pump, 1117
Impulse blading, 543, 736
design of simple, turbine, 573
turbines, 526
provisional proportions of, 564
Indicator, steam engine, 1404
card, cards, 1405
calculating I. H. P. from, 1412
combined, 1410
data required with, 1408
deductions from, 1409
defective engine, 1409
ideal, 1409
planimeters for, 1413
steam consumption from, 1412
taking, 1407
care of, 1409
commercial, 1409
cord, 1406
drum enertia and friction, 1406
errors, 1406
hook, 1406
inside spring, 1404
outside spring, 1404
Indicator, reducing motions, 1406
wheels, 1407
requirements of good, 1405
springs, 1405
calibration of, 1406
use of, 1403
Induction electric furnace, 96
Injection valve, main, 1297, 1319
Instantaneous centers, 31
Instruments, boiler room, 315
Insulbrix, 439
Integrals, definite, 25
fundamental theorem, 25
indefinite, 23
properties of, 25
Integration, methods of, 26
Internal combustion engines, 945–963
cycles of, 205, 206
Invar, 105
Iron and Steel, (*see also Steel, Steels*)
composition of slag, 88
metallography of, 82
outline of constitution of, 87
purification processes, 87
testing, 121–153
castings, 149
grey, 152
malleable, 150
ores, 79
Bessemer and Non-bessemer, 80
extraction of iron from, 80
smelting, 71
pyrites in coal, 242
wrought, (*see Wrought Iron*)
Irrational functions, indefinite integrals, 24
Isothemial expansion of vapors, 223
of gases, 202

Jason, U. S. S. trials, 1456
Joule cycle, 206
-Thompson effect (throttling), 237
Junkers Diesel Engine, 960
Jupiter, U. S. S. electric drive machinery, 639
test data, 721
Jet condensers, 983
design, 987
injection, water for, 986
shell, 987
size, 987
weight of, 988

Keel condensers, 988
Kerr thrust bearing, 638
turbines, 762–767
turning gear, 647
valve gear, emergency, 658
maneuvering, 659
Kilogram-calorie (def.), 180
Kinematics, 29–32
Kinetic energy (def.), 27
Kinetics, 38
Kingsbury thrust bearing, 637
Kinney rotary plunger pump, 894
Kish, 112
Kjellin electric furnace, 96
Knobbled charcoal iron, 89

Labyrinth packing, 556, 633, 761
diaphragm, 634
Lagging, (*See Pipe Covering and Lagging*)
turbine, 748
Lamps, electric, 1374
Lard oil, 1383
Latent heat of vaporization (table), 185
Law of cosine, 21
of sine, 21
Lead castings, shrinkage of, 46
grades of, 46
metallurgy of, 46
pig, composition of, 46
specifications for, 46
pipe, 1261
properties of, 46
sheet, max. sizes of, 47
tin-antimony bearing alloys, 76
tin-copper bearing alloys, 73
tin-copper-zinc bearing alloys, 74
tubing, 1261
Leakage, blade, 533, 552
gland, 555
Leaks in boiler casings, 292
Le Blanc air pump, 994, 996
Ledeburite, 84
Length equivalents (table), 15
of ships, 1013
Leslie reducing valve, 1000
Levers, principle of, 32
Lifting gear, turbine, 687
Lighting system voltage drop, 1364
circuits, 1371
fittings and fixtures, 1372
Lignum vitæ, 871, 941
lubrication of, 871
Limonite, 79
Linked up (valve gear), 877
Lipowitz's metal (fusible alloy), 78
Liquation in steel, 110
Liquid, liquids,
boiling points of, 208
expansion of, 178
freezing points of, 184
fuel, 250–276
thermal conductivities of, 191
Liquidus, 84
Liter, (def.), 14
Ljungstrom turbine and generator arrangement, 703, 781
Lloyd's Register of Shipping, 392
Load factor, 1055
water line of ships, 1022
Logarithms, common, table of, 2
Lovekin foundation liners, 686
pipe joint, 680
Lubricants,
carbon deposits from, 1399
classification of, 1381
graphite, 1385
greases, 1384
internal or fluid friction of, 1381, 1384
selecting, 1386
ultimate value of, 1386
Lubricating oil, bearings, turbine, 638, 661
filters, 671
piping, 679, 1254
pumps for turbines, 667, 744, 755, 790, 793
strainers, 669
Lubricating oil, systems, turbine, 661
Emergency Fleet type, 664
gravity type, 663
pressure type, 667
pressure-gravity type, 665
types of, 663
Westinghouse type, 666
tanks, 678
Lubrication, 1381
air compressor, 1205
Diesel engine, 1386
forced feed, 1388
friction, 1381
coeff. of, 1382
general law of, 1387
gravity feed, 1388, 1390
pumps, 1161
temperature of, working, 1386
viscosimeters, 1394
Lumen metal (bearing alloy), 77

McEntee's simplified ship form, 1023
Machinery arrangements, 698, 702, 794–804
S. S. Mjölner, 702
U. S. S. Duncan, 795
U. S. S. New Mexico, 698
Magnalium, 68
Magnesia, block, 299, 1358
Magnesium chloride, (boiler corrosion), 466
Magnetite, 79
Magnolia metal, 76
Manganese bronze, 62
Manifolds, 1297
Manometer, 315
Manometric efficiency, 1168
Marine coals, 243
Marine engine oil, 1388
Mars, U. S. S. trials, 1444
Martin's new theory of steam turbine, 590
Martinsite, 100
Mass (def.), 27
equivalents (conversion table), 17
Master Car Builder's Association, 1456
Materials, boiler, U. S. Supervising Inspectors requirements, 514
Mathematical tables and formulæ, 1–26
Maxwell's thermodynamic relations, 198
Mayari steel, 106
Mechanical equivalent of heat, 196
refrigeration (*see Refrigeration*)
Mechanics defined, 27
of rigid bodies, 27–38
Melting iron and steel, processes for,
points of alloys (table), 55
of metals (table), 41
of refractories, 184
of solids, 183
Merchant bar, 89
iron, 89
Merida, S. S., engine room, 803, 804
turbines, 805
Metacenter of ships, 1021
Metal, metals,
anti-friction, 68–78
atomic weights of (table), 41
bearing, 68–78 (*see Bearing Metals*)
coefficients of expansion for (table), 41
electrical conductivity of (table), 41
flange, 58

Metals, melting point of (table), 41
non-ferrous, 41–78
physical constants of (table), 41
sp. gr. and wt. of (tables), 41
specific heat of (table), 41
strength at high temperatures, 68
Metallic coatings, methods of testing, 155
microstructure of, 155
packing, alloys for, 78
Metallography
compounds, 83
eutectics, 83
iron and steel, of, 82
non-metallic inclusions, 83
pure metals, 83
solid solutions, 83
Meters, oil, 318
Bowser oil, 321
Empire oil, 318
Steam flow, 322
Venturi, 322
Methyl chloride, 229; (properties), 221, 222
Micrometer bridge gage, 651
gear, turbine, 649
Microstructure of iron-carbon alloys, 84
Middle body, parallel, of ships, 1018
Midship section of hull, 1029
Milk of lime, (boiler corrosion), 467
Mixer, iron, 91
Mixture of gases and vapors, 223
Mixtures, freezing, 183
temperatures of, 182
Mjölner, S. S., machinery, 703
Model test, E. H. P. from, 1026
Modulus, section, 838
Moisture and ash free (coal), M & A F., 242
effect of, on turbine performance, 710
in air, 223
in coal, 241
measurement of, 224
Mol (def.), 200
Mollier diagram, 200
for ammonia, 219
for steam, 215
Moment, moments, 32
about parallel axes, 36
of areas, 36
of inertia (def.) 36
polar, 36
Momentum, (def.), 27
Monel metal, 66
Monochest type pump, 1156
Motion (def.) 27
accelerated, 29, 38
curvilinear, 30
harmonic, simple, 29, 38
in space, 32
of rigid body, 31
of rotation, 31
of translation, 31
parallelogram of, 30
parallelopiped of, 30
rectilinear, 29, 38
uniform, 29, 38
uniformly accelerated
Motor circuits, 1376
voltage drop in, 1364
alternating current, 1377
controllers, 1376
direct current, 1376
Muck bar, 89
Muntz metal, 58
Mushet steel, 107

Napier's formula for steam flow, 233
Neats foot oil, 1383
Nelseco Diesel engine, 957
Nereus, U. S. trials, 1446
New Mexico, U. S. S., electrical drive machinery, 694
generator, 697
governor, 696
switchboard, 699
test data, 721
turbine, 695
Newport News direct drive turbine, 774
geared cruising turbine, 777
nozzles, 774
Newton's Laws of motion, 38
Nickel clad steel, 155
marine use, 51
pipe, 1260
properties of, 51
Nitrous oxide, 229; properties, 221, 222
Non-conducting material, lagging, 436
Non-ferrous metals and alloys, 41–78
Non-freezing mixture for telemotors, 1225
Nonpareil boiler lagging, 439
Norske Veritas, 391
Nozzle, nozzles, 538
angle of, 541
blocks, 624, 784
construction, 625
converging, simple, 538
converging—diverging, 539
De Laval, 758
divergent, 539
flow of steam through, 234
efficiency, 540
flow of steam through, 234, 542
General Electric, 747, 753
Kerr, 764
length of, 541
Newport News, 774
Poole, 768
Sturtevant, 734
Terry, 788, 791
throat area, 542
Westinghouse, 736
Nuts, materials for, 127–129
U. S. standard, dimensions for, 161
weight of, 162

Oil-burning vessels, safety precautions for, 274
Oil cargo vessels, safety precautions for, 275
Oil cooler, coolers, 1111–1114
duty, 1111
for geared turbines, 1111
heat transfer rates for, 1113
installation, 664–666, 1113
operation of, 1114
service of, 1111
size, 1111
space occupied by, 1113
types of, 1112–1114
annular baffle, 1112
copper shell, 1114
helical baffle, 1113

Oil cooler, weights of, 1113
fuel, fuels,
atomisation of, 281
Bunker oil "A," "B," and "C," 272
burners, (*See Atomisers*)
burning, 281–329
air pressure for, 299
carbon on atomising tips when, 325
carbon on tubes, furnace sides and bottoms, 325
damage to brick walls when, 329
dampers set during, 323
extinguishers, Foamite fire, 327
fires on, vessels, 326
flareback in, 326
to prevent, 326
furnace volume for, 299
heating surface for, 299
increasing boiler capacity during, 324
lighting fires for, 323
oil leaks into feed system from, 329
oil pressure during, 324
operation of, 322
precautions for, 326
presence of smoke in, 299
rate of combustion when, 299
smoke, heavy, when, 324
soot on tubes when, 324
coefficient of expansion of, 262
conservation of, 273
corrosion due to, 272
Diesel engines, for, 272
equipment, 301
expansion tables for, 266, 267
fire point of, 269
flash point of, 268
heater, 302, 307–309
test, 1429
heating, effect of, 288
in cargo tanks, 310
in tanks or bunkers, 309
values of, 256
measurements of, 272
meters, 318
Navy standard, 271
physical properties of, 256
piping, 314, 1252, 1254
relative advantages, coal and, 254
relative costs, coal and, 252
sediment in, 270
specific gravity of, 262, 268
specific heat of, 268
specifications for, 271
storage and distribution stations for, 253
of, on board ship, 273
strainers, 310
system, 1252, 1253
test after installation, 1342
testing, 329
temperature viscosity curves for, 309
valves, 314
viscosity of, 256
to test for, 258
water in, 270
Oil heaters, oil leaks in, 328
leaks into feed system, 328
lubricating, 1205
air compressor, 1205, 1392
animal, 1383
carbon deposits from, 1399
Oil, lubricating, cloud test of, 1396
cold test of, 1393, 1396
color of, 1397
compounded, 1383, 1388, 1392, 1398
cylinder, 1386, 1391
Diesel engine, 1386
dirt and water in, 1388, 1395
effect of pressure on, film, 1381
emulsification test of, 1395
engine, 1386, 1388
filtered, 1391
filters, 671
fire point of, 1393, 1395
fire test, 1394
flash point of, 1391, 1393, 1395
for bearings, turbine, 638, 661
for Falk gears, 662
forced lubrication, 1388, 1389
ice machine, 1392, 1396
machine, 1386
marine engine, 1388
mineral, 1383
motor, 1386, 1396
odor of, 1398
physical constants of, 1389, 1391, 1394–1399
piping, turbine, 679
pour test of, 1396
pumps, turbine, 667, 744, 755, 790, 793
specific gravity of, 1397
strainers, 669
tanks, 679
unfiltered, 1391
vegetable, 1383
viscosity of, 1391–1394
pipe fittings, 671
Ores of iron, 79
Bessemer and non-bessemer, 80
extraction of iron from, 80
Orifice, flow of air, through, 233, 235
of gas through, 232
of sat. steam through, 233, 235
Orion, U. S. S. trials, 1446
equivalent, 1167
method, evaporator operation, 1094
Orsat flue gas analysis apparatus, 483
Osmondite, 100
Otto cycle, 205
Oxide of iron, protection of iron and steel by, 156
Oxy-acetylene cutting, apparatus for, 171
speed and depth of, 171
flame, character of, 165
gases for brasing, cutting and welding, 163
welding process, 163
Oxygene, 163

Packing, packings, 78, 871, 1161
carbon ring, 755
dummy piston, 559
labyrinth, 556, 761, 791
metallic, alloys for, 78
pump, 1161
Painting, piping, 1290
Pantograph, 1407
Paraffine, 1383
Parallelogram of motion, 30
Parallelopiped of motion, 30

Parker process, protecting iron and steel, 157
Parsons augmenter, 1000
bearing (turbine), 636
blading, 548
bronze, 74
turbines, 528
design of, 581–588
provisional proportions of, 566, 572
Peabody draft formula, 344
Pearlite, 85
Perfect gases, 202–207 (*see Gases, Perfect*)
Performance, estimated of Scotch boilers, 492
turbine, 710–732
Petroleum, 250
production of, 250
properties of, 251
Phosphor bronze, 65, 72
Physical properties of oil fuel, 256
Picking tables, 244
Pictet's fluid, 221, 222, 229
Pig iron,
Bell Krupp process, 97
foundry, 111, 148
percentage of carbon in, 81
washed metal, 97
Pipe, pipes,
accessibility for repairs, 1254
anchorages, 1245
areas of, comparative transverse, 1255
Barbow's formula, strength of, 1262
bends, 680, 1266, 1270
Crane Co., 1268
practical application of, 1269
radius, minimum advisable, 1268
U. S. Navy specifications on, 1268
boiler feed, 1254
blow-off, 1254
branch, from steam lines, 1253
brass, uses of, 1254
weight of seamless, 1259
bursting tests of, 1263
connections, 1271 (*see also Pipe Fittings*)
for telemotors, 1224
flanged, 1272
screwed, 1272
threaded, 1271
copper, advantages of, 1254
in bilge, 1245
uses of, 1254
covering and lagging, 1348
application of, 1358
losses through, 1350
magnesia, 1351
savings from, 1350
sizes of, 1358
specifications for, 1359
thickness of, economic, 1356
types of, 1358
cut-out valves in, 1245
dimensions and weights of, 1255
double extra strong wrought, 1257
extra strong wrought, 1257
standard wrought, 1256
double extra strong, 1255
exhaust, 681
expansion of, linear, 1266
bends, 1267
joints, 681, 1268, 1271
accessibility of, 1245
extra strong, 1255
Pipe, fittings, 1284 (*see also Pipe Flanges, Pipe Joints*)
American standard, 1284
classification, Emergency Fleet, 1292
extra heavy, 1287
hydraulic, 1288
flanged, 1272, 1284
oil, 671
practice, general 1283
requirements, Emergency Fleet, 1292
screwed, 1272
temperatures allowed in, 1292
flanges, 1272
American standard, 1274
brass, extra heavy, 1276
heavy, 1276
bulkhead, 1278–1284
composition, 1276–1281
flat face, 1580
for copper pipe, 1276–1281
Naval practice, 1272, 1277–1284
Newport News practice, 1276
oil, 679
painting, 1290
peened type, 1272
steel, 1282–1284
Van Stone type, 680, 1272
flow of steam in, 236
flues, thickness of, 1266
heat losses from bare, 1349
heating, 1178
in ingots, 97
indentification colors on, 1290
iron, lead lined, 1260
joints, 1245
expansion or slip, 1268, 1271
Lovekin, 680
oil, 679
to turbines, 680
lead, 1261
lubricating oil, 679, 1254
materials for, 1254
oil fuel, 1252, 1254
lubricating, 1254
painting, 1290
plans, 1245
pressure, collapsing, of, 1265
radiator, 1177, 1180
standard wrought, 1255
steam, to sea injection, 1253
turbine, 679
steel,
advantages of, 1255
lap-welded, uses of, 1254
seamless, uses of, 1254
sheet, galvanised, 1254
weights of, 1255
strength of, 1262
collapsing, 1265
relative, wrought iron and steel, 1264
welds, 1263
systems, 1154
discharge, 1251
exhaust, 1248, 1249
fuel oil, 1252, 1253
steam, 1246, 1247
suction, 1250
tests, bursting, 1263
thickness of, 1261
threads, 1271, 1278
weights of (*see also Pipes, dimensions, etc.*)

Pipe, aluminum, 1260
brass, seamless, 1259, 1260
weights of, cast iron, 1260
copper, 1260
lead, 1260, 1261
nickel, 1260
seamless steel, 1258
tin, 1260, 1261
wrought iron, 1260
welds, Thermit, 176
wrought iron, advantages of, 1255
uses of, 1254
weights of, 1260
zinc boxes for, 1291
Piping, compressed air system, 1203
evaporator, 1093
fuel oil, 314
in merchant service, 314
in Naval service, 314
Newport News practice, 314
test before installation, 1429
Piston, balance, reciprocating engine, 893
turbine, 632
Planimeter, 1413
Amsler polar, 1413
Coffin, 1414
Plastic bronze, Ajax (bearing metal), 74
Plates, boiler, 124
flat, strength of, 848
floor, 132
nickel steel, 131
steel, for hulls, 127
Platinite, 105
Plenum system of ventilation, 1183
Plumber's solder, 78
Plumbic bronze, 72
Polytropic curves, construction of, 203
expansion of gases, 202; (table) 203
Poole turbine, 767–773
test data, 717
Porpoise jaw oil, 1383
Potential energy, (def.), 27
Pound (def.), 14
Power (def.), 27
equivalents (conversion table), 18
system, electrical, 1376
Powering of ships, 1024–1027
Admiralty formula, 1025
E. H. P. from model tests, 1026
Steven's formula, 1025
Taylor's curves, 1026
Preece test, 155
Preservation of idle boilers, 464
Pressure, absolute, 967
back, effect on therm. effy. recipro. eng., 929
collapsing of pipes, 1265
critical, in nozzles, 538
effect on turbine performance, 710
equivalents (table) 18
gages for welding gases, 163
governor, centrifugal pump, 1129
regulator, welding gases. 163
safe, in turbine bearings, 636
standard atmospheric, 967
-volume diagram, 199, 529
Pricker bar, 452
Prime movers, (*see also Turbines Reciprocating Engines, Diesel Engines*)
performance, comparison of, 723
Prismatic coefficient of hull, 1014, 1029
vertical, 1021, 1029
Projected area ratio, propeller, 1048
Propeller, propellers, 1028–1076
after body ship lines, 1029
analysis and design of, 1036–1061
forms for, 1046
blade sections, 1049
thickness, 1050
tip clearance, mean relative, 1033
broad, fan and narrow, 1034, 1059
general rule, 1033
block coefficient of hull, 1029
cavitation, 1043
correction of power for, 1045
design, 1055–1061
forms for, 1056
hubs, 1064–76
four, blades, built up, 1073
solid, 1064
three blade built-up, 1066
hull appendages, 1034
resistance of, 1035
coefficients, 1029
load factor, limitation of, 1055
materials used in, construction, 1062
midship section of hull, 1029
prismatic coefficient of hull, 1029
vertical, 1029
projected area ratios, 1038
forms, 1048
revolutions of, 1029
single screw ship, 1031
slip, apparent and real, 1029
block coefficient, 1029
correction for var. of mid. sec., 1031
equations for, 1030
submarine, 1031
thrust deduction, 1028, 1029
factor determining, 1032
tunnel boat, 1031
wake gain, 1028, 1029
wing screw, 1030
Protective coatings for iron and steel, 153–157
Proteus, U. S. S., trials, 1446
Proximate analysis of coal, 241
Publication, publications,
of A. I. E. E., 1456
of A. R. E. A., 1457
of A. S. M. E., 1454
of A. S. N. E., 1458
of A. S. T. M., 1454
of Amer. Bur. of Shipping, 1453
of Amer. Inst. of Weights and Measures, 1458
of Bureau of Standards, 1452
of Steamboat-Inspection Service, 1451
Puddle bar, 89, 95
Pump, pumps,
air, 990, 997
attached, 906, 991
classified, 990
dry, 992, 1006
Edwards, 991
ejector, 994
Le Blanc, 996
Radojet, 996
independent, 990
Le Blanc hydraulic, 994
reciprocating dry, 992
rotary entrainment, 994
Rotrex, 992
vertical twin beam, 992

Pump, air, wet, 991, 1006
and dry, 997
booster, oil fuel, 302
centrifugal, 1115–1130, 1132
bearings, 1121
bi-rotor, 1119
boiler feed, 1121
bushing rings, 1121
capacity of, 1117
casings, 1119
characteristic curves of, 1115
circulating, 1123
condensate, 1000
couplings, 1122
double suction, 1132
driving, methods of, 1122
feed, boiler, 1127
parallel operation of, 1129
govenor, pressure, for 1129
speed, for, 1128
head, suction, 1118
total, 1117
impellers, 1117
inspection of, 1128
multi-stage, 1127
operation of, 1128
parts of, 1117
pressure governor, 1129
relief valve for, discharge, 1129
repairs to, 1128
ring oil bearings, 1121
shafts, 1120
speed, variations in, 1116
governor, 1128
stuffing boxes, 1120
suction head, 1118
turbine, 1132
volute chamber, 1119
circulating, 988–990, 1006, 1123
double inlet, 989
high speed, 989
low pressure, 988
slow speed, 989
condensate, 1000, 1006
evaporator feed and brine, comb., 1094
functions of, 1132
governors,
auxiliary operated, 1339
constant pressure, 1337
excess pressure, 1339
types, 1339
hand, oil fuel, 302
Kinney, 304
capacity of, 305
oil turbine lubricating, 667, 744, 755, 782, 790, 793
capacity of, 668
geared type, 667
Quimby, 306
reciprocating, 1131–1162
air, 990–997
bound, 1162
chambers for, 1134, 1161
dry, 992, 1006
Edwards, 991
vertical twin beam, 992
wet and dry, 991, 997, 1006
bolts, 1153
capacities of, 1135–1148, 1152
cargo oil, 1149
classification of, 1131
Pump, reciprocating, condensate, 1000
connected, 1132
connections, 1161
sizes of, 1135–1148
construction of, 1133, 1136, 1139, 1143, 1147
crossheads, 1133
cylinders, 1152
Davidson valve gear for, 1156
delivery, actual, of, 1152
design factors, 1152
direct-acting, 1131
double-acting, 1132, 1135–1137, 1141–1148
duplex, 1132, 1137, 1145, 1146
valve gear, 1159
efficiency, hydraulic, 1151
indicated, 1152
mechanical, 1152
total, 1152
volumetric, 1151
fittings, 1134
heavy pressure, 1134, 1135–1137
independent, 1132
installation and care of, 1160
laying up, 1162
light pressure, 1134, 1143–1146
lubrication of, 1161
materials for, 1133
monochest type, 1156
oil cargo, 1149
fuel, 303
packing, 1161
performance factors, 1151
piston, 1134
rods, 1133
pressure, limits of, 1134
pump end, 1133
service of, 1134
simplex, 1132
single-acting, 1132, 1138–1140
sizes of, 1133, 1134–1148
space occupied by, 1135–1148
specification, commercial, 1132
speeds of, 1135–1148, 1152
steam end of, 1133
types of, 1131
valve, valves,
air pump, 1154
assembly, 1156
bronze, 1153
decks, 1154
gear, steam, 1133
Kinghorn, 1155
poppet, 1154
rotating, 1155
rubber, 1153
seats, 1133, 1154
service, 1153–1156
steam, 1154
turbine, 1155
valve pot type,
Worthington valve gear for, 1156
rotary plunger, oil fuel, 302, 304
screw oil fuel, 302, 306
service, oil fuel, 302
test before installation, 1429
turbines for driving, 742
valves, 909
Westinghouse turbine drive for, 742
Pure metals, 83

Pyrometers, 316
calibration of, temp. for, 177
optical, 316, 317
potentiometer type, 316

Quinby screw pump, 306

Radians to degrees (conversion table), 6, 11
Radiating capacities (heat), 194
Radiation constants of materials, 194
heat transmission by, 187, 194
in turbines, 533
Stefan-Boltzmann law of, 194
Radiators, heating, 1177, 1180
Radio-telegraph, 1378
Radius of gyration, (def.), 36
Radojet air pumps, 996
augmenter, 1000
Rankine cycle, steam, 226
Rapeseed oil, 1383
Rateau, formula for steam flow, 233
turbines, 527
design of, 577
Rational functions, indef. integrals, 23
Reaction, degree of (Parson blading), 549
blading, 620
Reaumer process, malleable iron, 117
Recarburizers, 91
Reciprocating engine, engines, 821–941
air pumps, attached, 906
valves, 909
Aspinwall governor, 912
back pressure, effect on thermal efficiency, 829
balance, means of improving, 932
balancing, 920
analytical method, 921
graphical method, 926
Yarrow-Schlick-Tweedy system of, 933
backing power, 938
bearing, bearings, 864, 940
boxes, 864
caps, 866
horse shoe thrust, 867
line shaft, 869
main, 864
oil boxes, 866
pressure, allowable, 866
shoes, 868
spring, 869
steady, 868
stern, 870, 941
thrust, 867
unbalanced load on, 866
white metal, 864
beds, 850
bedplates, 852
bull rings, 858
card factor, 832, 833
Carnot cycle, 827
center of gravity of, 938
classification of, 821
clearance, effect on expansion, 826
on steam consumption, 826
on work, 826
linear or piston, 826
volume, actual, 826
volumetric, 826, 842
Reciprocating engine, compounding, 83
connecting rods, 862
angularity of, 883
cranks, arrangement of, 916
crosshead, 860
guides, 852
pins, 861
slippers, 862
cut-off in, 836, 883
cylinder, cylinders, 840
arrangement of, 915
assistant, Lovekin, 894
balance, 893
barrels, 840
thickness of, 842
bolting, 843
by-pass valve, 845
clearance, 940
condensation in, 829
counterbore, 844
covers, 847
drains, 844
flanges, 843
H. P., 841, 842
indicator connections, 845
L. P., size of, 831
valve chest, 841, 842
lagging, 845
liners, 840, 841
securing, 843
thickness of, 842
Lovekin assistant, 894
ports, steam, 843
velocities in, 847
ratio of, 836
relief valves, 844
stuffing boxes, 845
testing of, water, 842
thickness of, 842
walls, thickness of, 843
designs, 915
development of, 822
dimensions, overall, 939
drafting room methods, 918
eccentric, eccentrics, 898
rods, 897
open and crossed, 878
straps, 898
economy, effect of superheat on, 830
Edwards air pumps, attached, 906
efficiency, mechanical, 830
thermal, 827
effect of varying pressures on, 829
engine room for, length of, 939
expansion, effect of clearance on, 826
in different types of, 826
quadruple, 917
ratio of, 825
total in, 825
triple, 916
factors of safety, 827
feed pump, attached, 909
framing, 850
frictional loss in, 830
gages and gage board, 915
governors, 912
Aspinwall, 912
gratings and platforms, working, 913
handling gear, 913
housings, 850
jackets, steam, 830
lap, steam and exhaust, 882

Reciprocating engine, laying out the, 1018
link, links, 897
blocks, 897
slip of, 878
lubrication of, 914
machining and tolerances of, 939
mean effective pressure, 833
actual vs. theoretical, 833
referred pressure, 832
moments, rocking (balancing), 925
overload, steam consumption at, 831
packing, metallic, double 846
for high pressure, 846
for low pressure, 846
for new engine, 847
for superheated steam, 846
interchangeable sets, 857
performance of different, 834
pipes, steam velocity in, 847
piping and fittings, 911
piston, pistons, 855
balance, 893
clearances, 844, 940
rings, 856, 940
rod stuffing box, 845
rods, 858, 940
speed of different, 832
platform and gratings, working, 913
power of, effect on steam consumption, 831
pressure, back, 826
drop in, 824
in cylinders, 824
in valve chests, 824
limits in, 824
unbalanced steam, in, 829
proportioning parts of, 837–918
pump, attached, 831, 906
feed, 909
gear, attached, 910
valves, 909
quadruple expansion, 917
R. P. M. of different, 832
for different percentages of full power, 938
Ramsbottom rings, 858
Rankine or ideal cycle, 827
ratio of expansion in, 825
receivers, steam velocities in, 847
reciprocating parts, 855
re-evaporation in, 829
return connecting rod type, 822
reverse shaft, 901
reversing engine arm, 902
size of, 906
gear, 899
all round type, 899
direct acting type, 900
load, 900
revolutions, critical, and hull vibration, 934
rocking moments (balancing), 925
shaft, shafts, 871
arrangement of, 871
brake, 877
couplings, 876
crank, 871
pins, 873
line, 875
moment, ratio of max. to mean, 873
propellers, 875
sleeves, 875

Reciprocating engine, shaft, reverse, 901
stress in, 874
stuffing boxes, 870
thrust, 874
collars, 875
tunnel, 871
webs, 873
for built up, 874
side lever, 822
size of, determining, 831
sole plate, 869
steam consumption of, 831
effect of expansion on, 826
jackets, 830
stern tube, 870
lubrication, 870
stuffing box, 870
stresses in, 837
stroke of, length of, 831
strut bearings, 871
stuffing boxes, 847
bulkhead, 871
for metallic packing, 847
stern tube, 870
superheat, effect on economy of, 830
suspension rod (reversing eng.) 904
test data, 719, 834
throttle valve, 911
thrust indicated and effective, 837
tolerances in machining, 939
tools for, 915
triple expansion, 916
trunk, 822
turning effort, 934
gear, 904
arrangement of, 905
size of, 904
twisting moments, 935
unbalanced load on different pistons, 839
vacuum in, 826
limits of, 969
valve, valves,
action of, 882
angle of advance, 882, 884
butterfly, 911
chest, chests, 841, 842
covers, 848
piston, 842
slide, 842
cut-off, 883
diagram data, 882
oval, 881, 887
Zeuner, 884
double ported, 877
expansion, 892
functions, 882
gear, 877–899
Joy, 877, 881
Marshall, 877, 881
radial, 877, 881
Stephenson link, 877
lap, steam and exhaust, 882, 884
lead, 882, 883
liners, 892
link, 878
load, 894
motions, 877
piston, 891
stuffing box, 846
point of admission, 888
of compression, 882, 885, 888

Reciprocating engine, valve, pumps, attached, 906
valve, release, 882, 888
slide, double ported, 890, 940
stuffing box, 846
stem, 895, 940
crossheads, 896
guides, 896
throttle, 911
size of, 847
travel, 882, 883
trick, 877, 893
vibration, hull, and critical revolutions, 934
water service for, 914
weights of, 938
work, effect of clearance on, 826
Reciprocating pumps (*see Pumps, Recipro*).
Red metal (brass), 64
short steel, 90, 103
Reducing motions, 1406
valves (*see Valves, Reducing*)
Reduction gear, Alquist, 747
cruising turbine, 778
electrical, 692–707
hydraulic, 707–709
mechanical, 806–817
Alquist, 808
design, 808–814
helix angle, 806
pinion, 808
bearings, 810
couplings, 811
jaw, 812
pin, 811
double, 807
journals, 810
number of teeth on, 810
single, 747
shafts, flexible drive, 813
teeth, involute, 806, 814
accident to, 815
corrosion of, 816
damage to, 816
lubrication of, 815
pitting, 816
scoring, 816
wear of, 817
testing, 816
type, types, 806
double helical, 806
fixed frame, 806
flexible or floating frame, 806
wheels, 813
Reflecting powers (heat), 194
Refractories, melting points of (table), 184
Refrigerating fluids, properties of, 218–222
Refrigeration produced by ammonia and CO_2, 230, 231
theory of, 227
Regenerative compressor (evaporator), 1091
Registers, air control, 291–297
Bureau S. E., forced draft, 291, 295
Bureau S. E. natural-forced draft, 292, 296
closed fire room system, 293
Dahl atomiser and, 295, 296
Howden system, 293
induced draft, 293
Natural draft, 298
Registers, Peabody, 294, 296
purpose of, 292
Schutte-Koerting, 296, 297
types of, 295–297
Registro Navale Italiano, 391
Reheat factor, 535
Relative advantages, coal and oil fuel, 254
costs, coal and oil fuel, 252
Retarders, boilers, 377
Richardson-Phenix oil filter, 674
Ringelmann's scale, 318
Rivet, rivets, 402
countersunk, 404
dimensions, 404
heads, 403
holes, 402
length of, stock, 403
Riveted joints, 393–402
Rochling-Rodenhauser electric furnace, 96
Rosary construction (turbine blading), 622
Rose's metal (fusible alloy), 78
Rosin oil, 1383
Rotation, 31, 38
and translation, combined, 31
Rotational losses, turbine, 561, 562
Roto tube cleaners, 487
Rotor, rotors (turbine)
critical speed of, 605
drum, design of, 609
lifting gear, 689
shafts, design of, 609
turbine, design of, 592
balancing, 640
Rotrex air pump, 992
Rudder torque, 1219

S. h. p., 1403
Safety, factor of, boilers, 391
precautions for oil-burning vessels, 274
oil cargo vessels, 275
size of, 448
valves, 447 (*see Valve, Safety*)
Salinometer, 381
Salt solutions, density of, 184, 185
freezing point of, 184, 185
specific heat of, 182, 184, 185
Sampling coal, 247
Saturated steam (*see Steam Saturated*)
properties of, 209
vapors, 208, expansions of, 223
Scale, boiler, removal of, 487
Scavenging Diesel engines, 947, 950
Schneider Diesel engine, 957
Schoop process, protection of iron and steel, 154
Scoop condensers, 982
Screw threads, Acme, 160
A. S. M. E. standard, 159
British Ass'n., 159
French (metric) standard, 159
International (metric) standard, 160
Metric, 159, 160
pipe, Briggs
Whitworth
power transmission, 160, 161
Sellers, 158, 161
Square, 161
U. S. Standard, 158
Whitworth, Standard, 158

Scum pans, 441
Searchlights, 1375
Sectional areas of ships, curve of, 1014
Segregation in ingots, 98
Semi-bituminous coal, 243
Service pump, oil fuel, 302
Shaft, centrifugal pump, 1120
  couplings, turbine, 609
  General Electric, 747
  Kerr, 763
  lining up, 647
  Ljungstrom, 780
  Poole, 767
  Sturtevant, 786
  Terry, 791
  turbine, De Leval, 761
  Westinghouse, 611
Shear (def.), 32
Sherardizing, 153
Ship, ships,
  body plan, 1022
  breadth of, 1013
  depth of, 1014
  draft of, 1014
  economical speed of, 1014
  forms, 1013–1024
  freeboard determined by, 1014
  length of, 1013
  load water line, 1022
  McEntee's simplified form, 1023
  metacenter, height of, 1021
  middle body, parallel, 1018
  prismatic coefficient of, 1014
    vertical, 1021
  sectional areas, curve of, 1014
  simplified forms, 1023
  speed, economical, 1014
  Whittelsey's simplified form, 1023
Shonberg M. Metal (for bearings), 72
Shrinkage of castings, 179
Shroud rings, turbine blading, 619
Siderite, 79
Sil-O-Cel brick, 439
Simplified ship forms, 1023
Simpson's Rule (integration), 26
Sine Tables, 6
Siren, steam, 1196
Siroccosone system, ventilation, 1191
Slice bar, 452
Slip, apparent and real, propeller, 1029
  block coefficient, propeller, 1029–1031
  lines in metal, 98
Smelting iron, 71
Smoke chart, 318
  observations (boiler tests), 492
  oil fuel burning, 299, 324
Smothering apparatus, oil burning vessels, 275, 327
Soda, use for removal of oil in boilers, 465
  ash, use in boilers, 468–474
Sodium-chloride solutions, 182–185
Soft solder, 78
Solders, 78; fluxes for, 78
Solidus, 84
Soot blowers, 380
  door, 380
  hoe, 380
  lance, 380
  on tubes, oil fuel burning, 324
Sorbite, 100
Sorbitic pearlite, 100
Souther babbitt (for bearings), 72
Specific gravity—Baume' tables, 263–265
  of metals (table), 41
  of fuel oil, 262, 268
  heat, mean, of air, 1199
  weight of air, 1198
Specific heat, heats, 180
  of air, mean, 1199
  of gas mixtures, 201
  of gases, 200
  of iron, mean, 181
  of liquid, mean, 181
  of oil fuel, 268
  of metals (table), 41
  of salt solutions, 182, 184, 185
  of solids, mean, 181
  of superheated steam, mean, 214
  of water, 180, 181
Specific volume of steam, 529
  chart for, 532
Specification Societies, 1450
Speed, critical, of turbine rotors, 605
  economical of ships, 1014
  turbine, effect on performance of, 711
Speller-Harter boiler water testing kit, 40
Spelter, 47 (*see also Zinc*)
  analyses of, 48
  grades and uses, 48
  solder, 78
  specifications for, 48
Sperm oil, 1383
Spherical excess, 22
Spiegel-eisen, 91
Spilling over, steam in turbines, 554
Spontaneous combustion of coal, 245
Spring steel, rolled and drawn, 133
Springs, phosphor bronze for, 65
Stage efficiency (turbines), 534
  ratio, 536
Standard locality, 14
Standardization trial of vessels, 1436
Stassano electric furnace, 95
Static balance, turbine rotors, 640
Statics, 32
Steadite, 112
Steam consumption, 355, 538
  dry, 490
  effect on boiler design, 354
  expansion of, actual, in turbines, 533
    adiabatic, 529
  flow in pipes, 236
    through diverging nozzles, 234
    through nozzles, coeff. for, 236
    through orifices, 234; coeff. for, 236
  from indicator cards, 1412
  geared turbines, 356
  Molliere's diagram for, 215
  outlets, boilers, 379
  pressure, 589
    -temperature relations, 212
  properties of, 528
  pumps (*see Pumps*)
  purifiers, 384
  raising, 464
    rate of, 465
  saturated, flow through orifices, 234
    heat content of, 212
    properties of (tables), 215
  speeds in piping, 680, 681
  superheated,
    mean specific heat of, 214
    properties of (table), 217
  tables, condenser calculations, 213

Steam, tables, saturated steam, 209–212
superheated steam, 217
thermal conductivity of, 191
throttling of, 237, 533
tiller, 1206
volume calculations, 531
Steamboat-inspection Service, 1450
Steel, steels
air hardening,
alloy, 135, 146
and iron, testing, 121–153
annealing, 99
A. S. T. M., specifications, for, 142–148
bars and shapes for hulls, 127
Bessemer, 89
acid, 90
basic, 90
deoxidation of, 90
recarburisation of, 91
uses of, 91
billets, blooms and slabs, 134, 142
boiler bracing, 126
plates, 124
rivets, finished and rods for, 125, 142
tubes, seamless, 123
blooms, billets and slabs, 134, 142
bolts, nuts and studs, 127–129
castings, 131, 138, 140, 146
chromium-molybdenum, 107
chromium-nickel, 106
chromium-vanadium, 106
cold drawn, 98
cold short, 103
crucible, 94
electric, 95
extra soft, 122
floor plates, 132
forgings for hulls, engines and ordnance, 136–138
galvanised plates, 127
grain growth of low carbon, 103
hardening, 99
heat treatment, effect of, 98
high silicon, 103
influence on physical properties of
chromium, 105
copper, 108
manganese, 104
molybdenum, 10—
nickel, 104
phosphorous, 103
silicon, 103
sulphur, 103
titanium, 108
tungsten, 107
zirconium, 108
ingots, 97
croppings of, 98
defects in, 97
segregation in, 98
machinery, 133
manganese, properties of, 104
Mayari, 106
mechanical treatment, effect of, 98
Mushet, 107
Navy specifications for, 122–141
nickel-molybdenum, 107
nickel-zirconium,
nuts, 129
open hearth, 92
acid, 92
Steel, open hearth, basic, 92
slag composition, 93
Talbot process, 93
pipe (*see Pipe Steel*)
plates for boilers, 124
for hulls, 127
galvanized, 127
rivet, 131
seamless boiler tubes, 123
shapes and bars for hulls, 127
sheet, black and galvanized, 132
silico-manganese, 103
single shear and double shear, 94
slag composition, 93
specifications for, 122
spring, rolled and drawn, 133
stainless, 105
structural, 131, 142
studs, 129
tempering, 100
test pieces, 121–153
tool, 100
carbon, 140
self hardening, 107
tungsten, 141
vanadium, 106
welding wire and rods, 132, 166
—White cast iron series, 83
Steering gear, 1206
Brown steam tiller, 1206
control, 1213
American Engineering Co. type, 1213
automatic follow-up, 1218
electro-hydraulic, 1215
Hele Shaw pump type, 1214
shafting, 1222
telemotor, 1223
electric, 1225
hydraulic, 1223
non-freezing mixture, 1225
pipe connections, 1224
tests of, 1432
wire rope, 1222
drum type, 1213
installation of, 1222
quadrant type, 1206
rudder torque, 1218
screw type, 1209
size of, 1209
steam tiller type, 1206
trials, final, 1442
Stefan-Boltzmann law (radiation), 194
Steven's formula, powering ships, 1025
Stirling cycle, 204
Stokers, mechanical, 381
Storage and distribution stations for oil fuel, 253
of coal, 245
Strainers, 310–313
basket type, 310
Cramp type, 313
duplex, 302, 311, 670
fuel oil, 310
lubricating oil, 669
Schutte & Koerting, 312
steam, 680, 781
Studs, material for, 127–129
Stuffing boxes, centrifugal pump, 1120
Sturtevant turbines, 784–787
Submarine signal sets, 1378
Sulphur dioxide, properties of, 219, 221
in coal, 242

Sulphurous acid (*see Sulphur Dioxide*)
Sulzer Diesel engines, 959
Superheat (turbines), 589, 710, 726
  effect on economy, (Idalia,) 513
  measurement of (boiler test,) 491
Superheated steam (*see Steam Superheated*)
  auxilliary machinery, desirable for, 356
  effect on steam consumption, 356
Superheaters, 461
  Babcock and Wilcox, 461
  Fire tube boiler, 463
  Foster, 463
  location of, 462
  Locomotive Superheater Co., 463
Supervising Inspectors, U. S. Board of, 390
Surface blow, boiler, 379
Surface-ignition engines, 945
Swash plates, boilers, 441
Sweedish iron, 89
Switchboard, U. S. S. New Mexico, 699
Switchboards, 1371
Switches, door, 1373

Tables and formulæ, mathematical, 1–26
Tack holes (riveting), 404
Tagliabue hydrometer, 262
Talbot process, 93
Tallow oil, F383
Tangent tables, 6
Tanks, oil lubricating, 678
  reserve, 678
  storage, oil fuel, 301
Taylor's curves, powering ships, 1026
Telegraphs, engine, 1377
Telemotor control, steering gear, 1223
  piping, test of, 1432
Telephones, 1377
Temper colors, 101
Temperature, temperatures,
  absolute, 195
  effect on fan performance, 1169
  -entrophy diagram, 200, 530
  final, of compressed air, 120
  measurements, 177, 315–318
  of iron and steel by color, 177, 178
  of mixtures, 182
  of ventilating air, 1182
  stress due to change of, 179
  -viscosity curves, oil-fuel, 288, 309
Tempering steel, 100
Tennessee, U. S. S., turbines and generators, 737–742
Tensile strength of alloys at high temperature, 68
Tensilite, 62
Tension (def.) 32
Terminal difference, condensers, 971
Terne, 154
Test, tests,
  boiler, 488
  calorimeter, 490
  care required making, 488
  data, reciprocating engines, 719, 834
  turbine, 715–732
    direct vs. geared drive and recipro. eng., 719
    electric drive, 720–723
    freighter, 718
    geared vs. direct drive, 719
    geared vs. recipro., eng., 719
    Poole geared, 717
Test, data, turbine, small non-condensing, 722
    U. S. S. Jupiter, 721
    U. S. S. New Mexico, 721
    U. S. S. Wadsworth, 720
  distilling plant after installation, 1432
  feed system after installation, 1432
  machinery, 1429–1434 (*see also Trials of Machinery*)
    after installation, 1431
    before installation, 1429
    on new vessels, 1430
  object of making, 488
  of U. S. S. Cincinnati boiler, 505–513
    Idalia (superheat), 513
  machinery, S. S. Oklahoma, 499
    triple expansion engine, 499
  oil fuel system after installation, 1432
  piping before installation, 1430
  refrigerating machines, 1432
  Scotch boiler, 492–495
  smoke observations during, 492
  telemotor piping, 1432
  tension specimen (boiler plate) 517
  turbines for air leaks, 1432
Testing air compressors, 1203
  fuel-oil system, 329
  outfit, boiler water, 469–471
  triple expansion engine, 499
  turbines, 712
Thermal conductivity of air, 191
  efficiency, effect of back pressure on (recip. eng.), 829
  of gases, 191
  of heat insulating materials, 189, 190
  of liquids, 191
  of metals, 189
  of steam, 191
  of various solids,
  unit, British, 180
Thermit
  composition of, 172
  in foundry practice, 176
  kinds of, 172
  physical properties of, 173
  pipe welds, 172
  welding process, 172
Thermocouples, base and precious metal, 316
Thermodynamics, 195
  changes of state of gases, 204
  considerations, turbine, 589
  development, probable, turbine, 589
  equation of gases with variable specific heat, 202
  first law of, 196
  general thermodynamic relations, 198
  graphical representation in, 199
  mixtures of gases and vapors, 223
  of condensers and vacuum, 970
  perfect gas equation, 202
  saturated vapors, 207, 223
  second law of, 197
  vapors, 207
Thermofan system, ventilation, 1189
Thermometer, thermometers,
  nitrogen, 316
  oil, 315
  readings, conversion of, 177
  scales, comparison of, 177
Thermo-tank system, heating ships, 1180

Threads, pipe, Briggs, 1271, 1273
screw, 158–162 (*see Screw Threads*)
Throttling, 237, 533
cooling effect of, 237
loss due to, 237
Thrust bearings (*see Bearings, Thrust*), 637
block losses, turbine, 563
deduction, propeller, 1028–1032
Tiller, steam, 1206
Time firing, 478
Tin, Banca, 49
-copper alloys (*see Copper-tin alloys*), 72
-copper-lead bearing alloys, 73
-copper-lead-zinc bearing alloys, 74
-copper-zinc bearing alloys, 77
grades of, 50
-lead-antimony bearing alloys, 76
phosphor, 50
pig, analysis of, 50
pipe, 1260
properties of, 50
Tipple, 244
Tobin bronze, 59, 71
Tool steel, carbon, 140
tungsten, 141
Tools, fire
Torque, 33, 38
Torsion dynamometer, 713
Torsion meter
constant by shaft calibration, 1417
by calculation, 1420
Denny Edgecombe, 1425
electro-mechanical type, 1417, 1425
Gary-Cummings, 1420
Hopkinson-Thring, 1423
mechanical type, 1417, 1420
mirror type, 1417, 1423
use of, 1403
Transcendental functions, indef, integrals, 24
Translation, 31
and rotation combined, 31
Transmission of heat (*see Heat Transmission*)
in condensers, 192, 793
Traps, steam,
ball float, 1340
lever type, 1341
bucket float, 1340, 1341
inverted, 1342
buoyancy of float of, 1344
capacities, 1345
design of, 1348
effective pull of float of, 1344
expansion or thermostatic, 1340, 1343
fittings, 1347
installation and operation,
Naval requirements, 1347
non-return, 1340
ratings, 1345
requirements, 1343
return, 1340
testing, 1347
types of, 1342
Trails of machinery, 1434
backing, 1442
builders, 1434
commissioning and post repair, 1448
contract acceptance, 1435
dock, 1435
maneuvering, 1442
merchant vessel, 1448
Trails of machinery, Naval, 1434
auxiliary vessel, 1448
object of, 1434
of S. S. Edward Luckenbach, 496
S. S. La Brea, 498
S. S. Los Angeles, 498
S. S. Oklahoma, 499
triple expansion engines, 498
U. S. S. Achilles, 1444
U. S. S. Cyclops, 1444
U. S. S. Hector, 1446
U. S. S. Jason, 1446
U. S. S. Mars, 1444
U. S. S. Nerus, 1446
U. S. S. Orion, 1446
U. S. S. Proteus, 1446
U. S. S. Ulysses, 1444
U. S. S. Wadsworth, 720, 1438
rules for, 1434
standardisation, 1436
of U. S. S. Wadsworth, 1437
steering engine, 1442
Triangles, plane, relations between three angles of, 21
relations between sides and angles of, 21
spherical, relations between angles and sides, 22
Trigonometric functions, 6
Troosite, 100
Try cocks, 447
Troosto—sorbite, 100
Tschibyscheff's Rule (Integregation), 26
Tube, tubes (*see also Pipe, Pipes*)
cleaner, Roto, 487
condenser, 976–978
plate, condenser, 976
Turbine, turbines, 523–817
atern, 562, 572
balance pistons, 632
barrel, 551
bearing, 563, 635
friction, 563
blade, blades (*see Blade, Blades, turbine*)
capacity, limiting factors of, 563
losses in, 561, 562
casing, 625, 735, 739, 747, 753
glands, 556, 633, 755
clearance, axial, 554
diaphragm gland, 557
shaft gland, 555
couplings, shaft, 609
cruising, geared, 777
Curtis stage type, 526, 577
design of, 575
-Parsons turbine design, 588
-Rateau turbine design, 580
cylinder, 551, 624
drains, 628
De Laval type, 526, 755
design, 573–588
diaphragm, 624
labyrinth packing, 634
direct connected, 528, 572, 774
disc, discs (*see Disc, Discs*)
divided flow, 716
drains, 628, 681, 1004
drum losses in, 561, 562
dummy packing, 559, 624, 631
efficiency, wheel, 563
end thrust in, 631
foundation, 683
liners, Lovekin, 686

Turbine, friction, steam, in, 534
geared, 528
General Electric, 745–755
Haerle's curves, discs, 601
Impulse, multiple, type, 527
provisional proportions of, 564
-reaction type, 527
simple type, 526
design of, 573
Kerr, 762–767
labyrinth packing, 556, 633, 634
leakage, blade, 533
lifting gear, 687
Ljungstrom, 703, 779
losses, bearing friction, 563
disc, blade and drum, 561
rotational, 561
thrust block, 563
lubricating system, 661, 751
Martin's new theory of the, 590
mechanical design, 592
New Mexico, U. S. S., 695
Newport News direct drive, 774
nozzle blocks, 624, 626
Parsons type, 528
design of, 581–588
provisional proportions of, 566, 572
parts, test of, 1429, 1430
performance, 710–732
Poole, 767–773
radiation in, 533
Rateau type, 527
design of, 577
reduction gears, 692–707, 806–817 (*see Reduction Gears*)
return flow, 792
reversing, 777
rotors (*see Rotor, Rotors*)
shafts, 647, 747, 761, 763, 767, 774, 780
shroud rings, 619
small, auxiliary drive, 742, 773
steam seal, 633
Sturtevant, 784–787
Terry, 788–795
test data, 715–731
for air leaks, 1432
testing, 712, 1429
turning gear, 646, 773
types, 526, 732–794, 801
vacuum, limits of, 970
velocity stage types, 526
vibrations in, blading, 613, 622
water gland, 633, 634
rate, ideal, 531
Weaver, S. H., on "discs," 595
Westinghouse, 732–745
wheel efficiencies, 563
Tungsten tool steel, 141
Turning gear, turbine, 646, 773
Tuyeres, blast furnace, 30
Twin beam air pump, 992

Ultimate analysis of coal, 241
Ulysses, U. S. S., trials of, 1444
Units, absolute system of, 27, 28
c. g. s. system of, 27, 28
gravitational system of, 27, 28
heat, 180
systems of, 27
Uptake openings, 378

Vacuum and condensers, 966–1009 (*see also Condensers*)
accessories, 1001
augmenters, 998–1000
breaker, 987
effect of, on turbine performance, 710
high, advantage of, 968
limits of, 969
for recipro. eng., 969
for turbine, 970
measurement, 967
Valve, valves,
air pump, 909
back pressure, 1330
blow-down, 449, 1301
bottom blow, 1301
brass, 1303
cast steel, 1302
check, 1297
bolted bonnet, 1298, 1319
dash pot, 1299, 1320
screw bonnet, 1298, 1320
screw down, 1298
swing, 1002, 1299, 1320, 1321
union bonnet, 1299, 1320
chest, test of, 1429
classification of, 1292
cocks, 1301, 1322
cut-out, in piping, 1245
De Laval maneuvering, 762
ferro-steel, 1302
fuel oil, 314
discharge, 314
high pressure, 314
suction, 314
gate, 1299, 1321
Naval requirements, 1301
gear, turbine, 653–660
General Electric emergency, 657, 755
maneuvering, 659
globe, angle and cross, 1294, 1302–1318
Naval specifications, 1295
handwheels and wrenches, 1301, 1323
huddling chamber, safety, 1324
injection, main, 1297, 1319
iron body, 1304
Kerr emergency, 658
Ljungstrom, 781
maneuvering, 659, 748, 762, 764, 768
manifold, 1297
materials, 1293
for safety and relief, 1331
non-rising stem gate, 1299
overboard discharge, 1296, 1319
Poole stop and maneuvering, 768
practice, general, 1293
pump control, 1004
governors, 1337 (*see Pump Governors*)
reducing, 1332
inside spring, 1335
installation of, 1336
Leslie, 1000
outside spring, 1332
relief, 1324
capacity of, 1325
condenser, 1002
hot water, 1329
pump discharge, 1129
water, 1330
requirements, Emergency Fleet, 1292
Naval, 1292

Valve, rising stem gate, 1299
safety, 1324
boiler, 447, 1326
duplex, 1326
materials, 1330
requirements, 1329
testing, 1330
twin, 1326
service, oil fuel pumps, 304
stop and check, brass, 1295
iron body, 1296
main, 1295
surface blow, 1301
temperature control, 1003
temperatures allowed in, 1292
testing safety, 1330
try-cock, 447
Westinghouse, governor control, 739
wrenches and handwheels, 1301, 1323
Van Stone joints, 680, 1272
Vanadium bronze, 64
Vapor, vapors, 207
flow of (*see Flow of Steam*)
saturated, 207
expansions of, 223
superheated, 207
thermodynamic relations of, 208
used in refrigeration, 218–222
Vaporization, latent heat of, 185
Velocity, velocities,
critical, 533
composition of, 29
defined, 27
equivalents (conversion table), 16
resolution of, 29
resultant of, 30
stage turbines, 526
Ventilation, 1181
and heating (*see Heating and Ventilation*)
design of, system, 1184
exhaust system of, 1183
mechanical, 1183
methods of, 1183
natural system, 1183
Newport News practice, 1184
of coal in storage, 246
oil bunker, 275
plenum system of, 1183
velocities in mains, 1184
ventilators for, 1184
Amer. Bur. Ship. rules for, 1184
Lloyd's rules for 1184
Ventilator, 1184
cowls for, 1185
Vibration Specialty Co., balancing mach., 645
Vibrations in turbine blades, 613, 622
Viscosimeter, viscosimeters, 257–262
Engler, 258
Redwood, 258
Admirality type, 261
Saybolt Universal, 258, 1394
Viscosity, 256
absolute, 256, 261
conversion curves, 260, 261
effect of pressure on, 1381
lubricating oil, 1391–1394
oil fuel, of, 256
to test for, 258
specific, 257
-temperature curves for oil fuel, 288, 309
Voice tubes, 1377
Volatile matter in coal, 241
Volume contraction (combustion), 280
Volume equivalents (conversion table), 16
Volute chamber, centrifugal pump, 1119

Wadsworth, U. S. S., engine room, 796
. test and trials, 720, 1438
Wake gain (propellers), 1028
Warping winches, 1242
Washed coal, 244
Water, density of, at saturation pressure, 180
gland, turbine, 633, 634, 736, 744
heat conductance of, 188
per capita required, various types vessels, 1097
rate effect on boiler design, 354, 355
from indicator cards, 1412
geared turbines, 356
ideal, of turbines, 531
reduction of, 726
seal, turbine gland, 558
specific heat of, 181
test for salt in distilled, 1093
vapor, thermal properties of, 208
volume at saturation pressure, 180
Weaver, S. H., on "discs," 596
Weight (def.), 27
of air (table), 1198
Welding, 163–176
aluminum, 169
boiler repairs by, 475
brass and bronze, 169
cast iron, 168
copper, 169
dissimilar metals, 169, 170
xpansion and contraction in, 164
fluxes, 167
heavy sections, 164, 173, 175
malleable iron, 168
oxy-acetylene process, 163
rods, 133, 166
steel, 168
Thermit process, 172
torch or burner, 164
wire, 133, 166
Werkspoor Diesel engine, 957
Westinghouse bearing, turbine, 635
blading, 619
casings, 735
coupling, 611
filter, lubricating oil, 672
governor, turbine, 653
oiling system, turbine, 669
-Parsons blading, 621
shroud rings, 619
turbines, 732–745
high pressure, 763
low pressure, 734
small, 739
test data, 715
valve maneuvering, 659
Whale oil, 1383
Whistle operator, electric, 1377
steam, 1196
White metal, 864
alloys, 78
bearings, 864
Whiteheart malleable iron, 117
Whittlesey's simplified ship form, 1023